NUMERICAL ANALYSIS OF SEVERAL PORT CONFIGURATIONS IN THE FAIRBANKS-MORSE 38D8-1/8 OPPOSED PISTON MARINE ENGINE (str. 1-11)
M. I. Lamas, C. G. Rodriguez, J. D. Rodriguez, J. Telmo
Izvorni znanstveni članak

INVESTIGATION OF THE CAUSES OF MARITIME ACCIDENTS IN THE INLAND WATERWAYS OF BANGLADESH (str. 12-22)
M.R. Islam, M. D. Rahaman, N. Degiuli
Pregledni rad

THE HYDROGEN-FUELLED INTERNAL COMBUSTION ENGINES FOR MARINE APPLICATIONS WITH A CASE STUDY (str. 23-38)
Ibrahim S. Seddiek, Mohamed M. Elgohary, Nader R. Ammar
Pregledni rad

RISK EVALUATION OF PIN JIG WORK UNIT IN SHIPBUILDING BY USING FUZZY AHP METHOD (str. 39-53)
Özkok Murat
Stručni rad

A HIERARCHICAL STRUCTURE FOR SHIP DIESEL ENGINE TROUBLE-SHOOTING PROBLEM USING FUZZY AHP AND FUZZY VIKOR HYBRID METHODS (str. 54-65)
Abit Balin, Hakan Demirel, Fuat Alarcin
Stručni rad
ADDED RESISTANCE IN WAVES OF INTACT AND DAMAGED SHIP IN THE ADRIATIC SEA (str.1-14)
Ivana Martić, Nastia Degiuli, Ivan Ćatipović
Original scientific paper

THE PROCEDURE FOR CALCULATION OF THE OPTIMAL CARRYING CAPACITY OF PUSHED CONVOY BASED ON PARAMETERS OBTAINED BY EXPERIMENTS IN ACTUAL NAVIGATING CONDITIONS (str.15-28)
Ivan Škiljaica, Ilija Tanackov, Vladislav Maraš
Original scientific paper

AN EXPERIMENTAL AND NUMERICAL PREDICTION OF MARINE PROPELLER NOISE UNDER CAVITATING AND NON-CAVITATING CONDITIONS (str.29-45)
Mohammad Reza Bagheri, Hamid Mehdigholi, Mohammad Saeid Seif, Omar Yaakob
Original scientific paper

EXPERIMENTAL STUDY OF TWO LARGE-SCALE MODELS’ SEAKEEPING PERFORMANCE IN COASTAL WAVES (str.47-60)
Shu-zheng Sun, Hui-long Ren, Xiao-dong Zhao, Ji-de Li
Original scientific paper

STEAM AND SOFC BASED REFORMING OPTIONS OF PEM FUEL CELLS FOR MARINE APPLICATIONS (str.61-76)
Mohamed M. El Gohary, Nader R. Ammar, Ibrahim S. Seddiek
Original scientific paper

SHIPBUILDING PRODUCTION PROCESS DESIGN METHODOLOGY USING COMPUTER SIMULATION (str.77-91)
Marko Hadjina, Nikša Fafandjel, Tin Matulja
Professional paper
RESIDUAL HULL GIRDER ULTIMATE STRENGTH OF DOUBLE HULL OIL TANKERS (str.1-13)
Jerolim Andrić, Stanislav Kitarović, Karlo Pirić
Izvorni znanstveni članak

ANALYTICAL SOLUTION OF BASIC SHIP HYDROSTATICS INTEGRALS USING POLYNOMIAL RADIAL BASIS FUNCTIONS (str.15-37)
Dario Ban, Josip Bašić
Izvorni znanstveni članak

NEW ALGORITHM FOR OPTIMAL DIELECTRIC MATERIAL SELECTION IN MARINE ENVIRONMENT (str.39-48)
Igor Vujović, Zlatan Kulenović, Ivica Kuzmanić
Prethodno priopćenje

ANALYSIS OF THE ENERGY EFFICIENCY DESIGN INDEX WITH A PROPOSAL FOR IMPROVEMENT (str.49-59)
Predrag Ćudina
Prethodno priopćenje

SWOT ANALYSIS OF DEFICIENCIES ON SHIP COMPONENTS IDENTIFIED BY PORT STATE CONTROL INSPECTIONS WITH THE AIM TO IMPROVE THE SAFETY OF MARITIME NAVIGATION (str.61-72)
Miroslav Randić, Dario Matika, Darko Možnik
Pregledni rad

CONSIDERING ANTI-PIRACY SHIP SECURITY: CITADEL DESIGN AND USE (str.75-90)
L. Carral, Carlos Fernández Garrido, José J. de Troya, José Ángel Fraguela
Stručni rad

AN ENHANCED EQUATION FOR VIBRATION PREDICTION OF NEW TYPES OF SHIPS (str.91-117)
Valter Cergol, Peter Vidmar
Izvorni znanstveni članak
HYDRODYNAMIC PERFORMANCES OF SMALL SIZE SWATH CRAFT (str.1-22)
Ermina Begovic, Carlo Bertorello, Simone Mancini
Izvorni znanstveni članak

A COMPUTATIONAL HYDRODYNAMIC ANALYSIS OF DUISBURG TEST CASE WITH FREE SURFACE AND PROPELLER (str.23-38)
Omer Kemal Kinaci, Metin Kemal Gokce
Izvorni znanstveni članak

A REAL TIME DECISION SUPPORT SYSTEM FOR THE ADJUSTMENT OF SAILBOAT RIGGING (str.39-56)
Inmaculada Ortigosa, Julio García-Espinosa, Marcel la Castells
Izvorni znanstveni članak

STEADY STATE PERFORMANCES ANALYSIS OF MODERN MARINE TWO-STROKE LOW SPEED DIESEL ENGINE USING MLP NEURAL NETWORK MODEL (str.57-70)
Ozren Bukovac, Vladimir Medica, Vedran Mrzljak
Izvorni znanstveni članak

OPTIMIZATION PROCEDURE FOR PRELIMINARY DESIGN STAGE OF CAIRO-DAMIETTA SELF-PROPELLED GRAIN BULK SHIPS (str.71-85)
M.M. Moustafa
Stručni rad

TURKISH SHIPBUILDING INDUSTRY – CHALLENGES AND POTENTIAL (str.87-101)
Eda Turan, Hülya Cengiz
Stručni rad
<table>
<thead>
<tr>
<th>Title</th>
<th>Authors</th>
<th>Pages</th>
</tr>
</thead>
<tbody>
<tr>
<td>NUMERICAL ANALYSIS OF SEVERAL PORT CONFIGURATIONS IN THE FAIRBANKS-MORSE 38D8-1/8 OPPOSED PISTON MARINE ENGINE</td>
<td>M. I. Lamas, C. G. Rodriguez, J. D. Rodríguez, J. Telmo</td>
<td>(str.1-11)</td>
</tr>
<tr>
<td>THE HYDROGEN-FUELLED INTERNAL COMBUSTION ENGINES FOR MARINE APPLICATIONS WITH A CASE STUDY</td>
<td>Ibrahim S. Seddiek, Mohamed M. Elgohary, Nader R. Ammar</td>
<td>(str.23-38)</td>
</tr>
<tr>
<td>RISK EVALUATION OF PIN JIG WORK UNIT IN SHIPBUILDING BY USING FUZZY AHP METHOD</td>
<td>Ozkok Murat</td>
<td>(str.39-53)</td>
</tr>
<tr>
<td>A HIERARCHICAL STRUCTURE FOR SHIP DIESEL ENGINE TROUBLE-SHOOTING PROBLEM USING FUZZY AHP AND FUZZY VIKOR HYBRID METHODS</td>
<td>Abit Balin, Hakan Demirel, Fuat Alarcin</td>
<td>(str.54-65)</td>
</tr>
</tbody>
</table>
Summary

The aim of the present paper is to analyze several port configurations in the Fairbanks-Morse 38D8-1/8 marine diesel engine. The motivation comes from the high number of intake and exhaust ports which characterizes this engine. The scavenging and trapping efficiency were studied by modifying several parameters related to the ports, such as the inclination, shape, pressure and number. To this end, a numerical model based on the commercial software ANSYS Fluent was employed. Numerical results were validated using experimental measurements performed on a Fairbanks-Morse 38D8-1/8 engine installed on a submarine. The results confirmed that the influence of the port shape is practically negligible; nevertheless, an adequate combination of the inclination, pressure and number of ports can modify the scavenging and trapping efficiency noticeably.

Key words: diesel engine; scavenging; CFD (Computational Fluid Dynamics).

1. Introduction

An opposed-piston engine is an internal combustion engine in which each cylinder has a piston at both ends, and no cylinder head. These engines were employed from the end of the nineteen century with the intention of improving the performance of two-stroke diesel engines. One of its advantages is the use of uniflow scavenging, much more efficient than loop or cross scavenging. Another advantages of opposed-piston engines are that unsymmetrical port timing improves the efficiency and performance by optimizing the timings (the exhaust opening point can be advanced and the inlet closing point can be delayed), lower piston speeds because the stroke is split into two pistons, no cylinder heads (avoiding the problem of heat losses to the jacket water), no valve mechanisms (avoiding their cooling and lubricating problems), fewer moving parts, etc.

In the field of opposed-piston engines, the Fairbanks-Morse Model 38D8-1/8 is a medium-speed, two-stroke diesel engine with a design inspired on German Junkers engines.
Numerical analysis of several port configurations in the Fairbanks-Morse 38D8-1/8 opposed piston marine engine

It has consolidated its success though many decades of manufacturing and thousands of units sold for marine propulsion, electric power plants and numerous mechanical drive applications. Although it was firstly developed in the 1930s for locomotive use, it was mostly used to propel submarines in World War II. Tambor class, Gato class, Balao class, Balao class, or more recently, Tang class, are submarines propelled by this engine. Its success in submarine service made Fairbanks-Morse a strong contender for post-war diesel marine engine sales. Its commercial production began in 1938 for several uses. Concerning ship propulsion, it was employed in many civilian ships such as tugs, fishing and small vessels in general. Nowadays, more than 80 years since the first prototypes, Fairbanks-Morse opposed-piston engines are still in production, mainly in the field of marine propulsion.

When improving the efficiency of these engines, there are too involving variables. Although nowadays there are many experimental techniques, Computational Fluid Dynamics (CFD) offers an alternative method to study the fluid flow inside the cylinder. In the field of medium and large engines, CFD is especially useful because experimental setups are extremely expensive and laborious, and down-scale models sometimes are not accurate enough. Nevertheless, their CFD simulation is complicated due to the big size, which requires high computational resources. Since the nineties, advancements in computer hardware have facilitated simulations of big geometries. Nevertheless, only a few works about medium and large engines can be found in the literature. One can refer to the following authors. Nakagaya et al. [1] developed a bidimensional CFD model to study a large two-stroke diesel engine. Andersen et al. [2, 3] centered on the scavenging process of marine engines. Bigos and Puskar [4] analyzed the influence of the cylinder shape and combustion space. Sigurdsson [5, 6] examined the scavenging process of the MAN B&W 4T50MX test engine. Haider [7] developed a numerical and experimental study of the swirling flow during part of the scavenging process of a downscale model of the MAN B&W 4T50MX test engine. Lamas and Rodriguez [8] studied the scavenging process in the MAN 7S50MC and the effect of EGR and water addition during the combustion process, Lamas et al. [9]. They also analyzed the valve overlap period in the Wärtsilä 6L 46 [10], the combustion process [11], and internal modifications to reduce pollutant emissions [12]. Larbi and Bessrour [13] analyzed numerically the emissions of the Wärtsilä 6R32 and the effect of EGR [14], ammonia injection [15] and water injection [16]. Kontoulis et al. [17] analyzed the combustion and emissions in a large two-stroke marine diesel engine. Kilpinen [18] focused on NO emission in 4-stroke marine engines. Seddiek and Elgohary [19] centered on an analysis of SO\textsubscript{x} and NO\textsubscript{x} emissions.

The aim of the present paper is to develop a CFD model to optimize the scavenging process of opposed-piston engines, i.e., the action of displacing the burnt gases from the cylinder and replacing them by the fresh-air charge. Particularly, the Fairbanks-Morse 38D8-1/8 marine engine was studied. The scavenging and trapping efficiency were analyzed in terms of several port configurations. Previously to this, numerical results were successfully compared with experimental measurements performed on a Fairbanks-Morse 38D8-1/8 engine installed on a submarine.

2. Materials and methods

2.1 Case studied

For the present work, a 38D8-1/8 installed on a submarine, Fig. 1, was employed. This engine has 8 cylinders and consequently 16 pistons. Each cylinder has 16 inlet ports and 10 exhaust ports.
Numerical analysis of several port configurations in the Fairbanks-Morse 38D8-1/8 opposed piston marine engine

Lamas, M.I.; Rodriguez, C.G.; Rodriguez, J.D.; Telmo, J.

Figure 1 Fairbanks-Morse 38D8-1/8 installed on a submarine

Figure 2 outlines the geometry. As can be seen, there are two pistons per cylinder and no cylinder head. The combustion space is formed in the center of the cylinder between both pistons, which move away from each other. The exhaust ports are located near the bottom of the cylinder and the inlet ports near the top. The ports are controlled by the movement of the pistons as they reach the outer positions of their travel, thus eliminating the use of valves. When the ports are covered/uncovered by a piston, they are closed/opened respectively. The pistons controlling the air-inlet ports are located in the upper crankshaft, while those controlling the exhaust ports are located in the lower crankshaft. Both crankshafts are connected together by a vertical drive gearing. The crankshafts turn in opposite directions and the upper crankshaft has a 12-degree angle delay over the lower crankshaft, permitting asymmetrical port timing. There are two fuel injection nozzles per cylinder, directly across from each other on the liner wall, that inject fuel into a firing chamber formed by the heads of the two pistons. For marine service, the engine is built directly reversible. Naturally, port timing is not optimum during reverse operation.

Fig. 2 Cylinder geometry. Fairbanks-Morse & Co [20]
The cycle of operation begins with the movement of the pistons from their outer dead centres. Fig. 2 (a) shows the lower crankshaft past outer dead centre and the upper crankshaft at outer dead centre. At this moment, the inlet and exhaust ports are opened (uncovered by the pistons), and consequently fresh air supplied by a blower is being admitted into the cylinder and exhaust gases are being expelled. As the pistons move inward, they close (cover) the exhaust and inlet ports and the air inside the cylinder is compressed. As mentioned above, the combustion space is formed between recessed heads of the two pistons as the crankshafts approach inner dead centre, Fig. 2 (b). At approximately 9º before the lower piston reaches inner dead centre, the injection of fuel oil begins. The high temperature and pressure of the air charge causes the fuel oil to ignite and combustion takes place. The high pressures resulting from combustion force the pistons to move outward, delivering power to the crankshafts. The expanding of the gases continues until the lower piston begins to open (uncover) the exhaust ports, allowing the burnt gases to escape to the atmosphere through the exhaust system. After that, the upper piston opens (uncovers) the inlet ports and scavenging air rushes into the cylinder. Finally, the cylinder is swept clean of the remaining exhaust gases and refilled with fresh air for the next compression stroke.

2.2 Numerical implementation

The engine described in the previous section was studied numerically. Particularly, the scavenging process was studied along a period of time from 90º lower crankshaft angle to 270º lower crankshaft angle.

Concerning the governing equations, the turbulent flow is governed by the conservation of mass, momentum and energy. Besides, one more equation is needed to characterize the local mass fraction of air and burnt gases. Turbulence was simulated by the \( k-\varepsilon \) model because it is robust, economic and reasonably accurate for a wide range of CFD cases.

All CFD models require initial and boundary conditions. Concerning the pressures, their values were measured experimentally. The initial (90º lower crankshaft angle) in-cylinder gauge pressure is 3.2 bar, the exhaust gauge pressure is 0.1 bar and the inlet gauge pressure, given by a lobe rotary compressor, is 0.4 bar. The inlet temperature was also experimentally measured, corresponding to 325 K. The inlet pressure, exhaust pressure and inlet temperature were provided by the engine itself. Nevertheless, the in-cylinder pressure was measured by a pressure sensor Leutert DPI-F2, which provided the PV diagram.

Concerning the heat transfer from the cylinder to the cooling water, a combined convection-radiation type was assumed:

\[
q = h(T_{\text{gas}} - T_{\text{water}})
\]

where \( q \) is the heat transferred, \( T_{\text{gas}} \) the in-cylinder temperature, \( T_{\text{water}} \) the cooling water temperature, 81.2ºC according to experimental measurements, and \( h \) the heat transfer coefficient (due to convection + radiation), given by the following expression, Taylor [21]:

\[
h = 10.4kb^{-1/4}(u_{\text{piston}}/\nu)^{3/4}
\]

where \( b \) is the cylinder bore, \( k \) the thermal conductivity, \( u_{\text{piston}} \) the mean piston speed, and \( \nu \) the kinematic viscosity. Substituting values into the equation above yields \( h = 3593 \text{ W/m}^2\text{K} \).

As all cylinders are identical, only one of them was simulated. The CAD 3D design of the cylinder and ports was realized using Solid Edge ST software. Concerning the pistons, they were not modeled because only the surfaces in contact with the fluids are needed for the computations.
The mesh was created by Gambit 2.4 and then exported to ANSYS Fluent 6.3. In order to implement the movement of the pistons, a deforming mesh was used. As it was verified that hexahedral elements provide better accuracy and stability than tetrahedral ones [22], a structured hexahedral mesh was adopted. In order to take into account the boundary layer, a finer mesh was employed near the walls. The mesh motion is shown in Fig. 3, in which the arrows indicate the direction of the pistons. The number of cells varies from 50000 to 350000.

Figure 3 (a) represents the start of the simulation, 90° lower crankshaft angle. As can be seen, the ports are uncovered by the piston and thus there is no fluid flow between the cylinder and ports, and the pistons start to move outward. Figure 3 (b) represents the position 200° lower crankshaft angle. At this moment, the ports are covered by the piston and thus there is fluid flow between the cylinder and ports, remaining opened, and the pistons move inward. Finally, Fig. 3 (c) represents the end of the simulation with no fluid flow and the pistons moving inward.

ANSYS Fluent is based on the finite volume method. The PISO algorithm was chosen for pressure-velocity coupling. A second order scheme was employed to discretize the continuity, momentum, energy and mass fraction equations. The time derivatives were discretized through a first order fully implicit scheme. Both mesh size and time step were analyzed to ensure that suitable values were employed.

3. Results and discussion

First of all, this section describes the results and validation of the model. After that, the scavenging and trapping efficiency are studied against several port configurations.

3.1 Mass fraction, pressure and velocity fields

Figure 4 shows the mass fraction field of air for several values of the lower crankshaft angle. It was represented the start of the simulation (90°), end of the simulation (270°), outer dead center of the lower crankshaft (180°), outer dead center of the upper crankshaft (192°) and other representative instants. It can be observed that initially the cylinder is full of burnt gases, Fig. 4 (a) and the pistons are moving outward, as indicated by the arrows. There is no any mass fraction passing inside the chamber as all ports are closed (covered by the pistons). After the exhaust ports are opened (uncovered), Fig. 4 (b), the burnt gases begin to flow out. Thereafter, the inlet ports are opened (uncovered) and consequently pressurized air enters the chamber and helps to throw the burnt gases away, Fig. 4 (c-d). After reaching 192°, the pistons move inward. Air continues flowing into the cylinder and burnt gases continue
flowing out, Fig. 4 (e-f). Finally, the inlet ports and then exhaust ports are closed (covered by the pistons), and at the end of the simulation, Fig. 4 (g), the cylinder contains a mixture of air and a small amount of residual gases which were not expelled.

**Fig. 4** Mass fraction field of air for several lower crankshaft angles

Figure 5 shows the velocity field. It can be seen that, when the exhaust ports are opened, Fig. 5 (a), burnt gases are expelled. When all ports are opened, Fig. 5 (b), air is admitted into the cylinder through the inlet ports and burnt gases are expelled through the exhaust ports. The pressure difference from the top to the bottom zones of the cylinder causes a motion from the intake to the exhaust ports. As can be seen in Fig. 5 (c), Section A-A, the inlet ports have an inclination of 20.9º around the cylinder axis. This promotes a swirling motion which is vanished as the fluid descends in the direction of the exhaust ports, shown in Section B-B.

**Fig. 5** Velocity field (m/s). (a) 130º lower crankshaft angle; (b) 200º lower crankshaft angle; (c) transversal sections at 200º lower crankshaft angle. → 200 m/s

**Fig. 6** In-cylinder gauge pressure (bar). EPO: exhaust ports opening; IPO: inlet ports opening; EPC: exhaust ports closing; IPC: inlet ports closing
Numerical analysis of several port configurations in the Fairbanks-Morse 38D8-1/8 opposed piston marine engine

Figure 6 shows the time history of the in-cylinder average gauge pressure. The calculated pressure was validated with experimental measurements, which also are represented in this figure. A good agreement between numerical and experimental results was obtained, being the average error 5.6%. Many factors are responsible for the difference between numerical and experimental results. CFD itself implies an error in the mesh generation and discretization process. The hypotheses assumed to simplify the computations constitute another source of errors.

3.2 Analysis of the scavenging and trapping efficiency

The scavenging efficiency indicates to what extent the burnt residuals have been replaced with fresh charge. It is computed as the mass of delivered air that was trapped by comparison with the total mass of charge that was retained at exhaust closure, Eq. (3). The present CFD simulation provided a value of 86.2% for the parameters studied. This value satisfactorily agrees with the 90% indicated elsewhere for this engine, Schweitzer [23].

\[
SE(\%) = \frac{\text{mass of delivered air retained}}{\text{mass of mixture in the cylinder}} \times 100
\]

On the other hand, the trapping efficiency indicates, from the air mass inducted in the cylinder, the fraction really retained. The CFD simulation provided 77.6% for the conditions studied.

\[
TE(\%) = \frac{\text{mass of delivered air retained}}{\text{mass of delivered air}} \times 100
\]

It is interesting to analyze the effect of some parameters on the scavenging and trapping efficiency. The first parameter studied was the intake ports inclination, employed in engines with uniflow scavenging in order to improve the scavenging process. The inclination induces a swirling flow which minimizes the amount of residual gases that remain trapped without being expelled.

As stayed above, the intake ports are inclined 20.9°. The results of scavenging and trapping efficiency for other values are represented in Fig. 7. As can be seen, as the inclination is increased the scavenging efficiency increases and then decreases. The reason is that the intake ports inclination optimizes the orientation of the flow inside the cylinder. Nevertheless, an excessive swirl promotes an inadequate movement which reduces the scavenging efficiency. For the same reason, the trapping efficiency also increases and then decreases with the inclination of the intake ports.
Numerical analysis of several port configurations in the Fairbanks-Morse 38D8-1/8 opposed piston marine engine

Fig. 7 Scavenging and trapping efficiency against the intake ports inclination

Another interesting parameter is the intake pressure given by the compressor. For this engine, the intake pressure is 0.4 bar. Fig. 8 shows the scavenging and trapping efficiency for other values of the intake pressure. As can be seen, the scavenging efficiency increases with pressure. The reason is that high pressures promote a more intense entrance of air from the inlet to the exhaust and consequently a higher amount of air is retained inside the cylinder. On the contrary, the trapping efficiency decreases with the inlet pressure because the amount of delivered air is highly increased with the intake pressure. High inlet pressures promote a high quantity of delivered air and, consequently, a lower trapping efficiency.

Fig. 8 Scavenging and trapping efficiency against the inlet pressure

Another aspect is the port shape. For the purpose three shapes were studied, rectangular, circular and elliptical, Fig. 9. At equal areas, the scavenging and trapping efficiency against the intake and exhaust ports shape are indicated in Table 1. As can be seen, the effect of the shape is almost negligible. The circular port has low drag losses due to its low relation perimeter/area. The most similar port is the elliptical, followed by the rectangular. For this reason, the scavenging efficiency is higher for the circular intake port shape, followed by the elliptical and rectangular. The low drag of the circular shape allows air to enter the cylinder easily and consequently increases the amount of air which remains trapped. On the contrary, the trapping efficiency is lowest for the circular intake port shape, followed by the elliptical and rectangular. The reason is that a higher amount of entering air implies a reduction of the trapping efficiency. Concerning the exhaust ports shape, the scavenging and trapping efficiency are highest for the circular shape, followed by the elliptical and rectangular ones.
The reason is that the amount of air retained in the cylinder is higher as the drag at the exhaust ports increases.

Table 1 Efficiency against the shape of the ports

<table>
<thead>
<tr>
<th></th>
<th>Rectangular</th>
<th>Circular</th>
<th>Elliptical</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake ports</td>
<td>SE (%)</td>
<td>86.2</td>
<td>86.5</td>
</tr>
<tr>
<td></td>
<td>TE (%)</td>
<td>77.6</td>
<td>77.3</td>
</tr>
<tr>
<td>Exhaust ports</td>
<td>SE (%)</td>
<td>86.2</td>
<td>85.9</td>
</tr>
<tr>
<td></td>
<td>TE (%)</td>
<td>77.6</td>
<td>77.4</td>
</tr>
</tbody>
</table>

Fig. 9 Exhaust port configurations. (a) rectangular; (b) circular; (c) elliptical

Finally, the last parameter studied is the number of ports. At equal total area, Figs. 10 and 11 show the scavenging efficiency and trapping efficiency against the number of intake and exhaust ports respectively. As can be seen, the scavenging and trapping efficiency increase with the number of both intake and exhaust ports. The reason is that the fluid is better directioned as the number of ports is increased. It is important to note that the number of the intake ports is more important than the number of the exhaust ports because the former is related to the swirl movement necessary to scavenge. Nevertheless, the exhaust ports have a low influence in the swirl movement. The trapping and scavenge efficiency decreases with the number of exhaust ports because it is easier the emerging of flow from the cylinder.
4. Conclusions

In the present work, a CFD analysis was developed to study the scavenging process of the Fairbanks-Morse 38D8-1/8 opposed-piston marine diesel engine. The results were satisfactory compared with experimental measurements carried out in an engine installed on a submarine. In general, this study shows that CFD predictions yield reasonably accurate results that can help to improve the knowledge of the flow characteristics at the scavenging process. Particularly, an error of 5.6% was obtained in the in-cylinder average pressure.

Once validated, this numerical model was employed to study the scavenging efficiency in terms of several modifications of the exhaust and intake ports. It was verified that the influence of the port shape is practically negligible. Nevertheless, the inclination, pressure and number of ports can modify the scavenge and trapping efficiency noticeably.

5. Acknowledgements

The authors would like to express their gratitude to “Talleres Pineiro, S.L.”, sale and repair of marine engines.

REFERENCES


Numerical analysis of several port configurations in the Fairbanks-Morse 38D8-1/8 opposed piston marine engine


Submitted: 14.01.2015. Lamas, M.I., isabellamas@udc.es, University of A Coruña
Escuela Politécnica Superior. C/Mendizábal s/n. 15403 Ferrol, A Coruña, Spain

Accepted: 02.03.2105. Rodriguez, C.G., c.rodriguez.vidal@udc.es, University of A Coruña
Rodríguez, J.D., jdedios@udc.es, University of A Coruña
Telmo, J., javier.telmo@usc.es, University of Santiago de Compostela
INVESTIGATION OF THE CAUSES OF MARITIME ACCIDENTS IN THE INLAND WATERWAYS OF BANGLADESH

UDC 629.5:629.552:629.5.017.25
Review paper

Summary

Water transport is the proven cheapest and safest mode of transportation but it is the agent of catastrophe in Bangladesh. The overall scenario of Bangladesh inland water transport has been studied. The form of occurrences of maritime accidents in the inland waterways of Bangladesh may be categorized based on mode of failure. Data analysis of major accidents shows that prevention of passenger vessels’ accident will drastically reduce the number of casualties in Bangladesh Inland Waterways where the two repeatedly reported causes of accidents are overloading and inclement weather. Literature review shows that analyses were carried out mostly to investigate the mechanism of capsizing due to violation of “The Inland Shipping Laws and Rules”. To enlighten the roles of professionals like Naval Architects and Law Enforcing Agencies, the reasons behind the accidents in Bangladesh Inland Waterways have been simplified and the nature of actions required for preventing the accidents have been identified from practical point of view. The effect of consideration of overloading condition and higher wind pressure in design has been studied and found that it will have adverse effect on the socio-economical condition of Bangladesh. Role of proper design and construction has been identified by dividing the accidental phenomenon into two phases, capsizing and sinking of vessels.

Key words: inland waterways; maritime accidents; Bangladesh; overloading; inclement weather

1. Introduction

Bangladesh is a riverine country. The river network of Bangladesh as the most important transport artery in the country's communication sector plays a vital role in national life. Almost all big cities, towns and commercial centers of the country grew up on the banks of its rivers. Figure 1 shows the river network of Bangladesh. Until now waterways is the principal mode of transporting goods and passengers in the southern region of the country. In the rest of the country where the places are inaccessible by land transport like roads and rail, water transport acts as the gateway to communication.
Since independence in 1971 movements of goods and passengers from the districts of the southern region of Bangladesh have increased and water transport is the only mode of transport to the capital city, Dhaka. Inter-district and intra-district movements within the southern districts of Barisal, Bhol, Patuakhali, Jhalokathi, Borguna, Pirozpur and their sub-districts are by water transport. These districts and sub-districts of southern Bangladesh are connected to the capital city, Dhaka with water transport although very few places are now accessible by all-weather road transport.

There is single decker; double-decker and triple-decker motor launches that ply on different destinations daily. Moto launches travelling to near destinations around the capital city are smaller in size and they make more than one return trip throughout the day. Large motor launches going to distant locations have specific time schedules. Also, mechanized country boats and steel bodied small-mechanized boats have dramatically changed the pattern of inland water transportation system. These boats use engines, which are normally intended for agricultural use. The numbers of such boats are unknown, but believed to be in tens of thousands. There are two types of such vessels i.e.:

a. old fashioned wooden country boats with propulsion fitted,
b. large dingy boats of simple hard chine flat bottom steel or wooden body construction.

The government has exempted these boats from the requirements of the Inland Waterways Act. There have been numerous accidents involving such vessels. Until now, Inland Maritime
Administration Authority of Bangladesh has found no means of regulating these boats and minimize accidents.

With the increase in population and the growing economy of the country, the waterways are getting congested despite the fact that the inland waterways are not expanding. Figure 2 shows the number of vessels registered for transporting various types of goods and passengers in the inland waters of Bangladesh [1]. Therefore, problems relating to maritime safety are emerging with new dimensions every day. In Bangladesh, maritime safety has become a severe issue when a number of passenger launch accidents killed several thousands of people within the past few years. In response to such emergencies, the government took minimum remedial measures. Therefore, accidents in inland waterways of Bangladesh are very common. Also, the extent of damages and losses of property are tremendously expensive which severely puts substantial amount of burden on the national economy. There yet remain numerous deficiencies on maritime safety and the scope for improvements in this aspect is a contemporary demand.

![Fig. 2 Number of different types of vessels registered in Department of Shipping (DOS), Bangladesh for inland waterways.](image)

Present investigation deals with the reasons behind the accidents particularly the passenger vessels in Inland Waterways of Bangladesh and identification of the measure that are needed for preventing the accidents.

2. Characteristics of inland waterways

Bangladesh with 24000 km waterways has a navigable network varying from 5968 km during the monsoon to 3865 km during the dry season. According to the traffic and economic importance, the inland waterways have been classified into four classes of routes [2]. In these routes various navigational aids are provided to make night-navigation possible.
Class I: Arterial/trunk routes maintain at a depth of 3.6 m throughout the year. Length of Class I route is 683 km or equivalent to only 11 per cent of the total network and links the two main seaports of Bangladesh Chittagong and Mongla and the inland ports of Dhaka, Barisal and Khulna.

Class II: Secondary routes maintain at a depth of at least 1.8 m throughout the year. This class of route comprises kilometres of secondary route or equivalent to only 17 per cent of the total river network and is kept open for passenger launches and cargo barges.

Class III: These are feeder routes of regional importance. The mean depth of the route is 90 cm, which is equivalent to 32 per cent of the total network, and having a length of 1885 km.

Class IV: Seasonal routes of less than 90 cm depth. Its total length of 2400 km is equivalent to 40 per cent of the total network.

Minimum vertical and horizontal clearance of these river routes are described in Table 1.

<table>
<thead>
<tr>
<th>Name of Route</th>
<th>Minimum Vertical Clearance</th>
<th>Minimum Horizontal Clearance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Class- I</td>
<td>18.30 m</td>
<td>76.22 m</td>
</tr>
<tr>
<td>Class- II</td>
<td>12.20 m</td>
<td>76.22 m</td>
</tr>
<tr>
<td>Class -III</td>
<td>7.62 m</td>
<td>30.48 m</td>
</tr>
<tr>
<td>Class -IV</td>
<td>5.00 m</td>
<td>20.00 m</td>
</tr>
</tbody>
</table>

3. Regulatory framework of inland navigation

There are two authorities responsible for the management of water transport sector of Bangladesh:

- Department of Shipping (DOS) and
- Bangladesh Inland Water Transport Authority (BIWTA).

Department of Shipping (DOS) is responsible for:

- Administer the national and international standards of shipping to the inland and seagoing ships through survey and inspection.
- Administer the standards for the seafarers by conducting examination and certification for various grades of examination for inland and international shipping.
- Monitor after survey conditions of the vessels and take appropriate steps to check any unlawful practices and ensure safe operation of watercraft across the country.
- Administer the Bangladesh Flag Vessels (Protection) Ordinance 1982 and the rules made there under.

As per Section 15 of the Ordinance E. P. Ordinance No. LXXV of 1958; amendment Ordinance No. LV of 1977, the Bangladesh Inland Water Transport Authority (BIWTA) performs the following principal statutory functions of development, maintenance and
regulatory nature:

- Carry out river conservancy works including river training works for navigational purposes and for provision of aids to navigation including marks, buoys, lights and semaphore signals.
- Disseminate navigational and meteorological information including publication of river charts.
- Provided pilotage and hydrographic survey services.
- Draw up programmes of dredging requirements and priorities for efficient maintenance of existing navigable waterways and for resuscitation of dead or dying rivers, channels, or canals, including development of new channels and canals for navigation.
- Fixation of maximum and minimum fares and freight rates for Inland Water Transport on behalf of the Government.
- Approve timetables for passenger launch services.
- Act as the Competent Authority of Bangladesh for the protocol on Inland Water Transit and Trade, looking after the use of waterways of Bangladesh on behalf of the Govt. of Bangladesh for the purpose of trade and transit between Bangladesh and India as provided in the Protocol.

4. **Accidents in inland waterways**

In present study, accidents data are collected from various sources such as Daily Newspapers, reports of Department of Shipping (DOS) and Bangladesh Inland Water Transport Authority (BIWTA). Statistics of death toll in waterways shows a large number of fishermen die each year due to sailing in bad weather. That toll has also been excluded from the data presented in year-to-year comparison plot.

![Year wise inland maritime accidents in Bangladesh](image3)

**Fig. 3** Year wise inland maritime accidents in Bangladesh [3]

![Year wise death tolls due to inland maritime accidents in Bangladesh](image4)

**Fig. 4** Year wise death tolls due to inland maritime accidents in Bangladesh [3]
In Figures 3 and 4, the column height is comparatively greater in the year of 1986, 1994, 2000, 2002, 2003, 2005 and 2009. The reasons behind the relatively higher death number in those four years are the major accidents shown in Table 2. Here accident level has been defined on the basis of no. of death toll. Table 2 shows that the types of all the vessels suffered the major accidents are passenger carriers.

In year 2014, two serious passenger vessels named M.L. Pinak-6 and M.V. Miraz-4 sank in the inland waterways of Bangladesh, which causes death tolls of 47 and 56 passengers respectively.

<table>
<thead>
<tr>
<th>Year</th>
<th>Total death</th>
<th>Vessel (Type)</th>
<th>Reported Cause</th>
</tr>
</thead>
<tbody>
<tr>
<td>1986</td>
<td>426</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>1994</td>
<td>303</td>
<td>M.V. Dinar – 2 (passenger)</td>
<td>Turbulence water &amp; over loading</td>
</tr>
<tr>
<td>2000</td>
<td>353</td>
<td>M.V. Salauddin -2 (passenger)</td>
<td>Northwestern</td>
</tr>
<tr>
<td>2002</td>
<td>297</td>
<td>--</td>
<td>--</td>
</tr>
<tr>
<td>2003</td>
<td>464</td>
<td>M.V. Mitali-3 (passenger)</td>
<td>Northwestern</td>
</tr>
<tr>
<td></td>
<td></td>
<td>M.V. Nasrin-1 (passenger)</td>
<td>Turbulence water &amp; over loading</td>
</tr>
<tr>
<td>2005</td>
<td>248</td>
<td>M.V. Moharaj (passenger)</td>
<td>Northwestern</td>
</tr>
<tr>
<td>2009</td>
<td>260</td>
<td>--</td>
<td>--</td>
</tr>
</tbody>
</table>

After every launch accident, which often takes a heavy toll of human life, the government immediately exhibits promptness, forming multiple high-level probe committees, promising punishment for the responsible persons behind the accident. It has been reported that over the last four decades more than 800 investigation committees have been formed to investigate over 500 inland waterway incidents. But no probe committee comes with the exact reasons behind the accidents. It is because of:

- Inclusion of official or officials from Department of Shipping (DoS), government regulatory body for registration of inland waterways crafts or, Bangladesh Inland Water Transport Authority (BIWTA), government regulatory body for maintenance of inland waterways who are involved in a vessel's registration or, survey and fitness checking or, inspecting the launch before the journey at the ghat, was included as a member in the investigation committee of that particular vessel's incident.
- Non-inclusion of experts particularly in the field of naval architecture; as a result the engineering or, technical elements behind any launch accident always remain out of focus.

Therefore, it is reasonable to emphasize on accidents of passenger vessels, as prevention of these accidents will drastically reduce the number of casualties in Bangladesh inland waterways accidents.
5. **Analysis of causes of accidents**

For simplifying the causes of accidents in the inland waterways of Bangladesh, the most common form of occurrences are divided into two groups based on the mode of failure as structural failure and stability failure as shown in Figure 5.

When accident occurs due to collision, grounding or fatigue, structure of vessel fails and when accident occurs due to overloading or inclement weather, vessel loses its stability in some manner. Structural failure always leads to stability failure, but stability may fail without the failure of structure.

![Mode of failure and form of occurrence of accidents in waterways](image)

Accidents may occur with different combinations of all the form of occurrences. The most common combined form in Bangladesh is overloading with inclement weather as shown in Figure 6. After accomplishing some voyage, it cannot be said that accident has occurred due to only faulty design or construction. Faulty design or construction acts as catalyst of accidents increasing the level of catastrophe.

![Combination of form of occurrences](image)

In Bangladesh, most of the time passenger vessels ply with overloading which makes them unstable. If they do not suffer any disturbance they can finish voyage safely and no one cares about that. It is rare that overloading in single form acts as the cause of accidents. Commonly storm, strong current, water turbulence combines with overloading. On the other hand inclement weather can act as a single form of occurrence of accidents.
Analysis of accident data indicates two major repeatedly reported causes of accidents in Bangladesh inland waterways mostly in combined form namely overloading and inclement weather.

5.1 Overloading

A safe design constructs a stable vessel. But in overloading condition, vessel does not remain safe making the vessel unstable. If an unstable vessel inclines up to a certain angle (flooding angle), it will not be able to return to its upright condition rather it will incline more resulting loss of stability and finally it will capsize. On the other hand, if inclination does not take place it may finish its voyage. That is why in all the cases of overloading, accidents do not take places. Inclination up to the flooding angle indicates presence of another form of occurrence like inclement weather or water turbulence.

In designing any vessel, usually four loading conditions are considered and it is to be confirmed that the vessel satisfies all the stability criteria at those conditions, namely:

1) Full Load Departure Condition,
2) Full Load Arrival Condition,
3) Departure Condition without Load,
4) Arrival Condition without Load.

Investigating the mechanism of capsizing due to overloading, Iqbal et al. [5] proposed to consider the departure and arrival conditions for some level of overloading in addition with the stated four conditions. Consideration of overloading condition in design means keeping a margin of safety up to a certain limit. In case of the existing traditional design of Bangladesh inland passenger vessels, the ratio of lateral area above the load water line (exposed to wind) to the underwater part is large and the down flooding angle is less due to opening of engine room. Due to this, the stability criteria are satisfied minimally i.e., there remains no margin for safety.

In this circumstances, if for instance 150% loading condition is to considered, in design 100% loading condition in which the stability criteria is satisfied marginally will simply converge to 150% loading condition and the previous 67% (more or less) loading will then be considered as 100% loading condition. So it is rational to say that the practical impact of considering overloading condition up to any level in design it will be: “If all other parameter remains unchanged, declaration of the passenger capacity in registration certificate will be decreased by some amount depending on how much overloading has been considered in design and due to less amount of passenger capacity declaration, the vessel owners legal earning will decrease all through the year”. On the other hand overloading has no relation with capacity declaration or the registration certificate. In fact overloading is not a problem to solve; it is a matter to prevent – by any means, at any cost. As per Iqbal et al [6] “Overloading is not a pure Naval Architecture problem”.

5.2 Inclement Weather

In case of inclement weather, strong wind creates pressure on lateral area of superstructure or lateral area of the vessel exposed to weather, which tends to incline the vessel in Figure 8. For inclining, the vessel has to overcome the resistance of water exerted on the underwater volume. Strong current due to inclement weather tends to roll the ship. It is clear that ratio of underwater volume to lateral area of superstructure plays a vital role in the stability of passenger vessels. That is why less superstructure and large underwater volume makes the sand carriers highly stable in full load condition as wind gets very less area to
create pressure and on contrary the water resistance against the inclination of vessel is very high.

![Image](Lateral AREA of Superstructure)

**Fig. 8 Lateral area of superstructure**

Equating the moment due to wind pressure and moment due to water resistance, a lever has been defined as wind lever. Passenger Vessels Stability under Wind Pressure largely depends on this wind lever:

$$\text{The Wind Lever} \ l_w = \frac{PAZ}{D}$$

where:

- $P$ - wind Pressure that determines the vessels limit of sustainability under strong wind.
- $A$ - lateral area of superstructure,
- $Z$ - distance between vertical center of buoyancy and centroid of the lateral area,
- $D$ - displacement.

As per Weather Criterion of the Inland Shipping Stability Rules, 2001 of Bangladesh [4], maximum allowable/ permissible wind pressure is equal to 0.0322 t/m$^2$.

Designers’ requirement is to keep the value of wind lever as low as possible. As per The Inland Shipping Stability Rules, 2001 passenger vessels are not permitted to ply at a wind speed more than 10 m/s and on the basis of this wind speed the wind pressure has been defined as 0.0322 t/m$^2$. Several Accidents have been occurred due to inclement weather when the wind pressure was more than the defined value (0.0322 t/m$^2$). Greater wind pressure at the time of accident means greater wind speed at which the vessel was not supposed to sail. Preventing vessels from sailing at a wind speed greater than the design wind speed is not a pure Naval Architecture problem.

Iqbal et al [4] has proposed to consider greater wind pressure in design so that a vessel can sustain when that higher pressure will be created by the stormy wind. Statistical analysis shows, passenger vessels of length around 40 m suffered accidents due to Inclement Weather/Northerner. For accommodating (for ensuring the same level of stability) higher wind pressure ($P$) in design of passenger vessels having specific length in order to ensure sustainability at the higher wind pressure a designer can increase displacement by increasing breadth. As for example ferry for vehicle transportation usually has a larger breadth, which makes it highly stable. If breadth of a vessel increases, water resistance also increases, which in turn increases fuel consumption along with the necessity of twin screw that is double engine. Naturally vessel owners always try to avoid this situation.

Another way of increasing the displacement of a vessel is to increase draught. But this dimension cannot be increased due to shallow water depth in the river routes.
The only way left for accommodating higher wind pressure to a designer that is to a naval architect is to decrease volume of superstructure which will decrease the value of $A$ and $Z$ in the equation of wind lever. The passenger vessel owners always have a tendency to modify their vessels by increasing the superstructure volume whether the design permits or not. That is, their tendency is to make their vessel more economical. The reason behind is either design criteria (wind pressure of 0.0322 t/m$^2$) is such that it does not produce economical vessel from the owners point of view or the profit the vessel owners want to make is undue profit.

On the other hand, the greater wind pressure is the criteria of extreme case and all the vessels will not suffer that pressure very frequently.

The passenger vessel owners can be offered only two available options:

- keeping superstructure volume less,
- stop doing business in stormy weather.

If the first one is applied in design, the owners earning will reduce all through the year whether the vessel faces any stormy weather or not. So there is a high possibility that the vessel owners will choose the second option.

For that obviously they have to know the weather forecast well before the incident and there must be some authority to ensure that the vessel owners are doing what they are saying. Reducing the superstructure volume and rescheduling passengers fare accordingly may not be a feasible solution considering the socio economic condition of mass people.

So the only feasible solution for preventing accidents in stormy weather is that the passenger vessels, which were designed based on the weather criterion of The Inland Shipping Stability Rules, 2001 must have to be stopped from sailing at a wind speed more than 10 m/s.

5.3 Role of proper design and construction

For a clear understanding of the role of proper design and construction, an accidental phenomenon can be divided into two phases:

- Phase- I: Capsizing,
- Phase- II: Sinking.

A vessel, which is properly constructed based on, a proper design may capsize if design criteria are not followed. On the other hand, prevention of capsizing of vessels will automatically prevent the sinking of vessels.

When a vessel is capsized due to accident, it is the design and construction, which is solely responsible for sinking or quick sinking of the vessel. The role of proper design and construction is to save the lives of ill-fated passengers by at least delaying the sinking of the vessel. Without the ease of access, only provision of sufficient lifesaving appliances in design and construction cannot ensure the survival of passengers till the moment of the arrival of rescue party. Not only a proper design but also an improved design is required for preventing the sinking of capsized vessels. Research in this area is yet to be done.

6. Conclusions and recommendations

Prevention of passenger vessels’ accident will drastically reduce the number of casualties in Bangladesh Inland Waterways. Accidental causes should be analyzed from two different viewpoints, causes of capsizing and causes of sinking. Following the design criteria
Investigation of the Causes of Maritime Accidents in the Inland Waterways of Bangladesh

Muhammad Rabiul Islam, Md. Mashiur Rahaman, Nastia Degiuli

strictly can prevent capsizing of vessels, which will automatically prevent the sinking of vessels. For preventing sinking of a capsized vessel not only proper designs but also improved designs are essential. It is the “Law Enforcing Agencies” who can stop the periodical accidents in Bangladesh waterways at once by preventing the violation of “The Inland Shipping Laws and Rules”.

As safe design does not remain safe if design conditions are not followed, only two recommendations can be made for preventing the capsizing of vessels in accidents:

a) Prevention of overloading is the only practical solution other than consideration of overloading in design;

b) Prevention of sailing in inclement weather (more than 10 m/s wind speed as per present law of Bangladesh Inland Shipping) is the feasible solution other than consideration of inclement weather in design due to socio-economical aspect.

REFERENCES


Submitted: 14.12.2014. Accepted: 01.03.2015

Muhammad Rabiul Islam
Naval Architecture & Marine Engineering Department
Military Institute of Science and Technology
Dhaka, Bangladesh

Md. Mashiur Rahaman
Department of Naval Architecture & Marine Engineering
Bangladesh University of Engineering & Technology
Dhaka-1000, Bangladesh
mashiurrahaman@name.buet.ac.bd

Nastia Degiuli
Faculty of Mechanical Engineering and Naval Architecture
University of Zagreb
Ivana Lucica 5, 10000 Zagreb
Croatia
THE HYDROGEN-FUELLED INTERNAL COMBUSTION ENGINES FOR MARINE APPLICATIONS WITH A CASE STUDY

UDC 629.5.026:662.769.2
Review paper

Summary

Modern marine power plants have been designed to improve the overall ship’s efficiency. This pushed the designers of marine machinery to search for unconventional fuels for these plants. During the previous years, diesel oil has been extensively used on-board ships. Due to the high price of light diesel oil and the environmental problems resulting from the use of heavy fuel oil, it has become necessary to search for an alternative to traditional fuels. As a result, natural gas fuel has been used on-board some types of ships, especially short-voyage cruise ships. Unfortunately, there are still some technical and logistic problems related to the use of natural gas as a fuel, especially as it is considered a non-renewable energy source. The use of hydrogen fuel on-board ships, particularly in modern power plants may contribute to overcoming the above problems. The present paper considers the possibility of the use of hydrogen fuel for marine applications and discusses different stages of hydrogen gas cycle beginning with hydrogen generation process from clean energy until using it as fuel for internal combustion engines on-board one RO/RO ship, named Taba, operating in the Mediterranean Sea. Compared to the diesel engine, the hydrogen fuelled engine is found to be lower in thermal efficiency and fuel consumption, however, some adjustments are needed.

Key words: alternative fuels; hydrogen engine; hydrogen storage; ship’s emissions

1. Introduction

Marine fuel plays a key role in operation of power plants onboard ships. The latest years have seen difficult challenges against the use of fossil fuels in marine applications due to the environmental damage caused by these fuels. That pushed the International Maritime Organization (IMO) to issue a number of regulations to reduce this effect [1]. Thus, in 2005 the amendments to the Marine Pollution Convention (MARPOL) were issued, i.e. Annex Six of MARPOL, to reduce air pollution from ships. The requirements of Annex Six establish limits on ship emissions. Some of these limits concern the permissible percentage of fuel elements content such as sulphur content [2] while some refer to the percentage of harmful
Ibrahim Seddiek, Mohamed Elgohary, Nader Ammar

The hydrogen-fuelled internal combustion engine for marine applications with a case study

Pollutants emitted from ships such as nitrogen oxides [3]. These regulations pressed all interested in the maritime field to consider potential alternatives to reduce dependence on fossil fuels [4-6] and search for alternative types of fuels [7]. Thus, many researchers studied the possibility of using some of alternative fuels, mostly liquefied natural gas [8-10]. Moreover, other researches pointed out the feasibility of using other types of alternative fuels such as methanol [11] and hydrogen (GH2) [12] for special marine application. On the other hand, it was shown by Banawan, et.al [13] that the main obstacles facing the reliance on marine alternative fuels are: availability, cost, reliability, safety and the compliance with IMO regulations. Among alternative fuels, hydrogen is considered to be more environmental friendly and renewable. Despite the safety risks of using hydrogen gas onboard ships, several researches proved the possibility of using it especially for power generation produced by fuel cells [14]. The problem arising now is searching for marine alternative fuels that can be produced through clean energy in order to prevent any further environmental damage caused by production process. The present paper considers the various steps of using hydrogen as an alternative marine fuel, including its production, storage, fuel system, and finally its application in internal combustion engines which represent more than 95% of marine power plants onboard ships. Also, the paper presents a brief introduction to solving the first step of calculations in the problem of Hydrogen Internal Combustion Engines (HICE) designing. A computer model, Engineering Equation Solver (EES) software [15], is used in solving the problem of designing the marine hydrogen engine. Hydrogen gas turbine design can be benefited by using the advantages of the computer programs; which was illustrated in authors’ previous work [16-17].

2. Hydrogen production

Hydrogen can be produced from a number of sources both renewable and non-renewable by various processes. At present, a large amount of hydrogen is produced by reforming of hydrocarbons. However, in order to minimize the reliance on conventional fuels, considerable developments in other GH2 production technologies from renewable resources have been made [18]. The following sections give a short description of hydrogen production methods with emphasis on hydrogen production from clean energy sources.

2.1 Hydrogen from fossil fuels

This method depends on converting the hydrogen-containing materials derived from fossil fuels into a hydrogen-rich gas. Fuel processing of methane is considered to be the most common commercial hydrogen production technology today. By this method hydrogen gas can be produced through three basic technologies: partial oxidation, auto thermal reforming and steam reforming [19]. A major drawback of these technologies is that they produce a huge amount of carbon monoxide (CO). Consequently, additional steps to remove CO are needed. The process of production follows the following equations:

\[
\text{CH}_4 + H_2O + \text{heat} \rightarrow \text{CO} + 3H_2 \quad (1)
\]

\[
\text{CO} + H_2O \rightarrow \text{CO}_2 + H_2 + \text{heat} \quad (2)
\]
The hydrogen-fueled internal combustion engines for marine applications with case study

Ibrahim Seddiek, Mohamed Elgohary, Nader Ammar

2.2 Hydrogen from renewable sources

Hydrogen could be also produced by other methods than reforming of fossil fuels, including biomass, pyrolysis, aqueous phase reforming, water electrolysis, photoelectrolysis, and thermochemical water splitting [20].

2.2.1 Hydrogen from solar energy

Solar energy can be used as a source of energy to achieve hydrogen production through water electrolysis, photoelectrolysis, and thermochemical water splitting process [21]. Choice of solar energy for this purpose is the matter of environmental effect [22]. Among the previously mentioned technologies, the solar thermo chemical process is better from the viewpoint of production rate and environmental effect. For example, in traditional thermochemical technology, fossil fuels are combusted with air, which emits not only green house gases but also gases that contribute on the ozone layer depletion and acid rains. On the other hand, the solar thermo chemical technology offers either zero or low hazardous gas emissions [23].

As the main purpose of the present research is searching for environmental friendly marine fuel, the emphasis of the paper is on the production of hydrogen using solar energy as renewable source. Figure 1 presents a schematic diagram of solar thermo chemical production of hydrogen using fossil fuels and water (H₂O) as chemical source, including: solar cracking, solar reforming and gasification.

![Figure 1 Clean hydrogen fuel production](image-url)
Hydrogen is an extremely difficult gas to store, this will limit its use until convenient and cost effective storage technologies can be developed and commercialized. One gram of hydrogen gas, for instance, occupies about 12 litres of space at atmospheric pressure. In order to be more convenient, \( \text{GH}_2 \) must be pressurized under a high atmospheric pressure, and stored in a pressure vessel. In liquid form, hydrogen can only be stored under cryogenic temperatures.

Only the two major problems pending for solutions may make the full application of hydrogen fuel not achievable in the near future: hydrogen storage and production cost. Liquefied hydrogen has a density of 70.1 kg/m\(^3\), which is a very small value if compared to ordinary liquid fuels with densities in the range from 840 to 1010 kg/m\(^3\) [24-25], taking into account that liquid hydrogen heating value is about 3.3 times higher than that for diesel fuel. The production cost cannot be accurately determined since the hydrogen fuel is not produced on a mass production basis. Also, for the two major processes of hydrogen extracting, the water electrolysis and the steam reformation of natural gas; the production process will be more expensive than the ordinary fossil fuels. The cost of removing carbon dioxide (\( \text{CO}_2 \)), resulting from the natural gas steam reformation, increases the cost of the ‘fossil’ hydrogen option. Moreover, cost of hydrogen production by electrolysis is about three times higher than that produced by steam reforming of natural gas.

3.1 Options of hydrogen storage for marine use

Hydrogen storage is considered to be one of the main obstacles against adopting hydrogen as fuel onboard ships due to its very low energy and due to safety issues [26]. In this section, storage alternatives, which include compressed gas, liquefied gas and metal hydrides, are discussed in order to decide which of them will be suitable for marine use. Also, the transportation is discussed with special reference to the liquefied hydrogen (\( \text{LH}_2 \)) carriers under development.

3.1.1 Compressed hydrogen storage

Compressed hydrogen in hydrogen tanks under pressure of 350 bar to 700 bar is used for hydrogen tank systems in vehicles [27]. Storing of hydrogen in form of compressed gas is the simplest storage method. It needs a few devices such as a compressor and a pressure tank. On the other hand, the drawback of this method is low storage density, which depends on the storage pressure.

As the storage pressure increase, capital and operating costs will increase. It is important to know that when compared with traditional fuels, the energy in a compressed hydrogen tank is very low for the same tank volume density. Regarding the economics of this type of storage, both capital and operating costs must be well studied.
3.1.2 Liquefied hydrogen storage

Liquefied hydrogen storage refers to the storage of hydrogen in the liquid state by cooling of hydrogen vapour to the cryogenic temperatures of −253 °C. In addition, it can be stored as a constituent in other liquids, such as NaBH₄ solutions, rechargeable organic liquids, or anhydrous ammonia NH₃ [28]. By this method, the weight of hydrogen can be increased by approximately 20 times compared to the compressed form for the same volume [29].

3.1.3 Metal hydride hydrogen storage

Storage of hydrogen in metal hydrides can be achieved through bonding the hydrogen to the surface of metal. Metal hydride hydrogen storage has the following advantages: high hydrogen energy density, volumetric capacity, low pressures and low temperatures [30]. The safety of this method is exciting since no leakage is possible if the tank is broken or fractured due to accidents and hydrogen is not released unless heat is provided to break the bonds with the metal. Nevertheless, hydrogen absorption using metal hydrides, chemical hydrides and carbon systems, requires further development and evaluation [31].

There are some factors which play a role when deciding which method of storage might be adopted, including: the required energy density, the amount to be stored, the desired storage period, and the acceptable cost limit. By analyzing the three main systems, the following conclusions can be made: liquid storage – large hydrogen quantities can be stored, long-term storage if permanent cooling is applied, and low electricity costs for liquefaction; compressed storage – small storage quantities, and short storage time; a hydrogen gas tank that contains a store of energy equivalent to a traditional fuel tank would be more than 3,000 times bigger. Of course, this value varies with the pressure, but as already mentioned, higher pressure means higher cost [32].

For marine use, where large quantities of fuel in addition to long storage time are needed, the storage of hydrogen in the liquid state may appear as the best storage form. Although liquid hydrogen can provide a lot of advantages, its uses may be restricted because liquefying hydrogen by existing conventional methods consumes a large amount of heat. Practically, compressed hydrogen storage has short filling time and long storage time, while a liquid hydrogen tank has short filling time and short storage time. Moreover, during the long period of storage, to minimize and control the boil-off losses, the permanent cooling using another medium such as liquid nitrogen is needed, which requires additional costs [33].

4. Hydrogen bunkering process & regulations

The best way of the hydrogen fuel bunkering process will be in the form of liquefied gas. Liquefied hydrogen (LH₂) will be stored in cryogenic tanks at the temperature of 20 K and the amount of bunker fuel will depend mainly on the ship’s sailing time and engines specific fuel consumption. Figure 2 shows the principle components of hydrogen fuel bunkering either onshore or onboard ships. Some other considerations should be taken into account in the fuel amount estimation, such as the expected amount of fuel that has to evaporate to the gas form during consumption of LH₂ to maintain the tank temperature [29].
Bureau Veritas (BV) Classification Society has developed a comprehensive set of guidelines for the use of hydrogen fuel onboard commercial ships. The guidelines combine existing regulations for gas fuelled ships with regulations for terrestrial fuel cell power systems adapted for the application onboard ships. The guidelines are now being tested on a number of pilot projects, of which the Hydrogen-Powered Hybrid Electric Harbour Tug is a good example [34]. Bureau Veritas is looking forward to work together with partners within the industry to further develop the use of clean technologies in shipping [35-36].

The last decade has produced significant advancements in the development of the hydrogen-fuelled internal combustion engine. The beginning was the use of hydrogen as fuel for spark ignition engines. However, some problems were encountered related to the issues such as pre-ignition, knock, NOx control and loss in power density. Therefore, much effort has been put forth in the development of advanced hydrogen engines with improved power densities. The hydrogen fuelled internal combustion engine (H2ICE) technology passed through several stages as follows [37-38]:

i. Pressure-boosted H2ICE: problems of pre-ignition, knock and NOx control are heightened during boosted operation because boosting pressure increases charge pressure and temperature.

ii. Liquid-H2ICE was the second phase of H2ICE development where the primary benefit is the higher stored-energy density of hydrogen available with liquefaction. Moreover, the charge-cooling effect of the cold hydrogen provides for several advantages compared to conventional gaseous port fuel injection. However, practical difficulties of liquid storage include the energy penalty of liquefaction, evaporation during long-term storage, and the cost of onboard cryogenic dewars.

iii. Direct-injection hydrogen-fuelled internal combustion engine (DI-H2ICE): the direct injection H2ICE has long been viewed as one of the most attractive advanced H2ICE options. Preferential of direct-injection hydrogen is based on: the high volumetric efficiency, and the
potential to avoid pre-ignition. The challenge with DI-H_{2}ICE operation is that in-cylinder injection requires hydrogen-air mixing in a very short time.

iv. H_{2}ICE-electric hybrid: A hybrid-electric version of an H_{2}ICE offers the potential for improved efficiencies and reduced emissions without the need for aftertreatment. In a hybrid electric system, the ICE operates either in series or parallel with an electric motor.

6. Hydrogen marine power plants

Hydrogen is suggested to be used for the existing diesel engines to minimize the cost as much as possible. The suggested engine will be operated with hydrogen being directly injected into the cylinders, as shown in Figure 3. To initiate the combustion process, low energy sparks will be needed to avoid using amounts of diesel fuel. Fuel pumps and sparks are to be electronically controlled (camless engines) to ensure the optimum performance at various operating conditions, as shown in Figure 4.

One of the main problems related to the adoption of hydrogen for internal combustion engines is the engine knocking that arises due to malfunction of air fuel ratio and intake temperature.

![Figure 3 Schematic engine control systems](image)

![Figure 4 Hydrogen fuel system](image)
Different propulsion arrangements can be used to propel the ship. One of them is to use the hydrogen internal combustion engine connected to the propeller via gearbox, and another one is a modern arrangement – generating electricity by alternators to drive electric motors coupled to the propellers. Each arrangement has its advantages and disadvantages related to the field of usage. Figures 5 and 6 show these arrangements. Smaller hydrogen engines will be needed as shown in Figure 6 for operation at part load in harbours and for manoeuvring purposes, and also for use with auxiliary engines (A/E).

Generally, waste heat in exhaust gases, which is mainly high temperature, is very important because it eliminates the need for exhaust gas boiler, steam may be directly used onboard ship for heating or for any other process requiring high temperature [39].

Another use of hydrogen is for fuel cells. Huge developments have been achieved in this sector over the past few years. However, in the marine field it has been used only in the naval vessels market for auxiliary power generation and for quiet operation of submarines. Concerning the commercial market, the development achieved so far is not enough to convince ship owners to use this fine technology product. From all types of fuel cells, only two types are the candidates for the use onboard. The proton exchange membrane fuel cell (PEMFC) and the molten carbonate fuel cell (MCFC) fuelled by hydrogen rich fuels like natural gas or alcohols [40].

A number of ship design firms introduced designs for many ship types working with fuel cells as an auxiliary power source or for propulsion in hybrid modes [41]. In the marine sector, several research programs focused on the use of liquid hydrogen onboard ships in combustion engines or fuel cells. Although the use of hydrogen for marine applications provides a lot of advantages, especially regarding environmental issues, it is still restricted due to safety and storage problems.

### 7. Case study

The case study refers to repowering operation for a RO/RO ship, named *Taba*, operating in the Mediterranean Sea. The ship [42] is originally fitted with 2 MaK 9M453 engines with 2700 kW brake power each. Two alternatives were under consideration, either using pre-mixing of hydrogen and air before entering the combustion chamber or direct injection of hydrogen into the cylinders after compression. The analysis was done for both 2
The hydrogen-fueled internal combustion engines for marine applications with case study

and 4 stroke cycles based on the Otto standard cycle. The highest useful compression ratio (HUCR) for the hydrogen-air mixture is limited to 6 to avoid explosions [43]. The compression ratio is also limited between 2 and 6; compression ration less than 2 produces a very high combustion temperature and the values higher than 6 produce unstable combustion. Figure 7 shows the comparison between diesel and hydrogen fuels and the variation of combustion temperature at different excess air factor.

![Figure 7 Hydrogen and diesel combustion temperature versus excess air factor](image)

There are some points to be clarified concerning the comparison in Figure 7:

i. The efficiency of the hydrogen engine is lower than that of the diesel engine due to higher cooling water losses in the case of hydrogen. Without all these losses the temperature of combustion inside the cylinders may reach very high levels that may put the engine in a critical state. The obtained result is close to the results published in [44] which demonstrated the ability of achieving efficiency of about 32%.

ii. Due to the higher energy contained in hydrogen, less fuel is used by mass to produce the same power, but the product of fuel consumption and calorific value in both cases will yield a lower value in the case of hydrogen and this is due to lower efficiency.

iii. Very low brake mean effective pressure is available in case of hydrogen if compared with diesel despite the higher turbo charging pressure in the hydrogen engine (in diesel engine only 3.3 bar turbo charging pressure is used), this is due to the different nature of gaseous fuel with a very low density.

iv. Due to lower efficiency and lower density of hydrogen fuel, bigger engine dimensions are needed to produce the same power at the same speed of the diesel engine. It can be concluded that the hydrogen engine is still in need of a considerable effort in order to reach the competition phase. However, attempts to use the hydrogen fuelled engine instead of the diesel fuelled engine must be continued to overcome those obstacles.

*Condition (1): Pre-mixing*

Temperature = 127°C (at start of compression inside the cylinder) [45]
Pressure = 1 bar (atmospheric),
Compression ratio = 6
Figure 8 shows that at the excess air factor for the bore of 4-stroke exceeds the bore of 2-stroke engines by percentage of 25%. The 2 stroke cycle gives fewer diameters; however, this is larger by 6cm than the installed engine bore of 32 cm. Figure 9 explains the relation between brake specific fuel consumption at different excess air factor.

**Figure 8** Cylinder bore as calculated for condition (1)

**Figure 9** Thermal efficiency and fuel consumption for condition (1)

*Condition (2): Direct injection*

Temperature = 127°C

Pressure can be varied since no restriction on compression ratio. Figures 10 and 11 illustrate the variation of engine bore at different compression ratio for different pressures for the 4-stroke and 2-stroke engine respectively.
The hydrogen-fueled internal combustion engines for marine applications with case study

Ibrahim Seddiek, Mohamed Elgohary, Nader Ammar

Figure 10 Bore diameter for Condition 2 (4 strokes)

Figure 11 Bore diameter for condition 2 (2 strokes)

The required diameter can be achieved with a reasonable supercharging pressure. Thus, direct injection with 2-stroke cycle is the best choice for the engine design. From Figure 12 is evident that for obtaining the required diameter, a pressure ratio of 9 with supercharging pressure of 3.7 bar is adopted.

Figure 12 Thermal efficiency and fuel consumption for direct injection condition
Ibrahim Seddiek, Mohamed Elgohary, Nader Ammar
The hydrogen-fuelled internal combustion engine for marine applications with a case study

The comparison between the engine characteristics of the hydrogen engine and the ship’s original engine (M32C Diesel Engine) can be summarized as shown in Table 1.

**Table 1** Comparison between engine characteristics of hydrogen engine and M32C diesel engine

<table>
<thead>
<tr>
<th></th>
<th>Hydrogen</th>
<th>M32C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed (r.p.m)</td>
<td>775</td>
<td>600</td>
</tr>
<tr>
<td>Pressure ratio</td>
<td>9</td>
<td>12.8</td>
</tr>
<tr>
<td>Heating value of fuel (MJ/kg)</td>
<td>130</td>
<td>42.7</td>
</tr>
<tr>
<td>Bore (cm)</td>
<td>32</td>
<td>32</td>
</tr>
<tr>
<td>Stroke (cm)</td>
<td>42</td>
<td>42</td>
</tr>
<tr>
<td>Engine power (kW)</td>
<td>2700</td>
<td>2700</td>
</tr>
<tr>
<td>Thermal efficiency (%)</td>
<td>29.57</td>
<td>47</td>
</tr>
<tr>
<td>Specific fuel consumption (g/kW.hr)</td>
<td>93.46</td>
<td>178.98</td>
</tr>
<tr>
<td>No. of cylinders</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>Mean effective pressure</td>
<td>10.02</td>
<td>25.9</td>
</tr>
<tr>
<td>Compression pressure</td>
<td>75.57</td>
<td>84</td>
</tr>
<tr>
<td>Combustion pressure</td>
<td>132.4</td>
<td>124</td>
</tr>
</tbody>
</table>

The outcome of the preliminary design calculations reveals that the bore of the engine piston will be the same for the two engines, but the thermal efficiency of the hydrogen fuel cycle shows a decline compared to that of the original diesel fuel cycle. On the other hand, the specific fuel consumption is reduced to 93.46 g/kW.hr in case of hydrogen fuel.

The engine flow rates can be summarized as:

- **Fuel**: 93.46 g/kW.hr (252 kg/hr)
- **Cooling water**: 67.83 kg/kW.hr (183007 kg/hr)
- **Air**: 13 kg/kW.hr (35087 kg/hr)
- **Exhaust**: 13.1 kg/kW.hr (35339 kg/hr)

Regarding the fuel storage tanks onboard the ship; the volume of the ship fuel tanks (DMA and DMB) is 706.8 m³. When these tanks are used for LH₂, only 90% of this volume can be filled due to insulations. Thus, volume will be 636.1 m³. The volume of LH₂ required for 8 days’ voyage (2~3 stops in ports) is 1492 m³. This means that the present volume, for complete hydrogen power plant, i.e. both the main engines and the generators run on hydrogen, will be sufficient only for a voyage of 3.4 days. Consequently, the extra volume of about 950 m³ will be needed. This volume can be deducted from the twine deck area as shown in Figure 13. As can be seen, two times larger tank space in the ship under consideration is needed to accommodate the hydrogen fuel for only 8 days rather than 20 days as the ship originally travels.

This problem will occur only in cases when considering the conversion of existing ships to run on hydrogen, but new designs should include their own hydrogen tank spaces. Another solution may be available if new storage techniques are invented to overcome this problem. New techniques are required to provide fuel storage with appropriate energy density.
The hydrogen-fueled internal combustion engines for marine applications with case study

Ibrahim Seddiek, Mohamed Elgohary, Nader Ammar

Figure 13 Modified hydrogen fuel tanks

8. Conclusion

Several issues must be taken into consideration when trying to adopt a new type of fuel like hydrogen, especially for marine applications where strict rules and regulations control the design and manufacture of waterborne vehicles. Safety and storage problems are the main issues arising when talking about the use of hydrogen as fuel. Combustion of hydrogen inside internal combustion engines has been, and still is, the subject of numerous research programmes in many countries. Like in the case of natural gas, one of the main problems associated with the application of hydrogen in internal combustion engines is engine knocking; air fuel ratio and intake temperature were found to be the main causes for this problem and their optimization is a must in order to have a knock free engine.

The present paper discussed the different stages of hydrogen gas cycle beginning with the production process, from clean energy, until using it as fuel for internal combustion engines onboard a RO/RO ships, named Taba, operating in the Mediterranean Sea. Compared to the diesel engine, the hydrogen engine was found to be lower in thermal efficiency, mean effective pressure and fuel consumption, while the both engines seemed to have the same value of compression and combustion pressure. In addition, some adjustments are needed regarding the engine’s dimensions, valves timing and fuel system.

However, regarding the ship’s operation, some problems need to be solved. First of all, it is necessary to make some modifications for the use of hydrogen onboard ships, like for instance, the fuel storage capacity, as the fuel tanks of the ship under research are not capable to accommodate the hydrogen needed. The study has shown the need to increase the volume of bunkering tanks from 706.8 m$^3$ (heavy and diesel fuel) to 1492 m$^3$ (hydrogen fuel) so that the vessel can provide the necessary space for the hydrogen storage. The design results may seem strange to the professional reader and yield to the heavy, big and expensive engine, but it must not be forgotten that this is a first step. Other refinement procedures will follow and also prototype experiments have to be made to assess how far the calculations are from the real world and after that a fine tuning processes will follow. This procedure can also be applied to different types of power plants.
Ibrahim Seddiek, Mohamed Elgohary, Nader Ammar

The hydrogen-fuelled internal combustion engine for marine applications with a case study

Acknowledgment

This work was funded by the Deanship of Scientific Research (DSR), King Abdulaziz University, Jeddah, under grant no. (980-580-D1435). The authors gratefully acknowledge the DSR for their technical and financial support.

Nomenclature

<table>
<thead>
<tr>
<th>A/E</th>
<th>Auxiliary engines</th>
<th>HUCR</th>
<th>Higher Useful Compression</th>
</tr>
</thead>
<tbody>
<tr>
<td>B.V</td>
<td>Bureau VERITAS</td>
<td>IMO</td>
<td>International Maritime</td>
</tr>
<tr>
<td>BTE</td>
<td>Brake Thermal Efficiency</td>
<td>LH₂</td>
<td>Liquefied hydrogen</td>
</tr>
<tr>
<td>CO</td>
<td>Carbon Monoxide</td>
<td>MARPOL</td>
<td>Marine Pollution</td>
</tr>
<tr>
<td>EES</td>
<td>Engineering Equation Solver</td>
<td>NOx</td>
<td>Nitrogen-Oxides</td>
</tr>
<tr>
<td>GH₂</td>
<td>Gas Hydrogen</td>
<td>P.M</td>
<td>Particulates Mater</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbons</td>
<td>SOx</td>
<td>Sulfur- Oxides</td>
</tr>
<tr>
<td>HICE</td>
<td>Hydrogen</td>
<td>Internal</td>
<td></td>
</tr>
</tbody>
</table>

References

The hydrogen-fueled internal combustion engines for marine applications with case study

Ibrahim Seddiek, Mohamed Elgoehary, Nader Ammar


The hydrogen-fuelled internal combustion engine for marine applications with a case study


RISK EVALUATION OF PIN JIG WORK UNIT IN SHIPBUILDING BY USING FUZZY AHP METHOD

UDC 629.5.081:629.5.081.4
Professional paper

Summary

Shipbuilding industry includes many different industry branches in itself so various kind of work accidents occur. These work accidents often cause serious injuries and also deaths. It is a crucial thing to prevent or minimize these accidents. In order to reduce work accidents in shipyards, the most hazardous activities are needed to be determined and then, shipyard management must work on it in order to remove these hazard sources. In this study, pin jig work unit, where the curved parts are mounted on adjustable pin jigs, was considered. At first, the work activities and operations of pin jig work station were identified and they were classified as main and sub risk criterions. Then, pair comparison scales were built and these risk criterions were evaluated by experts who have been working for a shipyard located in Turkey. As a result of the evaluations of the experts, the risk weights of the activities carried out at pin jig work unit were defined by using fuzzy AHP method. Therefore, it is aimed for the shipyard management to take some precautions at pinjig work unit on the risky operations before failures happen.

Key words: Shipyard, risk criterions, risk evaluation, pin jigs, fuzzy AHP

1. Introduction

Shipbuilding is a heavy industry that the vessel production is performed and it includes many different work branches. The fact that it includes various industry fields and has different sort of work activities increase the accidents occurring in shipyards. The quantity and severity of the accidents have been increasing for years in Turkish shipyards and many serious injuries and also deaths have taken place. It is a very important thing to remove or minimize the failures in many ways.

In literature, there are many works regarding fuzzy AHP and risk evaluation. Zeng et al [1] used a modified Analytic Hierarchy Process (AHP) in order to determine the risks on steel erection in a shopping centre construction. Morate and Vila [2] utilized a fuzzy AHP to determine the risk weights on the rehabilitation project of a building at the University of Cartagena. Chan and Kumar [3] determined the risk weights in selecting global supplier by

In shipbuilding, there are also various studies concerning shipyard risk evaluation. Barlas [9] investigated the fatal occupational accidents in Turkey shipyards and classified them according to fatality reason, age etc. and presented some results based on statistical data. Barlas [10] used AHP in order to find the most suitable precautions for prevention from accidents occurred in Turkish shipyards and made some suggestions to reduce fatality reason. Shinoda et al [11] analyzed occupational accidents in Japanese shipyards and classified the failures to accident types, occurrence date, occurrence site etc., so presented the shipyard accidents in detail. Celebi et al [12] investigated all wastes and pollutants on worker health resulting from shipyard activities and also analyzed accidental injuries in Turkish shipyard.

In this study, the work unit called as pin jig was taken into consideration and a risk evaluation based on fuzzy AHP (Analytic Hierarchy Process) was carried out. For this, firstly, main risk criterions, which may be source of potential risks, were identified and then sub risk criterions were defined for each main risk criterion. After main and sub risk criterions were determined, pair wise comparison scales were built up between main and sub risk criterions. These scales were submitted to the experts in order to take their evaluations and the risk weights of the risk criterions were calculated by using Buckley’s method.

Furthermore, number of three experts, who are naval architecture and marine engineers, evaluated the performance criterions. They work at the department of quality control of a shipyard located in Tuzla Region in Istanbul/Turkey.

2. Pin Jig Work Unit

It is almost impossible to perform the mounting and welding activities of curves panel and stiffeners on a flat surface. Therefore, pin jigs are needed to complete the necessary operations of curved parts and sections. At pin jig work unit which is situated in shipyard production system, there are adjustable pins which are used in fixing the curved panel and stiffener in order to facilitate the mounting, welding and grinding operations. Each curved block, which form the vessel structure, is placed on heightened pin jigs and it is not moved until its assembly work is finished once it has been positioned [13]. Figure 1 shows the general arrangement of pin jig work unit in SIMIO simulation environment.
Figure 2 depicts the curved panel on adjustable pins. At the beginning, the first curved plate is positioned on pin jigs and the second plate is fixed near the first one. Then, they are assembled together by welding. If necessary, the other plates are welded together in the same way. Finally, a curved panel of a block is fabricated.

Figure 3 demonstrates the curved panel with stiffeners. Upon the curved panel, the stiffeners are lined and fixed by fillet weld and the curved panel with stiffeners is manufactured in this way.
Pin jig work unit is fed by the parts coming from nesting, pre-production, frame bending, and plate bending work units. Curved stiffeners, minor assembly and sub assembly structures are mounted at this work unit and finally they constitute a curved block of the vessel. A general work flow at pin jig is illustrated in Figure 4.

Fig. 4 Work flow of pin jig work unit

3. Materials and Methods

The evaluations of the experts or individuals are able to be easily expressed with fuzzy logic. If a person is needed to make a decision, he could express his evaluations using linguistic statements instead of assigning any crisp score to the evaluations and fuzzy AHP is presented for the purpose of resolving lack of manifesting human perception and thought of AHP developed by Thomas Saaty [14]. In this work, Buckley’s technique was utilized.

In the first step of the study, the definition of the performance parameters is carried out and main and sub criterions are determined. In the second step, identification of the linguistic terms including fuzzy numbers are performed. Then, the comparison scales are formed and submitted to the experts in order for them to evaluate and collected finally (Step 3). In the next step (Step 4), the linguistic expressions are transformed into fuzzy numbers. In Step 5, the evaluations of the experts are aggregated and aggregated pair wise matrix is created. Then, the criteria weights are calculated by utilizing Buckley’s fuzzy AHP in order to determine the effects of the parameters on decision making (Step 6). In Step 7, the normalization of the fuzzy numbers are carried out in order to find the crisp values and in the last step of the study (Step 8), relative criteria weights are calculated so as to separately determine the effects of each criteria.

3.1 Definition of performance parameters (Step 1)

In this step, the performance parameters or criterions are determined. The performance criterions are divided into two parts which are called “main criterion” and “sub criterion”. At first, main criterions are identified. Then, for each main criterion, sub criterions are defined.

3.2 Identification of the linguistic terms (Step 2)

In this step, linguistic scale and fuzzy numbers utilized in this study are identified by benefiting from literature.
3.3 Collecting of expert preferences (Step 3)

Expert opinions or preferences concerning performance criterions are collected by utilizing a questionnaire. Experts evaluate performance criterions by using a pairwise comparison scale. Fuzzy decision matrix is demonstrated as below:

\[
\tilde{C}^k = \begin{bmatrix}
1 & \tilde{c}_{12} & \ldots & \tilde{c}_{1n} \\
\tilde{c}_{21} & 1 & \ldots & \tilde{c}_{2n} \\
\vdots & \vdots & \ddots & \vdots \\
\tilde{c}_{m1} & \tilde{c}_{m2} & \ldots & 1
\end{bmatrix}
\]

(1)

where \(\tilde{C}^k\) is fuzzy decision matrix given by \(k^{th}\) expert for importance degrees of criteria.

In Equation 2, there are some abbreviations due to a lack of space on page. Here, “row demon. im.” means row is demonstrated important in comparison with column. Furthermore, “row very str. im.” implies that row has very strong importance according to column while “row str. im.” implies row has strong importance. Moreover, “row mode. im.” means that row has moderate importance while “row-column eq.” is meaning that row and column have equal importance degree. The same definitions are valid for column abbreviations in Equation 2.

3.4 Data transformation into triangular fuzzy numbers (TFN) (Step 4)

Linguistic statements coming from experts must be expressed into triangular fuzzy numbers (TFN) because linguistic statements are not mathematically operable. A TFN is represented by a membership function and \(\mu_{\bar{a}}(x)\), in the range \([0, 1]\) defines the membership degree of the fuzzy number to a fuzzy set [15]. A triangular fuzzy number is shown as below:

\[
\mu_{\bar{a}}(x) = \begin{cases}
\text{if } n_1 \leq x \leq n_2, & (x - n_1)/(n_2 - n_1) \\
\text{if } n_2 \leq x \leq n_3, & (n_3 - x)/(n_3 - n_2) \\
\text{if } x > n_3 \text{ or } x < n_1, & 0
\end{cases}
\]

(3)

where \(\mu_{\bar{a}}(x)\) is membership function; \(n_1\) is lower boundary; \(n_3\) is upper boundary; \(n_2\) is mean value. Figure 5 depicts a triangular fuzzy number.
3.5 Collection of the experts’ evaluations (Step 5)

At this stage, the evaluations of the experts are aggregated. The weighted average method is utilized in order to aggregate the preferences of the experts. Aggregated pair wise matrix is defined as below:

\[
\tilde{C} = \begin{bmatrix}
1 & \tilde{c}_{12} & \ldots & \tilde{c}_{1n} \\
\tilde{c}_{21} & 1 & \ldots & \tilde{c}_{2n} \\
\vdots & \vdots & \ddots & \vdots \\
\tilde{c}_{m1} & \tilde{c}_{m2} & \ldots & 1
\end{bmatrix}
\]

where \( \tilde{C} \) is aggregated pair wise comparison matrix in accordance with importance degrees of criteria.

3.6 Calculation of criteria weights (Step 6)

In this study, Buckley’s fuzzy AHP is used to determine the fuzzy weights. After aggregated pair wise matrix (\( \tilde{C} \)) is achieved, the fuzzy weight matrix is determined by Buckley’s Method as below:

\[
\tilde{r}_i = (\tilde{c}_{i1} \otimes \tilde{c}_{i2} \otimes \ldots \otimes \tilde{c}_{in})^{1/n}
\]

\[
\tilde{w}_i = \tilde{r}_i \otimes (\tilde{r}_1 + \tilde{r}_2 + \ldots + \tilde{r}_n)^{-1}
\]

where \( \tilde{c}_{in} \) is the fuzzy comparison value of criterion \( i \) to criterion \( n \), \( \tilde{r}_i \) is the geometric mean of fuzzy comparison value of criterion \( i \) to each criterion.

3.7 Defuzzification and normalization process for fuzzy weights (Step 7)

In order to transform the fuzzy weights into crisp values, median method is implemented:

\[
w_i = \frac{n_1 + n_2 + n_3}{3}
\]

where \( n_1 \) is lower boundary; \( n_3 \) is upper boundary; \( n_2 \) is mean value of fuzzy weight. Crisp values are normalized to have more comprehensible results by using Eq. 8 [16].
\[
(w_N)_i^c = \frac{w_i^c}{\sum_{i=1}^{n} w_i^c}
\]

(8)

where \((w_N)_i^c\) is normalized weight of \(i^{th}\) main criterion, \(n\) is number of main criteria; for sub criteria the Eq. 9 is used:

\[
(w_N)_i^sc = \frac{w_i^sc}{\sum_{i=1}^{n} w_i^sc}
\]

(9)

where \((w_N)_i^sc\) is normalized weight of \(i^{th}\) sub criterion, \(n\) is number of sub criteria.

3.8 Calculation of relative criteria weights (Step 8)

In order to evaluate sub criteria between themselves, relative fuzzy weights and relative crisp weights are calculated by utilizing Eq. 10 and Eq. 11,

\[
(\tilde{w}_R)_i^c = (\tilde{w})^c \otimes (\tilde{w})^c
\]

(10)

where \((\tilde{w}_R)_i^c\) is relative fuzzy weight of \(i^{th}\) sub criterion, \((\tilde{w})^c\) is fuzzy weight of main criterion which includes that sub criterion, \((\tilde{w})_i^c\) is fuzzy weight of \(i^{th}\) sub criterion.

\[
(w_R)_i^c = (w_N)_i^c \times (w_N)_i^c
\]

(11)

where \((w_R)_i^c\) is relative crisp weight of \(i^{th}\) sub criterion, \((w_N)_i^c\) is normalized crisp weight of main criterion which includes that sub criterion, \((w_N)_i^c\) is normalized crisp weight of \(i^{th}\) sub criterion.

4. Results and Discussions

In this study, three experts who work in Turkish shipyards evaluated the performance criterions and the assessments of experts were collected and considered in determining the weights of the criterions.

4.1 Determination of criterions (Step 1)

In this section, the risk criterions in pin jig workshop were determined. Four main criterions specified as “crane movements, welding, grinding, and mounting” were defined. Under these main risk criterions, sub risk criterions were defined. Figure 6 and Table 1 show the main and sub risk criterions.
Fig. 6 Main and sub risk criterions used in the study

Table 1 Definitions of sub risk criterions

<table>
<thead>
<tr>
<th>Risk criteria</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Holding piece parts (C1)</td>
<td>How much risk is there when the crane holds the surface of the piece parts in order to lift it up?</td>
</tr>
<tr>
<td>Lifting piece parts (C2)</td>
<td>How much risk is there when the crane lifts the piece parts?</td>
</tr>
<tr>
<td>Transporting piece parts (C3)</td>
<td>How much risk is there when the crane transports the piece parts to the places where they are needed?</td>
</tr>
<tr>
<td>Putting down piece parts (C4)</td>
<td>How much risk is there when the crane puts the piece parts on the ground?</td>
</tr>
<tr>
<td>Holding sub assembly unit (C5)</td>
<td>How much risk is there when the crane holds the surface of sub assembly unit in order to lift it up?</td>
</tr>
<tr>
<td>Lifting sub assembly unit (C6)</td>
<td>How much risk is there when the crane lifts sub assembly unit?</td>
</tr>
<tr>
<td>Transporting sub assembly unit (C7)</td>
<td>How much risk is there when the crane transports sub assembly unit to the places where they are needed?</td>
</tr>
<tr>
<td>Putting down sub assembly unit (C8)</td>
<td>How much risk is there when the crane puts sub assembly unit on the ground?</td>
</tr>
<tr>
<td>Tack welding preparation (W1)</td>
<td>How much risk is there when the worker is preparing the tack welding machine and torch before doing tack welding operation?</td>
</tr>
<tr>
<td>Tack welding (W2)</td>
<td>How much risk is there when the worker connects the parts with tack welding?</td>
</tr>
<tr>
<td>Gas metal arc welding preparation (W3)</td>
<td>How much risk is there when the operator is preparing the gas metal arc welding machine and torch before performing gas metal arc welding activity?</td>
</tr>
<tr>
<td>Gas metal arc welding (W4)</td>
<td>How much risk is there during assembling the parts with gas metal arc welding?</td>
</tr>
<tr>
<td>Submerged arc welding preparation (W5)</td>
<td>How much risk is there when the worker is preparing the submerged arc welding machine and torch before carrying out submerged arc welding operation?</td>
</tr>
<tr>
<td>Submerged arc welding (W6)</td>
<td>How much risk is there when the worker is preparing the submerged arc welding machine and torch before carrying out submerged arc welding operation?</td>
</tr>
<tr>
<td>Starting up grinding machine (G1)</td>
<td>How much risk is there while assembling the parts with submerged arc welding?</td>
</tr>
<tr>
<td>Grinding after tack welding (G2)</td>
<td>How much risk is there while performing grinding activity after tack welding operation?</td>
</tr>
<tr>
<td>Grinding after gas metal arc welding (G3)</td>
<td>How much risk is there while carrying out grinding activity after gas metal arc welding operation?</td>
</tr>
<tr>
<td>Grinding after submerged arc welding (G4)</td>
<td>How much risk is there while performing grinding activity after submerged arc welding operation?</td>
</tr>
<tr>
<td>Assembly of backing (M1)</td>
<td>How much risk is there during assembling the ceramic backing to the connection edges of the plates?</td>
</tr>
<tr>
<td>Alignment of parts (M2)</td>
<td>How much risk is there while the parts are aligned on the marking points?</td>
</tr>
<tr>
<td>Getting clearance between parts (M3)</td>
<td>How much risk is there while the gaps between the parts are removing?</td>
</tr>
</tbody>
</table>
4.2 Identification of the linguistic statements (Step 2)

Linguistic statements and their fuzzy number definitions performed in this study are demonstrated in Table 2 [17].

<table>
<thead>
<tr>
<th>Linguistic statements</th>
<th>Fuzzy number definitions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equal risky</td>
<td>(1,1,1)</td>
</tr>
<tr>
<td>Moderate risky</td>
<td>(1,3,5)</td>
</tr>
<tr>
<td>Strong risky</td>
<td>(3,5,7)</td>
</tr>
<tr>
<td>Very strong risky</td>
<td>(5,7,9)</td>
</tr>
<tr>
<td>Demonstrated risky</td>
<td>(7,9,11)</td>
</tr>
</tbody>
</table>

4.3 Collection of expert evaluations (Step 3)

In this step, experts rated the risk parameters according to their experience and the evaluations of them were collected. Here, only Expert 1 evaluation of main risk criterions was illustrated in Table 3.

<table>
<thead>
<tr>
<th>Crane movements (C)</th>
<th>Welding (W)</th>
<th>Grinding (G)</th>
<th>Mounting (M)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>-</td>
<td>Column very str. risky</td>
<td>Column demon. risky</td>
</tr>
<tr>
<td>W</td>
<td>Row very str. risky</td>
<td>-</td>
<td>Column very str. risky</td>
</tr>
<tr>
<td>G</td>
<td>Row demon. risky</td>
<td>Row very str. risky</td>
<td>-</td>
</tr>
<tr>
<td>M</td>
<td>Row demon. risky</td>
<td>Row mode. risky</td>
<td>Row-column eq. risky</td>
</tr>
</tbody>
</table>

4.4 Conversion of linguistic statements into triangular fuzzy numbers (TFN) (Step 4)

In this step, the expert evaluations, which include linguistic statements, were converted to triangular fuzzy numbers. In the same way, for only Expert 1’s evaluation, the fuzzy number transformation was demonstrated here. Table 4 shows the evaluation of Expert 1 with fuzzy numbers.

<table>
<thead>
<tr>
<th>Crane movements (C)</th>
<th>Welding (W)</th>
<th>Grinding (G)</th>
<th>Mounting (M)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C (1,000,1,000,1,000)</td>
<td>(0,111,0,143,0,200)</td>
<td>(0,091,0,111,0,143)</td>
<td>(0,091,0,111,0,143)</td>
</tr>
<tr>
<td>W (5,000,7,000,9,000)</td>
<td>(1,000,1,000,1,000)</td>
<td>(0,111,0,143,0,200)</td>
<td>(0,200,0,333,1,000)</td>
</tr>
<tr>
<td>G (7,000,9,000,11,000)</td>
<td>(5,000,7,000,9,000)</td>
<td>(1,000,1,000,1,000)</td>
<td>(1,000,1,000,1,000)</td>
</tr>
<tr>
<td>M (7,000,9,000,11,000)</td>
<td>(1,000,3,000,5,000)</td>
<td>(1,000,1,000,1,000)</td>
<td>(1,000,1,000,1,000)</td>
</tr>
</tbody>
</table>

4.5 Aggregation of the evaluations of the experts (Step 5)

As mentioned above, there are three experts who rate the risk criterions. In this section, the evaluations of the experts were aggregated. The aggregated fuzzy decision matrix was demonstrated in Table 5-10.
### Table 5  Aggregated values for main risk criterions

<table>
<thead>
<tr>
<th>Crane movements (C)</th>
<th>Welding (W)</th>
<th>Grinding (G)</th>
<th>Mounting (M)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>(1.000,1.000,1.000)</td>
<td>(3.370,4.714,6.066)</td>
<td>(2.697,4.037,5.381)</td>
</tr>
<tr>
<td>W</td>
<td>(1.745,2.437,3.158)</td>
<td>(1.000,1.000,1.000)</td>
<td>(1.104,1.825,2.733)</td>
</tr>
<tr>
<td>G</td>
<td>(1.745,2.437,3.158)</td>
<td>(2.047,3.400,4.777)</td>
<td>(1.000,1.000,1.000)</td>
</tr>
<tr>
<td>M</td>
<td>(2.400,3.085,3.781)</td>
<td>(0.467,1.222,2.333)</td>
<td>(0.733,0.778,1.000)</td>
</tr>
</tbody>
</table>

### Table 6  Aggregated values for risk criterions based on crane movements

<table>
<thead>
<tr>
<th>Holding piece parts (C1)</th>
<th>Lifting piece parts (C2)</th>
<th>Transporting piece parts (C3)</th>
<th>Putting down piece parts (C4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>(1.000,1.000,1.000)</td>
<td>(0.411,0.437,0.492)</td>
<td>(0.108,0.141,0.206)</td>
</tr>
<tr>
<td>C2</td>
<td>(3.637,5.000,6.333)</td>
<td>(1.000,1.000,1.000)</td>
<td>(0.145,0.215,0.492)</td>
</tr>
<tr>
<td>C3</td>
<td>(5.667,7.667,9.667)</td>
<td>(3.667,5.667,7.667)</td>
<td>(1.000,1.000,1.000)</td>
</tr>
<tr>
<td>C4</td>
<td>(4.333,6.333,8.333)</td>
<td>(1.400,2.111,3.000)</td>
<td>(0.704,0.714,0.733)</td>
</tr>
<tr>
<td>C5</td>
<td>(2.067,2.777,3.667)</td>
<td>(0.430,1.148,2.048)</td>
<td>(0.126,0.170,0.200)</td>
</tr>
<tr>
<td>C6</td>
<td>(1.400,2.777,4.333)</td>
<td>(1.381,2.067,2.778)</td>
<td>(0.430,0.481,0.714)</td>
</tr>
<tr>
<td>C7</td>
<td>(4.333,6.333,8.333)</td>
<td>(1.667,3.000,4.333)</td>
<td>(2.067,2.778,3.667)</td>
</tr>
<tr>
<td>C8</td>
<td>(3.000,5.000,7.000)</td>
<td>(2.704,4.048,5.400)</td>
<td>(0.704,0.714,0.733)</td>
</tr>
</tbody>
</table>

### Table 7  Aggregated values for risk criterions based on crane movements (continue)

<table>
<thead>
<tr>
<th>Holding sub assembly unit (C5)</th>
<th>Lifting sub assembly unit (C6)</th>
<th>Transporting sub assembly unit (C7)</th>
<th>Putting down sub assembly unit (C8)</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>(0.704,1.381,2.067)</td>
<td>(0.448,1.178,2.111)</td>
<td>(0.134,0.196,0.448)</td>
</tr>
<tr>
<td>C2</td>
<td>(2.733,4.111,5.667)</td>
<td>(1.381,2.067,2.778)</td>
<td>(0.448,0.511,0.778)</td>
</tr>
<tr>
<td>C3</td>
<td>(4.333,6.333,8.333)</td>
<td>(3.000,4.333,5.667)</td>
<td>(0.704,1.381,2.067)</td>
</tr>
<tr>
<td>C4</td>
<td>(1.000,3.000,5.000)</td>
<td>(0.437,0.492,0.733)</td>
<td>(0.704,1.381,2.067)</td>
</tr>
<tr>
<td>C5</td>
<td>(1.000,1.000,1.000)</td>
<td>(0.126,0.170,0.270)</td>
<td>(0.448,0.511,0.778)</td>
</tr>
<tr>
<td>C6</td>
<td>(4.333,6.333,8.333)</td>
<td>(1.000,1.000,1.000)</td>
<td>(2.067,3.444,5.000)</td>
</tr>
<tr>
<td>C7</td>
<td>(1.667,3.000,4.333)</td>
<td>(0.437,1.159,2.067)</td>
<td>(1.000,1.000,1.000)</td>
</tr>
<tr>
<td>C8</td>
<td>(2.333,4.333,6.333)</td>
<td>(0.437,1.159,2.067)</td>
<td>(0.697,1.370,2.048)</td>
</tr>
</tbody>
</table>

### Table 8  Aggregated values for risk criterions based on welding operation

<table>
<thead>
<tr>
<th>Tack welding preparation (W1)</th>
<th>Tack welding (W2)</th>
<th>Gas metal arc welding preparation (W3)</th>
<th>Gas metal arc welding (W4)</th>
<th>Submerged arc welding preparation (W5)</th>
<th>Submerged arc welding (W6)</th>
</tr>
</thead>
<tbody>
<tr>
<td>W1</td>
<td>(1.000,1.000,1.000)</td>
<td>(1.097,1.815,2.714)</td>
<td>(0.437,1.159,2.067)</td>
<td>(1.078,1.770,2.492)</td>
<td>(1.114,1.844,2.778)</td>
</tr>
<tr>
<td>W2</td>
<td>(2.714,4.067,5.444)</td>
<td>(1.000,1.000,1.000)</td>
<td>(2.067,3.444,5.000)</td>
<td>(0.448,1.782,2.111)</td>
<td>(1.667,3.667,5.667)</td>
</tr>
<tr>
<td>W3</td>
<td>(2.067,3.444,5.000)</td>
<td>(0.437,1.159,2.067)</td>
<td>(1.000,1.000,1.000)</td>
<td>(3.381,4.733,6.111)</td>
<td>(1.000,1.000,1.000)</td>
</tr>
<tr>
<td>W5</td>
<td>(1.381,2.733,4.111)</td>
<td>(0.181,0.289,0.778)</td>
<td>(1.000,1.000,1.000)</td>
<td>(0.411,1.104,1.825)</td>
<td>(0.411,1.104,1.825)</td>
</tr>
<tr>
<td>W6</td>
<td>(2.714,4.067,5.444)</td>
<td>(0.735,2.111,3.667)</td>
<td>(2.067,3.444,5.000)</td>
<td>(0.162,0.244,0.555)</td>
<td>(0.411,1.104,1.825)</td>
</tr>
</tbody>
</table>

48
4.6 Determination of criterion weights (Step 6), defuzzification and normalization procedure for fuzzy weights (Step 7) and calculation of relative criteria weights (Step 8)

In this stage, three steps of the methodology were completed. The significance degrees of risk parameters were calculated according to Buckley’s Fuzzy AHP and the results are shown in Table 11.

Table 9 Aggregated values for risk criterions based on grinding operation

<table>
<thead>
<tr>
<th></th>
<th>Starting up grinding machine (G1)</th>
<th>Grinding after tack welding (G2)</th>
<th>Grinding after gas metal arc welding (G3)</th>
<th>Grinding after submerged arc welding (G4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>G1</td>
<td>(1.000, 1.000, 1.000)</td>
<td>(2.437, 3.159, 4.067)</td>
<td>(2.437, 3.159, 4.067)</td>
<td>(2.437, 3.159, 4.067)</td>
</tr>
<tr>
<td>G2</td>
<td>(2.030, 3.370, 4.714)</td>
<td>(1.000, 1.000, 1.000)</td>
<td>(1.800, 2.555, 3.667)</td>
<td>(2.733, 4.111, 5.667)</td>
</tr>
<tr>
<td>G3</td>
<td>(2.030, 3.370, 4.714)</td>
<td>(0.704, 2.048, 3.400)</td>
<td>(1.000, 1.000, 1.000)</td>
<td>(1.667, 3.667, 5.667)</td>
</tr>
<tr>
<td>G4</td>
<td>(2.030, 3.370, 4.714)</td>
<td>(0.418, 1.114, 1.844)</td>
<td>(0.181, 0.289, 0.778)</td>
<td>(1.000, 1.000, 1.000)</td>
</tr>
</tbody>
</table>

Table 10 Aggregated values for risk criterions based on mounting operation

<table>
<thead>
<tr>
<th></th>
<th>Assembly of backing (M1)</th>
<th>Alignment of parts (M2)</th>
<th>Getting clearance between parts (M3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>M1</td>
<td>(1.000, 1.000, 1.000)</td>
<td>(0.151, 0.225, 0.511)</td>
<td>(0.151, 0.225, 0.511)</td>
</tr>
<tr>
<td>M2</td>
<td>(3.000, 5.000, 7.000)</td>
<td>(1.000, 1.000, 1.000)</td>
<td>(0.429, 1.133, 1.889)</td>
</tr>
<tr>
<td>M3</td>
<td>(3.000, 5.000, 7.000)</td>
<td>(2.067, 3.444, 5.000)</td>
<td>(1.000, 1.000, 1.000)</td>
</tr>
</tbody>
</table>

Table 11 The significance degrees of risk criterions

<table>
<thead>
<tr>
<th>Main and sub risk criterions</th>
<th>Overall fuzzy weights</th>
<th>Relative fuzzy weights</th>
<th>Crisp</th>
<th>Relative crisp</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crane movements (C)</td>
<td>(0.234, 0.383, 0.642)</td>
<td>(0.003, 0.011, 0.041)</td>
<td>0.378</td>
<td></td>
</tr>
<tr>
<td>Holding piece parts (C1)</td>
<td>(0.015, 0.029, 0.063)</td>
<td>(0.013, 0.039, 0.135)</td>
<td>0.106</td>
<td>0.040</td>
</tr>
<tr>
<td>Lifting piece parts (C2)</td>
<td>(0.054, 0.102, 0.211)</td>
<td>(0.012, 0.038, 0.120)</td>
<td>0.098</td>
<td>0.037</td>
</tr>
<tr>
<td>Transporting piece parts (C3)</td>
<td>(0.128, 0.239, 0.439)</td>
<td>(0.004, 0.014, 0.056)</td>
<td>0.041</td>
<td>0.015</td>
</tr>
<tr>
<td>Putting down piece parts (C4)</td>
<td>(0.053, 0.100, 0.186)</td>
<td>(0.020, 0.063, 0.209)</td>
<td>0.166</td>
<td>0.063</td>
</tr>
<tr>
<td>Holding sub assembly unit (C5)</td>
<td>(0.019, 0.036, 0.087)</td>
<td>(0.021, 0.070, 0.229)</td>
<td>0.182</td>
<td>0.069</td>
</tr>
<tr>
<td>Lifting sub assembly unit(C6)</td>
<td>(0.084, 0.165, 0.326)</td>
<td>(0.017, 0.056, 0.180)</td>
<td>0.144</td>
<td>0.054</td>
</tr>
<tr>
<td>Welding (W)</td>
<td>(0.103, 0.211, 0.396)</td>
<td>(0.275, 0.531, 0.854)</td>
<td>0.378</td>
<td></td>
</tr>
<tr>
<td>Tack welding preparation (W1)</td>
<td>(0.061, 0.142, 0.326)</td>
<td>(0.008, 0.040, 0.181)</td>
<td>0.194</td>
<td>0.041</td>
</tr>
<tr>
<td>Tack welding (W2)</td>
<td>(0.074, 0.191, 0.458)</td>
<td>(0.007, 0.033, 0.135)</td>
<td>0.152</td>
<td>0.032</td>
</tr>
<tr>
<td>Gas metal arc welding preparation (W3)</td>
<td>(0.069, 0.156, 0.342)</td>
<td>(0.013, 0.055, 0.223)</td>
<td>0.254</td>
<td>0.054</td>
</tr>
<tr>
<td>Gas metal arc welding (W4)</td>
<td>(0.123, 0.261, 0.564)</td>
<td>(0.006, 0.025, 0.110)</td>
<td>0.121</td>
<td>0.026</td>
</tr>
<tr>
<td>Submerged arc welding preparation (W5)</td>
<td>(0.056, 0.118, 0.277)</td>
<td>(0.008, 0.028, 0.130)</td>
<td>0.137</td>
<td>0.029</td>
</tr>
<tr>
<td>Submerged arc welding (W6)</td>
<td>(0.053, 0.132, 0.328)</td>
<td>(0.016, 0.069, 0.238)</td>
<td>0.275</td>
<td>0.068</td>
</tr>
<tr>
<td>Grinding (G)</td>
<td>(0.142, 0.248, 0.432)</td>
<td>(0.026, 0.073, 0.221)</td>
<td>0.287</td>
<td>0.071</td>
</tr>
<tr>
<td>Starting up grinding machine (G1)</td>
<td>(0.182, 0.294, 0.512)</td>
<td>(0.024, 0.075, 0.243)</td>
<td>0.300</td>
<td>0.074</td>
</tr>
<tr>
<td>Grinding after tack welding (G2)</td>
<td>(0.116, 0.278, 0.552)</td>
<td>(0.016, 0.069, 0.238)</td>
<td>0.275</td>
<td>0.068</td>
</tr>
<tr>
<td>Grinding after gas metal arc welding (G3)</td>
<td>(0.058, 0.126, 0.288)</td>
<td>(0.008, 0.031, 0.125)</td>
<td>0.138</td>
<td>0.034</td>
</tr>
<tr>
<td>Mounting (M)</td>
<td>(0.090, 0.158, 0.287)</td>
<td>(0.045, 0.078, 0.199)</td>
<td>0.093</td>
<td>0.015</td>
</tr>
<tr>
<td>Assembly of backing (M1)</td>
<td>(0.173, 0.376, 0.737)</td>
<td>(0.016, 0.059, 0.212)</td>
<td>0.371</td>
<td>0.060</td>
</tr>
<tr>
<td>Alignment of parts (M2)</td>
<td>(0.293, 0.545, 1.019)</td>
<td>(0.026, 0.086, 0.293)</td>
<td>0.536</td>
<td>0.086</td>
</tr>
</tbody>
</table>

49
The most risky criterion was found to be crane movement in comparison with the other main risk criterions such as welding, grinding and mounting. Besides, the grinding activity was the second one as risk level at pin jig work unit. Figure 7 depicts the risk weights of main risk criterions.

![Fig. 7 Risk weights of main risk criterions](image)

Furthermore, the most risky activity is to transport piece parts (C3) in crane movements. Transporting sub assembly unit (C7) is the second most risky activity. The least risky activity is to hold piece parts (C1). Figure 8 demonstrates the risk weights of sub risk criterions.

![Fig. 8 Risk weights of sub risk criterions of crane movement](image)

Figure 9 demonstrates the risk weights of sub risk criterions of welding operation. It was seen that the most risky activity was based on Gas Metal Arc Welding (W4). Tack welding is the second one.

![Fig. 9 Risk weights of sub risk criterions of welding operation](image)
Figure 10 shows the risk weights of sub risk criterions of grinding operation. It can be seen that the grinding activity after tack welding is the most risky activity.

![Fig. 10 Risk weights of sub risk criterions of grinding operation](image)

In Figure 11, risk weights based on mounting operation are shown. According to this, the activity of getting clearance between parts is the most risky mounting activity. The second most risky one is the activity of alignment of parts.

![Fig. 11 Risk weights of sub risk criterions of mounting operation](image)

In Figure 12, the whole sub risk criterions (or sub activities in other words) and their risk weights were demonstrated. Getting clearance between parts (M3) and transporting piece parts (C3) have a risk weight of approximately 9%. On the other hand, transporting sub assembly unit (C7), starting-up grinding machine (G1), grinding after tack welding (G2), and grinding after gas metal arc welding (G3) have a risk weight of around 7%. Holding piece parts (C1), holding sub assembly unit (C5), and assembly of backing (M1) have least weights, 2% and 1% respectively.
5. Conclusion

According to the evaluations of experts, the most risky activities in pin jig work unit are those transporting piece parts (C3) and getting clearance between parts (M3) because they have the highest risk weights. The other risky activities are transporting sub assembly unit (C7), starting-up grinding machine (G1), grinding after tack welding (G2), and grinding after gas metal arc welding (G3), lifting sub assembly unit (C6), alignment of parts (M2), respectively.

Therefore, shipyard management must investigate the processes of transportation of the parts at pin jig work unit and remove the hazardous risk sources or attempt to minimize them. Furthermore, fairing activity (or getting clearance between parts) was found to be the other most risky activity at pin jig work unit. In the same way, the shipyard management must examine the fairing activity in detailed and try to reduce the hazardous risk sources.

This kind of risk assessment presented in this study should perform for the other work units at shipyards. If this is done, the most hazardous activities for each work unit could be determined and the work accidents taking place at each work unit can be reduced. In this way, the rates of work loss and injuries can be minimized.

REFERENCES


Although considerable technical preventive measures have been taken in marine diesel engine and auxiliary systems, it is possible to observe unexpected faults in the course of the operating conditions. These faults can become so severe that they can cause losses which can be irreversible. This study aims to present Fuzzy Analytic Hierarchy Process (AHP) and VIKOR (Vise Kriterijumska Optimizacija I Kompromisno Resenje) methods applied for the expert failure detection of marine diesel engine and auxiliary systems. In this study, the failures of marine diesel engine have been revealed and prioritized. Accordingly, the section of the machine from which the failures primarily arise has been determined. At the same time, the importance of the effective use of time in determining and responding to the failures has been indicated. By means of the evaluation of decision-making groups, the system most severely affected by failures has been decided.

**Summary**

Although considerable technical preventive measures have been taken in marine diesel engine and auxiliary systems, it is possible to observe unexpected faults in the course of the operating conditions. These faults can become so severe that they can cause losses which can be irreversible. This study aims to present Fuzzy Analytic Hierarchy Process (AHP) and VIKOR (Vise Kriterijumska Optimizacija I Kompromisno Resenje) methods applied for the expert failure detection of marine diesel engine and auxiliary systems. In this study, the failures of marine diesel engine have been revealed and prioritized. Accordingly, the section of the machine from which the failures primarily arise has been determined. At the same time, the importance of the effective use of time in determining and responding to the failures has been indicated. By means of the evaluation of decision-making groups, the system most severely affected by failures has been decided.

**Key words:** Fuzzy AHP, Fuzzy VIKOR, MCDM, Failure detection, Auxiliary systems.

**1. Introduction**

When considering marine diesel engines, it is required that fuel, governor and the other systems work correctly to acquire desired power and ranges of rotation determined by the engine producers. Operating the engine out of this range and for a long time leads to serious failures.

Early warning instruments and measures such as heat, pressure, and flow sensors are available to detect failures. Precautions can be taken according to the values of these indicators that reflect failures. In case of the disruption of the operation of ship diesel engines, the engines should be removed entirely and the failures in power transfer are needed to be identified. Explicit connection of these failures with other systems should be revealed and efficiency values should be analyzed through expert systems.
Calder introduced a failure detection tool to control the fuel, oil, exhaust, combustion air and cooling water systems [1].

Even if the utilization of warning indicators and alarms are taken into account, early detection of possible machine failures is still quite difficult because of the dependency of these systems on each other.

In order to handle this problem, fuzzy multi-criteria decision-making method is suggested. Fuzzy analytic hierarchy process and the fuzzy technique are adopted in FAHP-VIKOR methods in order to detect marine diesel engine failures.

There have been several techniques discussed in the literature about failure analysis. Sharma et al. introduced a multi-factor decision-making approach for prioritizing Failure Mode Analysis using Technique for Order Preference by Similarity to Ideal Solution (TOPSIS) [2].

Çebi et al. developed an expert failure detection system to anticipate and overcome failures which take place in ship cooling system by the use of PROLOG programming language [3]. Taking into consideration the failure types that are already encountered; they created action tables to demonstrate what to do in the event of an emergency.

In the study carried out by Liu et al., linguistic variables, which are described in trapezoidal or triangular fuzzy numbers, are used to evaluate the ratings and weights for the risk factors [4]. When selecting the most severe failure modes, the expanded VIKOR method is utilized in order to determine the risk priorities of the failure modes that have been defined. Consequently, a fuzzy FMEA based on fuzzy set theory and VIKOR method is presented to prioritize failure modes which are specifically aimed to refer to some restrictions of the classical FMEA.

Ju and Aihua introduced a new method that makes it possible to overcome multi-criteria group decision-making problems in which both the criteria values and criteria weights take the form of linguistic information on the basis of the traditional idea of VIKOR method [5].

Anojkumar et al. depicted the implementation of four Multi Criteria Decision Making methods in order to solve the material selection problem of piping in sugar industry [6]. The four methods utilized to choose the best alternative among several different materials are FAHP-TOPSIS, FAHP-VIKOR, FAHP-ELECTRE (Elimination et choix traduisant la realite) and FAHP-PROMETHEE (Preference Ranking Organisation Method for Enrichment Evaluation).

Vinodh et al. introduced a research in which the concept selection in fit environment was developed as Multi Criteria Decision Making (MCDM) problem, and solutions were offered by utilising fuzzy based compromise solution method VIKOR [7]. Alarcin et al. examined failure detection in auxiliary systems and marine diesel engine determined by group of experts and determined the system most affected by failures [8].

Perovic et al. revealed guidelines on how to formalize fuzzy relational database queries [9]. The stability analysis of fuzzy logic control systems was done according to Lyapunov’s direct method by Precup et al [10]. Fodor and Baets examined uninorms of which both the underlying t-norm and underlying t-conorm are strict [11]. Martinez-Martín et al. presented a general framework to solve the representation magnitude and the basic step of inference process of qualitative models based on intervals [12].

In this study, it is aimed to present Fuzzy Analytic Hierarchy Process (AHP) and VIKOR (Vise Kriterijumska Optimizacija I Kompromisno Resenje) hybrid methods applied for the expert failure detection of marine diesel engine and auxiliary systems. In this respect, the failures of marine diesel engine have been revealed and prioritized. Accordingly, the section of the machine from which the failures primarily arise has been determined. At the same time, the importance of the effective use of time in determining and responding to the failures has been
indicated. By means of the evaluation of decision-making groups, the system most severely affected by failures has been decided.

In this present paper, Fuzzy AHP and Fuzzy VIKOR hybrid methods are used for the failure detection of marine diesel engine and auxiliary systems. The framework of this study is organized into five sections: In Section 1, the research methodologies are introduced. The model based on the Fuzzy AHP and Fuzzy VIKOR method is presented in Section 2 and Section 3. In Section 4, a discussion on the hierarchical structure employed for the problems of the operation of the ship diesel engine trouble-shooting using Fuzzy AHP and Fuzzy VIKOR methods is provided. Finally, the last section offers some concluding remarks.

2. Fuzzy AHP Approach

The research on Fuzzy AHP approach found in the literature can be summarized as follows. A method for group decision-making based on the multi-granularity uncertain linguistic information was proposed by Fan and Liu [13]. Ma et al. established a decision support system based on a model for enhancing the level of overall satisfaction in the multi-criteria group decision-making [14]. Yeh and Chang proposed a hierarchical weighting method for assessing weights; furthermore, they suggested an algorithm for classifying MDCM to combine criteria weights including decision makers’ subjective judgments [15]. Jiang and Fan examined the probability degree for triangular fuzzy number and introduced a new method on the basis of judgment matrix [16]. Xu and Da described the probability degree of interval number, and some desired properties were verified [17]. Lee presented a repetitive approximation procedure for aggregating individual opinions into the optimal consensus [18]. Mohammad et al. suggested a new method to overcome parametric form of fuzzy numbers problem and applied it to a case study of diversion of water [19]. Kacprzyk et al. put forward the assignment of fuzzy relations made by each expert [20]. They obtained a resulting preference relation from individual fuzzy preference relations to determine the best alternative. Dubois and Koning examined numerous fuzzy set aggregation connectors to assess their significance as social choice functions [21]. Cholewa propounded a collection of axioms for the aggregation of fuzzy weighted opinions and pointed out that the weighted mean satisfied those axioms [22].

Linguistic variable: A linguistic variable can be defined as a variable of which values consist of words or sentences in language naturally and artificially. Here, we employ this sort of expression to make a comparison among auxiliary system selection evaluation criteria by using several basic linguistic terms; “absolutely important,” “very strongly important,” “essentially important,” “weakly important” and “equally important” as to a fuzzy five level scale [23].

This study grounds the computational technique on the ensuing fuzzy numbers given in Table 1.

<table>
<thead>
<tr>
<th>Fuzzy number</th>
<th>Linguistic scales</th>
<th>Scale of fuzzy number</th>
</tr>
</thead>
<tbody>
<tr>
<td>̅</td>
<td>Equally important (EQ)</td>
<td>(1,1.3)</td>
</tr>
<tr>
<td>3</td>
<td>Weakly important (WK)</td>
<td>(1,3,5)</td>
</tr>
<tr>
<td>5</td>
<td>Essentially important (ES)</td>
<td>(3,5,7)</td>
</tr>
<tr>
<td>7</td>
<td>Very strongly important (VS)</td>
<td>(5,7,9)</td>
</tr>
<tr>
<td>9</td>
<td>Absolutely important (AB)</td>
<td>(7,9,9)</td>
</tr>
</tbody>
</table>

The linguistic variables shown in Table 1 are enjoyed to indicate the superior or weak
dimensions of AHP method by the five appointed groups in the criteria-criteria comparison.

Alternatives measurement: if the measurement of linguistic variables to show the criteria performance (effect-values) by expressions such as “very good,” “good,” “medium good,” “fair,” “medium poor,” “poor,” “very poor,” is used, the evaluators are required to carry out their subjective judgements, and all variables can be demonstrated by a Triangular Fuzzy Number (TFN) within the scale range 0–10, as shown in Table 2.

<table>
<thead>
<tr>
<th>Linguistic terms</th>
<th>Fuzzy score</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>

The linguistic variables presented in Table 2 are used to demonstrate the superiority or weakness status of VIKOR method by the five designated groups in the alternative-criteria comparison.

Besides, personal range of the linguistic variable that are possible indicators for the membership functions of the expression values of each evaluator can be assigned in a subjective way by evaluators. If \( E^k_j \) is taken to indicate the fuzzy performance value of evaluator \( k \) towards alternative “i” under the criterion j, and all of the criteria to evaluate are due to be illustrated by \( E^k_j = (LE^k_j, ME^k_j, UE^k_j) \) For the perception of all evaluators differs according to the evaluator s experience and knowledge, and the descriptions of the linguistic variables diverge as well, this study rests on the concept of average value to join the fuzzy judgment values of m evaluators, that is,

\[
\hat{E}^k_i = 1/m(LE^k_i, ME^k_i, UE^k_i)
\]

\( \hat{E}^k_i \) points out the average fuzzy number of the judgment of the decision-makers, which a triangular fuzzy number can display as \( LE^k_i, ME^k_i, UE^k_i \). The end-point values \( LE^k_i, ME^k_i, UT^k_i \) can be worked out by the method, as Buckley put it Buckley [26], that is,

\[
LE^k_i = \frac{1}{m} \sum_{i=1}^{m} LE^k_i \quad ME^k_i = \frac{1}{m} \sum_{i=1}^{m} ME^k_i \quad UE^k_i = \frac{1}{m} \sum_{i=1}^{m} UE^k_i
\]

**Fuzzy synthetic decision:** The weights of the all criteria of auxiliary system selection evaluation in addition to the fuzzy performance values need be unified by the calculation of fuzzy numbers, with a view to being located at the fuzzy performance value (effect-value) of the integral evaluation. According to the each criterion weight obtained by F-AHP, the criteria weight vector \( \bar{W} = (\bar{w}_1, ..., \bar{w}_n) \) can be acquired, but on the other hand the fuzzy performance matrix \( \bar{E} \) of the alternatives are highly likely to be derived from the fuzzy performance value of each alternative under n criteria, that is, \( \bar{E} = \bar{E}_y \) From the criteria weight vector \( \bar{W} \) and fuzzy performance matrix \( \bar{E} \), the final fuzzy synthetic decision can be carried
out, and the fuzzy synthetic decision matrix \( \hat{R} \) will provide the derived result, that is,
\[
\hat{R} = E \hat{w}
\] (3)

The sign ‘‘o’’ points out computing the fuzzy numbers as well as fuzzy addition and fuzzy multiplication. For the calculation of fuzzy multiplication can be taken quite complex, it is usually signified by the approximate multiplied result of the fuzzy multiplication and the approximate fuzzy number \( \hat{R} \) i, of the fuzzy synthetic decision of each alternative can be described as \( \hat{R} = (L\hat{R}_i, M\hat{R}_i, U\hat{R}_i) \), in which, \( L\hat{R}_i, M\hat{R}_i \) and \( U\hat{R}_i \) are the lower, middle and upper synthetic performance values of the alternative i, that is:
\[
L\hat{R}_i = \sum_{j=1}^{n} LE_{x,j} xL_{w,j}; M\hat{R}_i = \sum_{j=1}^{n} ME_{x,j} xM_{w,j}; U\hat{R}_i = \sum_{j=1}^{n} UE_{x,j} xU_{w,j};
\] (4)

**Ranking the fuzzy number:** The result of the fuzzy synthetic decision acquired by each alternative is a fuzzy number. Hence, it is essential that a nonfuzzy ranking method for fuzzy numbers be utilized to compare each building P&D alternative. To put in a different way, the procedure of defuzzification is to find the Best Nonfuzzy Performance value (BNP). Methods of such defuzzified fuzzy ranking generally involve mean of maximal (MOM), center of area (COA), and a-cut. To use the COA method to find out the BNP is a simple and practical method, and it is not needed to bring in the preferences of any evaluators, so it is benefited in this study. The BNP value of the fuzzy number \( \hat{R}_i \) can be found by the following equation:
\[
BNP_i = [(UR_i - LR_i) + (MR_i - MR_i)]/3 + LR_i \quad \forall i
\] (5)

According to the value of the derived BNP for each of the alternatives, the ranking of the building P&D of each of the alternatives can then proceed.

3. **Fuzzy VIKOR Approach**

VIKOR is a method developed on the basis of the compromise programming of MCDM. The implementation of the steps of VIKOR can be maintained by acquiring the weight vector through the extensive analyses. Yu and Zeleny first presented the concepts of compromise solutions [27-28]. The methodology, merely works on the principle that each alternative can be evaluated by each criterion function, which enables the compromise ranking to be obtained by comparing the degrees of proximity to the ideal alternative. In fuzzy VIKOR, it is proposed that decision makers utilise linguistic variables to evaluate the ratings of alternatives according to the criteria. The linguistic scale for the evaluation of alternatives is presented in Table 2. Supposing that a decision-making group has K people, the ratings of alternatives with reference to each criterion can be computed as herein below [29]:
\[
x_{ij} = \frac{1}{K} \left[ x_{ij}^{(1)} + x_{ij}^{(2)} + \ldots + x_{ij}^{(K)} \right]
\] (6)

where \( x_{ij}^{(k)} \) is the rating of the K-th expert for ith alternative with regard to jth criterion.
After acquiring the weights of criteria and fuzzy ratings of alternatives corresponding to each criterion, the fuzzy multi-criteria decision-making problem in matrix format can be explained as,
\[
D = \begin{bmatrix}
    x_{11} & \ldots & x_{1n} \\
    \vdots & \ddots & \vdots \\
    x_{m1} & \cdots & x_{mn}
\end{bmatrix}
\]
\[
W = [w_1, w_2, \ldots, w_n] \quad j = 1, 2, \ldots, n
\]
where \( x_{ij} \) is the rating of Alternative Ai with reference to Criterion j (i.e. \( C_j \)) and \( w_j \) indicates
A Hierarchical Structure for Ship Diesel Engine Trouble-Shooting

Abit Balin, Hakan Demirel, Fuat Alarçin

Problem Using Fuzzy AHP and Fuzzy VIKOR Hybrid Methods

the importance weight of $C_j$.

The next step will be the determination of the Fuzzy Best Value ($FBV$, $\tilde{t}_j^*$) and the Fuzzy Worst Value ($FWV$, $\tilde{t}_j^-$) of each criterion function.

$$\tilde{t}_j^* = \max_{x_j \in B}, \quad \tilde{t}_j^- = \min_{x_j \in C}$$ (8)

Then, the values $w_j(\tilde{t}_j^* - x_j) / (\tilde{t}_j^* - \tilde{t}_j^-)$, $\tilde{S}_i$ and $\tilde{R}_i$ are calculated as follow,

$$\tilde{S}_i = \sum_{j=1}^{n} w_j(\tilde{t}_j^* - x_j) / (\tilde{t}_j^* - \tilde{t}_j^-)$$ (9)

$$\tilde{R}_i = \max_j \left[ w_j(\tilde{t}_j^* - x_j) / (\tilde{t}_j^* - \tilde{t}_j^-) \right]$$ (10)

where $\tilde{S}_i$ signifies the separation measure of $A_i$ from the fuzzy best value, and $\tilde{R}_i$ the separation measure of $A_i$ from the fuzzy worst value.

$$\tilde{S}_i^- = \min_i \tilde{S}_i, \quad \tilde{S}_i^* = \max_i \tilde{S}_i, \quad \tilde{R}_i^- = \min_i \tilde{R}_i, \quad \tilde{R}_i^* = \max_i \tilde{R}_i$$ (11)

In the next step, $\tilde{S}_i^-$, $\tilde{S}_i^*$, $\tilde{R}_i^-$, $\tilde{R}_i^*$, and $Q_i$ values are calculated as

$$Q_i = v(\tilde{S}_i^- - \tilde{S}_i^*) / (\tilde{S}_i^* - \tilde{S}_i^-) + (1 - v)(\tilde{R}_i^- - \tilde{R}_i^*) / (\tilde{R}_i^* - \tilde{R}_i^-)$$ (12)

The indices $\min_i \tilde{S}_i$ and $\min_i \tilde{R}_i$ are relevant to a maximum majority rule and a minimum individual regret of an opponent strategy, respectively. In addition, $v$ is presented as the weight of the strategy of the maximum group utility. “v” is usually assumed to be 0.5.

The next task is the defuzzification of the triangular fuzzy number $Q_i$, and ranking the alternatives by the index $Q_i$. Different defuzzification strategies have been suggested in the literature. In this present study, the graded mean integration approach is adapted [30]. According to the graded mean integration approach, for triangular fuzzy numbers, a fuzzy number $C = (c_1, c_2, c_3)$ can be changed into a crisp number by utilising the equation below:

$$P(C) = \frac{c_1 + 4c_2 + c_3}{6}$$ (13)

Finally, the best alternative with the minimum of $Q_i$ is determined.

Methodology steps of application for Fuzzy AHP-VIKOR hybrid method is summarized as follows in Figure 1.
Step 1: Constructing pairwise comparison matrices among all the criteria in the dimensions of the system hierarchy.

Step 2: Calculating the elements of synthetic pairwise comparison matrix by utilising the geometric mean method proposed by Buckley:

Step 3: Likewise, we can obtain the remaining $\tilde{r}_i$.

Step 4: For the weight of each dimension, below mentioned processes can be followed

Step 5: Fuzzy best value (FBV, $\tilde{r}_i^*$) and fuzzy worst value (FWV, $\tilde{r}_i^-$) of each criterion function are determined.

Step 6: Separation measures ($\bar{S}_i$ and $R_i$) are computed.

Step 7: $Q_i$ values are calculated.

Step 8: $Q_i$ values are defuzzified and the alternatives are ranked by the index $Q_i$.

Step 9: The best alternative with the minimum of $Q_i$ is determined.

4. Trouble Shooting Application In Marine Diesel Auxiliary Engines Via FAHP-VIKOR Approach

In most cases, it is seen that faults cause serious damage and considerable loss of capital investment. In this paper, five auxiliary systems, resulting in various realistic events are taken into consideration. The failures listed herein below are explained further. The severity levels of these faults are different. Some of these failures are so severe that a fast fault detection and adjustment is needed to avoid serious accidents in case of a component failure during the operating conditions.

Causes and symptoms of failures in marine diesel engines examined mostly turn out to be precursors of a further breakdown. In every failure, any reason is not found instantly but during the operating conditions. The hierarchical structure suited in this work to cope well with the problems of operation of the machine assessment for ship is shown in Fig. 2.

Figure 2. The hierarchical structure designed for ship machine systems

Any probable main engine breakdown can be identified by using the efficient main engine failure detection. In addition to the recognized symptoms and the detected faults, the frequency of faults related to auxiliary systems should also be taken into account in order to find out the possible causes of failure which increases the productivity of the managing systems.
The key aspects of the criteria to evaluate and select machine operation systems for ship alternatives were obtained from extended investigation and consultation in three groups, with a professor in department of Naval Architecture and Marine Engineering.

They were requested to rate the accuracy, adequacy and relevance of the criteria and dimensions and to confirm “content validity” with regard to operation of the machine assessment. Reasons for failures in the main engine systems were drawn from former records, maintenance log-books and is consolidated with the experience of personnel. Six kinds of failures of high priority show up when aforementioned failures are monitored. Failures are identified as $C_i$ in which “i” is the number of pertinent failure.

**Table 3.** Auxiliary systems for main engine failures criteria

<table>
<thead>
<tr>
<th>$C_1$. High heat level in all exhaust cylinders of the engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{11}$. Fuel injector problems</td>
</tr>
<tr>
<td>$C_{12}$. Exhaust valve failure</td>
</tr>
<tr>
<td>$C_{13}$. Blower not working fully</td>
</tr>
<tr>
<td>$C_{14}$. Wrong adjustment of governor</td>
</tr>
<tr>
<td>$C_{15}$. Insufficient intake air</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$C_2$. Unstable engine speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{21}$. Dirty fuel oil filter</td>
</tr>
<tr>
<td>$C_{22}$. Booster pump pressure</td>
</tr>
<tr>
<td>$C_{23}$. Fouling in the turbocharger</td>
</tr>
<tr>
<td>$C_{24}$. Wrong adjustment of governor</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$C_3$. Shut down of the engine during normal operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{31}$. Low-level day tank</td>
</tr>
<tr>
<td>$C_{32}$. Low- low Oil pressure</td>
</tr>
<tr>
<td>$C_{33}$. High Pressure Fuel pump failures</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$C_4$. Increase of the oil level during engine operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{41}$. Cooling water leakage</td>
</tr>
<tr>
<td>$C_{42}$. Fuel oil leakage</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$C_5$. Fire in the Scavenging area</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{51}$. Dirty scavenging manifold inlet</td>
</tr>
<tr>
<td>$C_{52}$. Scuffing of the piston oil ring and piston</td>
</tr>
<tr>
<td>$C_{53}$. Air cooler problem</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$C_6$. Surge in the turbocharger</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{61}$. Exhaust valve burns</td>
</tr>
<tr>
<td>$C_{62}$. Mechanical failure in the turbocharger</td>
</tr>
<tr>
<td>$C_{63}$. Scavenging pressure high</td>
</tr>
</tbody>
</table>

Criteria were explained how the individual subsystems affect the engine operation as follows [1-31];
**High heat level in all exhaust cylinders of the engine:** Wrong adjustment of governor determine the amount of fuel supplied to the combustion chamber. The lack of an optimal mixture ratio in the combustion chamber reduces the combustion quality and this situation causes an increase of the exhaust temperature.

The ventilation does not work properly can cause insufficient amount of oxygen incoming from the combustion chamber. Exhaust temperature increases due to the lack of non-uniform combustion. Exhaust valve failure reduces the combustion quality because of the decrease in compression pressure. Problems in the fuel injector cause taking the unburnt fuel inside the combustion chamber, combustion continues after ignition and this situation cause increasing the exhaust temperature.

**Unstable engine speed:** Dirty fuel oil filter and low booster pump pressure reduce the inlet pressure of the fuel supplied to the engine and this situation makes it difficult to provide sufficient fuel and unstable engine speed occur. Fouling in the turbocharger cause failure in the the balance of the turbocharger and turbocharger speed changes, this situation cause fluctuations in compressed air pressure and counter- pressure on the exhaust side. Wrong adjustment of governor gives rise to errors in the fuel feed rate and leads to imbalance in engine speed.

**Shut down of the engine during normal operation:** Low-level day tank give rise discontinuation of fuel supplied to the engine and engine stops. In any pump failure, oil pressure decreases and if oil pressure is not enough, engine will not work so switch gives the instruction and engine is stopped. High Pressure Fuel pump failures cause absence fuel into combustion chamber because of insufficient pressure so engine stops or engine speed changes.

**Increase of the oil level during engine operation:** Cooling water leakage cause water leakage into the crankcase and this situation increases oil level in crankcase. Fuel oil leakage cause spread of fuel into the crankcase.

**Fire in the Scavenging area:** Dirty inlet manifold means that the presence of combustible materials at the location and combustion takes place here in the formation of the necessary conditions for combustion. Scuffing of the piston oil ring and piston cause to move scavenging area from the combustion chamber of combustion and combustion occur in here. Due to air cooler problem, compression air come to the scavenging area without cooling, high temperature air cause combustion in here.

**Surge in the turbocharger:** Burns that occur in the exhaust valve cause gas leakage into the exhaust manifold except egzost time. This situation cause temperature fluctuations in the turbine inlet and occur the turbine speed fluctuations. Mechanical failure in the turbocharger disrupt the turbocharger balance and this situation cause speed fluctuations in addition it gives rise to noisy operation.

When above mentioned engine faults, which vary from one another in terms of basic characteristics are technically analyzed with the aim of classifying, it is recognized that each has a relationship with a different system. The fact that failures in particular groups which build a relationship along with the ones in other groups is also known. Considering the causes for failures, auxiliary systems in connection with the failures can be categorized as follows:

**A1. Fuel System**
**A2. Cooling System**
**A3. Governor System**
**A4. Air supply System**
We can acquire the fuzzy evaluation and "\( Q \) " values of other alternatives for comparison; finally, details of the results are shown in Table 5.

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Weights</th>
<th>BNP</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>C1. High heat level in all exhaust cylinders of the engine</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C11. Fuel injector problems</td>
<td>(0.076, 0.166, 0.436)</td>
<td>0.226</td>
</tr>
<tr>
<td>C12. Exhaust valve failure</td>
<td>(0.186, 0.407, 0.962)</td>
<td>0.518</td>
</tr>
<tr>
<td>C13. Blower not working fully</td>
<td>(0.116, 0.292, 0.667)</td>
<td>0.358</td>
</tr>
<tr>
<td>C14. Wrong adjustment of governor</td>
<td>(0.038, 0.070, 0.203)</td>
<td>0.104</td>
</tr>
<tr>
<td>C15. Insufficient intake air</td>
<td>(0.058, 0.153, 0.343)</td>
<td>0.185</td>
</tr>
<tr>
<td><strong>C2. Unstable engine speed</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C21. Dirty fuel oil filter</td>
<td>(0.091, 0.257, 0.671)</td>
<td>0.34</td>
</tr>
<tr>
<td>C22. Booster pump pressure</td>
<td>(0.064, 0.145, 0.492)</td>
<td>0.234</td>
</tr>
<tr>
<td>C23. Fouling in the turbocharger</td>
<td>(0.112, 0.268, 0.762)</td>
<td>0.381</td>
</tr>
<tr>
<td>C24. Wrong adjustment of governor</td>
<td>(0.107, 0.329, 0.757)</td>
<td>0.397</td>
</tr>
<tr>
<td><strong>C3. Shut down of the engine during normal operation</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C31. Low-level day tank</td>
<td>(0.059, 0.110, 0.317)</td>
<td>0.162</td>
</tr>
<tr>
<td>C32. Low- low Oil pressure</td>
<td>(0.128, 0.285, 0.733)</td>
<td>0.382</td>
</tr>
<tr>
<td>C33. High Pressure Fuel pump failures</td>
<td>(0.257, 0.605, 1.206)</td>
<td>0.689</td>
</tr>
<tr>
<td><strong>C4. Increase of the oil level during engine operation</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C41. Cooling water leakage</td>
<td>(0.376, 0.781, 1.457)</td>
<td>0.871</td>
</tr>
<tr>
<td>C42. Fuel oil leakage</td>
<td>(0.132, 0.219, 0.513)</td>
<td>0.288</td>
</tr>
<tr>
<td><strong>C5. Fire in the Scavenging area</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C51. Dirty scavenging manifold inlet</td>
<td>(0.128, 0.285, 0.733)</td>
<td>0.382</td>
</tr>
<tr>
<td>C52. Scuffing of the piston oil ring and piston</td>
<td>(0.257, 0.605, 1.206)</td>
<td>0.689</td>
</tr>
<tr>
<td>C53. Air cooler problem</td>
<td>(0.059, 0.110, 0.317)</td>
<td>0.162</td>
</tr>
<tr>
<td><strong>C6. Surge in the turbocharger</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C61. Exhaust valve burns</td>
<td>(0.128, 0.285, 0.733)</td>
<td>0.382</td>
</tr>
<tr>
<td>C62. Mechanical failure in the turbocharger</td>
<td>(0.257, 0.605, 1.206)</td>
<td>0.689</td>
</tr>
<tr>
<td>C63. Scavenging pressure high</td>
<td>(0.059, 0.110, 0.317)</td>
<td>0.162</td>
</tr>
</tbody>
</table>

Depending on the Fuzzy AHP results, for the decision-making groups, we conclude that the first two most important aspects are the Shutdown of the engine during normal operation (0.395) and the Fire in the Scavenging area (0.370) whereas the least important aspect is the unstable engine speed (0.053). When considered the decision-making groups, the first two important sub-criteria in Shut down of the engine during normal operation are the High Pressure Fuel pump failures (0.689) and the Low- low Oil pressure (0.382), whereas the least important aspect is the Low-level day tank (0.162). Additionally, for the groups of experts, the most important sub-criteria in the Fire in the Scavenging area are presented respectively; as the Scuffing of the piston oil ring and piston (0.689), the dirty scavenging manifold inlet (0.382) and the Air cooler problem (0.162). Nevertheless, the first two important dimensions in the least important criteria are the Wrong adjustment of governor (0.397) and the Fouling in the turbocharger (0.381), and the least is the Booster pump pressure (0.234).

These results denote that the decision-making groups’ concern is the safety of managing the Shutdown of the engine during normal operation. They also pay attention to the Fire in the Scavenging area, which will be considered the suitability of freighter operating. The decision-making groups focus on the associated professional issues for the Shutdown of the engine during normal operation, but they consider that the High Pressure Fuel pump failures and Low- low Oil pressure are stable to be secured under professional calculations, so they rate it attaching great importance.

We can acquire the fuzzy evaluation and “\( Q \) ” values of other alternatives for comparison; finally, details of the results are shown in Table 5.
Table 5. The evaluation results

<table>
<thead>
<tr>
<th>Alternatives</th>
<th>Fuzzy Evaluation</th>
<th>Q_i</th>
<th>Ranking</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1: Fuel System</td>
<td>0.500</td>
<td>0.500</td>
<td>0.500</td>
</tr>
<tr>
<td>A2: Cooling System</td>
<td>0.345</td>
<td>0.350</td>
<td>0.367</td>
</tr>
<tr>
<td>A3: Governor System</td>
<td>0.715</td>
<td>0.693</td>
<td>0.658</td>
</tr>
<tr>
<td>A4: Air supply System</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
</tr>
</tbody>
</table>

As can be seen from the results of alternative evaluation in Table 5, the Cooling System is considered as the most affected alternative by errors regarding the weights of all decision-making groups. The results shown in Table 4 demonstrate the common perception that the changes in criteria weights may have an impact on the evaluation outcome to a certain extent. Moreover, it can obviously be seen that the air supply system is the least affected alternative by errors in comparison to the other alternatives, which is the most common consensus among the groups.

5. Conclusions

The engine can quickly be affected by a failure that occurs in any system and this failure can cause a breakdown or a malfunction in the engine. The reason of the failure should be immediately found out and repaired by expert applications. To help the chief engineers, the conditions in which those failures occurred in marine engine system should be figured out and methods must be developed to decrease the rates of failures.

In this paper, the hierarchical structure is adapted to the troubleshooting of main engine auxiliary systems, including cooling, governor, air supply and fuel systems. By means of FAHP and VIKOR hybrid methods, a more efficient decision for engine failure evaluation can be made. Taking into account all the results in Table 5, in FAHP-VIKOR approach, it can be concluded that all decision making groups agree that the most severely influenced system is the Cooling System.

REFERENCES

A Hierarchical Structure for Ship Diesel Engine Trouble-Shooting

Problem Using Fuzzy AHP and Fuzzy VIKOR Hybrid Methods

Abit Balin, Hakan Demirel, Fuat Alarçın


Submitted: 20.10.2014. Abit Balin, Engineering Faculty Yalova University
Hakan Demirel, Naval Architecture and Maritime Faculty

Accepted: 20.12.2104. Yıldız Technical University, 34349 Beşiktas – İstanbul Turkey
Fuat Alarçın, falarcın@gmail.com, Naval Architecture and Maritime Faculty
Yıldız Technical University, 34349 Beşiktas – İstanbul Turkey
ADDED RESISTANCE IN WAVES OF INTACT AND DAMAGED SHIP IN THE ADRIATIC SEA (str.1-14)
Ivana Martić, Nastia Degiuli, Ivan Ćatipović
Original scientific paper

THE PROCEDURE FOR CALCULATION OF THE OPTIMAL CARRYING CAPACITY OF PUSHED CONVOY BASED ON PARAMETERS OBTAINED BY EXPERIMENTS IN ACTUAL NAVIGATING CONDITIONS (str.15-28)
Ivan Škiljaica, Ilija Tanackov, Vladislav Maraš
Original scientific paper

AN EXPERIMENTAL AND NUMERICAL PREDICTION OF MARINE PROPELLER NOISE UNDER CAVITATING AND NON-CAVITATING CONDITIONS (str.29-45)
Mohammad Reza Bagheri, Hamid Mehdigholi, Mohammad Saïd Seif, Omar Yaakob
Original scientific paper

EXPERIMENTAL STUDY OF TWO LARGE-SCALE MODELS’ SEAKEEPING PERFORMANCE IN COASTAL WAVES (str.47-60)
Shu-zheng Sun, Hui-long Ren, Xiao-dong Zhao, Ji-de LI
Original scientific paper

STEAM AND SOFC BASED REFORMING OPTIONS OF PEM FUEL CELLS FOR MARINE APPLICATIONS (str.61-76)
Mohamed M. El Gohary, Nader R. Ammar, Ibrahim S. Seddiek
Original scientific paper

SHIPBUILDING PRODUCTION PROCESS DESIGN METHODOLOGY USING COMPUTER SIMULATION (str.77-91)
Marko Hadjina, Nikša Fafandjel, Tin Matulja
Professional paper
ADDED RESISTANCE IN WAVES OF INTACT AND DAMAGED SHIP IN THE ADRIATIC SEA

UDC 629.5.015.24(262.3)
Original scientific paper

Summary

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

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

1. Introduction

In order to evaluate the safety of ship navigation and the possibility of carrying out the designed purpose of the ship, it is necessary to know its seakeeping characteristics. In order to ensure good seakeeping characteristics, the ship motion amplitudes and accelerations must be within acceptable limits and the added resistance in waves should not have a significant effect on the decrease in ship speed and the increase in power and fuel consumption [1]. Sailing in waves also affects the ship maneuverability. Due to the ship added resistance in waves, the power and fuel consumption of the ship can increase up to 85% compared to sailing in calm waters [2]. When a ship oscillates in waves, energy is transmitted to the surrounding water due to damping of oscillatory motions. Added resistance is caused by the drift force due to the
interference of incoming waves and the waves due to heave and pitch motions, the hydrodynamic damping of heave and pitch motions in calm water (radiation) and diffraction due to the interaction of the incoming waves and the ship. Drift force is the most important component of added resistance. The diffraction induced added resistance is dominant for relatively high wave frequencies, while the viscous damping can be neglected [3]. Added resistance can be determined with certain accuracy experimentally and by using different analytical and numerical methods. The accuracy of added resistance calculation depends on the accuracy of the ship motion prediction [4].

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

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

The analysis was performed using hydrodynamic software [7].

2. Added resistance in waves

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

When a ship sails in waves, its forward speed decreases due to the increasing resistance. Therefore, to sustain the forward speed, engine power and fuel consumption are increasing. The ship speed loss is also caused by several other factors such as the resistance due to wind acting on the hull and superstructure, resistance due to diffraction of waves, due to reduction of the propulsion efficiency as well as due to ship maneuverability, etc. [1]. Generally speaking, a ship can experience a 15-30% resistance increase in a seaway mainly caused by added resistance in waves [3].
There are two basic approaches for the estimation of time averaged added resistance: a radiated wave energy method and an integrated pressure method [8]. The radiated energy method is based on the strip theory and determines radiated wave energy during one period of ship oscillation as follows:

\[ P = \int_{t}^{T} \left[ b_{33} \nu^2 \right] dx \, dt \]  

where \( b_{33} \) is the hydrodynamic damping coefficient of the vertical motion of the cross section, \( \nu^2 \) is the vertical average velocity of the water particles relative to the cross section, \( T \) is the period of vertical oscillation of the cross section.

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

\[ \frac{R_{AW}}{\xi^2} = -\frac{k \cos \beta}{2\omega_e} \int \left[ N_{33} - v \frac{dM_{33}}{dx} \right] \left( \frac{\nu^2}{\xi^2} \right)^2 \, dx \]  

where \( k \) is the wave number, \( \beta \) is the wave direction, \( \omega_e \) is the encounter frequency, \( N_{33} \) is the two-dimensional potential damping coefficient for heave, \( \nu \) is the ship speed, \( M_{33} \) is the two-dimensional potential mass coefficient for heave.

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

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

The contribution to the total mean added resistance at the given encounter frequency \( \omega_e \) is finally given by:

\[ \frac{R_{AW}}{\xi^2} = \frac{1}{2} \rho g \left[ \frac{s_{a}}{\xi} \right]^2 \, dx_{s} + \frac{1}{2} \rho \nabla \omega_{e} \frac{z_{a}}{\xi} \frac{\theta_{a}}{\xi} \cos \left( \varepsilon_{e} - \varepsilon_{e} \right) \]  

where \( s_{a} \) is the vertical relative motion, \( s = \zeta - z_{s} \), \( \nabla \) is the volume of displacement, \( z_{a} \) is the vertical motion, \( \theta_{a} \) is the pitch angle, \( \varepsilon_{e} \) is the phase shift for heave, \( \varepsilon_{e} \) is the phase shift for pitch.

In order to simulate the fluid flow around a ship when sailing in waves as simply as possible with satisfactory suitability, the linear potential flow theory is used. It is based on the simplest form of two-dimensional progressive surface wave of small amplitude. Fluid flow in potential theory is irrotational, homogeneous, inviscid and incompressible, and the velocity field is described by a continuous scalar function that has a finite value in every point of the fluid and defines the velocity vector. The linear potential flow theory is based on the Laplace equation (continuity of homogeneous incompressible fluid), the Euler equation that defines the balance of forces on the material particle and the Bernoulli equation, i.e. the law of conservation of energy per unit mass.
The hydrodynamic software [7] used to perform necessary calculations is based on the boundary integral equations and linear potential flow theory. In order to take into account the ship forward speed, encounter frequency approximation is implemented in the software [7] based on the Green function. The translating-pulsating source method based on the Green function is used in hydrodynamic calculations for cases where forward speed is simulated. Since the method is very time consuming and numerically demanding, when the forward speed is small in amplitude, an approximate free surface boundary condition can be used:

\[-\omega \varphi + g \frac{\partial \varphi}{\partial z} = 0 \quad \text{on} \quad z = 0 \tag{4}\]

where \(\varphi\) is the fluid potential.

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

3. Quadratic transfer function QTF and approximation of QTF

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

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

\[ F(\omega_i, \omega_j) = F_s(\omega_i, \omega_j) + F_r(\omega_i, \omega_j) \tag{5}\]

where \(F_s(\omega_i, \omega_j)\) is the first-order wave load, \(F_r(\omega_i, \omega_j)\) is the second-order wave load, \(\omega_i\) and \(\omega_j\) are the incoming wave frequencies.

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

The Haskind integral allows the elimination of the unknown diffraction potential by replacing it with radiation potential [1]:
\[ \int_{S} \partial \frac{\phi_{i}}{\partial n} dS = -\int_{S} \partial \frac{\phi_{k}}{\partial n} dS, \quad k = 1\ldots6 \]  
(6)

where \( \phi_{i} \) is the diffraction potential, \( \phi_{k} \) is the radiation potential, \( \phi_{0} \) is the incoming wave potential, \( S_{0} \) is the wetted surface.

Since the QTF is assumed to be a regular function of the incoming wave frequencies \( \omega_{i} \) and \( \omega_{j} \) and taking into account the relation \( \Delta \omega = \omega_{i} - \omega_{j} \), the QTF is developed as Taylor series expansion:

\[ F(\omega_{i}, \omega_{j}) = F_{0}(\omega_{i}) + F_{1}(\omega_{i}) \Delta \omega + F_{2}(\omega_{i})(\Delta \omega)^{2}/2 + \ldots \]  
(7)

The zeroth-order term \( F_{0}(\omega_{i}) \) represents the load due to integration of pressure along the hull wetted surface, i.e. the drift force. The often used approximation proposed by Newman when determining that term can significantly underestimate the second-order wave loads and also can provide wrong phase shift since the approximation is a real function while the QTF is a complex function. The zeroth-order term depends on the incoming wave frequency as the mean value of two frequencies \( (\omega_{i} + \omega_{j})/2 \), the first-order term is linearly proportional to the difference of two frequencies \( \omega_{i} - \omega_{j} \) and the second-order term is proportional to the square of difference between the two frequencies.

The term \( F_{1}(\omega_{i}) \) is composed of four parts:

\[ F_{1}(\omega_{i}) = F_{1}^{q} + F_{1}^{r_{1}} + F_{1}^{r_{2}} + F_{1}^{r_{3}} \]  
(8)

where \( F_{1}^{q} \) is the contribution of the first-order wave load, \( F_{1}^{r_{1}} \) is the contribution of the second-order incoming wave load and diffraction waves, \( F_{1}^{r_{2}} \) is the second-order correction of the boundary condition on the hull, \( F_{1}^{r_{3}} \) is the effect of forcing pressure over the free surface (second-order correction of the boundary condition on the free surface).

The term \( F_{1}(\omega_{i}) \) does not have analytical expression but it can be defined as the difference of the total QTF function of the bichromatic wave field and the zeroth and the first-order term previously mentioned:

\[ F_{1}(\omega_{i}) = \frac{2}{(\Delta \omega)^{2}} \left[ F\left(\omega_{i}, \frac{\Delta \omega}{2}, \omega_{i} - \frac{\Delta \omega}{2}\right) - F_{0}(\omega_{i}) - F_{1}(\omega_{i}) \Delta \omega \right] \]  
(9)

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

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

In order to numerically calculate the added resistance, hydrodynamic software was used [7]. The software is based on the potential flow theory and provides the solutions of the first- and second-order motions and loads. The hull characteristics and parameters of the waves are necessary as an input. It allows calculation of diffraction and radiation components of the ship
added resistance, the interaction between the ship and the waves and the interaction between several bodies, etc. Since the software is based on the potential flow theory, a fictitious force in dynamic motion equations is added in order to take into account the energy dissipation and damping to avoid the infinite response at resonant frequencies.

![Figure 1 Body plan of the S-175 container ship [6]](image1)

![Figure 2 Panel model of the S-175 container ship [7]](image2)

<table>
<thead>
<tr>
<th>Table 1 Principal dimensions of the S-175 container ship</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Full scale</strong></td>
</tr>
<tr>
<td>Length $L$ (m)</td>
</tr>
<tr>
<td>Breadth $B$ (m)</td>
</tr>
<tr>
<td>Draft of fore peak $T_f$ (m)</td>
</tr>
<tr>
<td>Draft of midship $T$ (m)</td>
</tr>
<tr>
<td>Draft of aft peak $T_a$ (m)</td>
</tr>
<tr>
<td>Displacement volume $\nabla$ (m$^3$)</td>
</tr>
<tr>
<td>Block coefficient $C_b$</td>
</tr>
<tr>
<td>Position of LCG $x_g$ (m)</td>
</tr>
<tr>
<td>Radius of gyration $k_y/L$</td>
</tr>
</tbody>
</table>

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

1. The **near field formulation** needs the evaluation of the first-order wave field around the hull and along the waterline, as well as the first-order motions caused by that field. Drift load is determined by direct numerical integration of pressure along the defined hull form wetted surface.

2. Another formulation based on the momentum theorem for the horizontal drift forces involves the first-order wave field in the far field and is often called **far field formulation** and is preferable in practice thanks to its better convergence of drift forces and stability. However, the far field formulation cannot provide the accurate vertical drift loads and the low-frequency loads $Q_{TF}$ which can be both of great importance in shallow water.

3. The **middle field formulation** combines the advantages of both near field and far field formulation providing sufficiently accurate numerical results and possibility of calculating all components of drift loads and low-frequency loads $Q_{TF}$ as the near field formulation. The method is based on the analysis in a finite volume limited by
Added resistance in waves of intact and damaged ship in the Adriatic Sea

Ivana Martić, Nastia Degiuli, Ivan Ćatipović

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

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

$$F(t) = \Re \left\{ \sum_{i=1}^{\infty} \sum_{j=1}^{\infty} F^i_j a^i_j a^*_j \exp \left[ -i(\omega_i - \omega_j) t \right] \right\}$$

(10)

where $F^i_j$ is the drift force in waves, $a^i_j$ is the complex amplitude of incoming wave, $a^*_j$ is the complex conjugate amplitude of incoming wave, $\omega_i, \omega_j$ are the incoming wave frequencies.

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

The added resistance force in waves is defined as:

$$F(t) = F^i_j a^i_j^2$$

(11)

where $a^i_j$ is the amplitude of head incoming waves.

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

$$C_{xw} = \frac{F(t)}{\rho g a^i_j^2 B^2 / L}$$

(12)

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

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

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

<table>
<thead>
<tr>
<th>Froude number $Fr$</th>
<th>Velocity $v$, m/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>0.087</td>
<td>0.472</td>
</tr>
<tr>
<td>0.148</td>
<td>0.803</td>
</tr>
<tr>
<td>0.198</td>
<td>1.074</td>
</tr>
</tbody>
</table>

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

The abbreviations of the aforementioned methods of calculating the second-order low-frequency wave loads are the following:
• PRE - direct integration of pressure along the wetted surface (near field formulation),
• MOM - method based on the momentum theorem (far field formulation),
• CSF - method using control surface and far field potentials (middle field formulation).

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

**Figure 4** Comparison of added resistance coefficients for Froude number $Fr = 0.087$
As shown in Figures 3-6 there are certain deviations in the numerically calculated added resistance coefficients compared to the experimental data. Since the calculations are based on the potential flow theory and certain simplifications were introduced, it was not possible to fully simulate the real flow and the forces and loads acting on the model during the tests. The method based on the integration of pressure along the hull wetted surface (PRE) and the method that uses a combination of pressure integration and analysis of volume bounded by the ship hull and the control surface (CFS) give identical results and show satisfactory agreement with the experimental data.

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

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

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

$$ R_{aw} = C_{xw} \rho g \zeta \frac{B^2}{L} $$

(13)

Added resistance in dependence on the ratio of wavelength and length of the model $\lambda / L$ is shown in Figure 7.
5. **Added resistance of intact and damaged ship at defined sea states**

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

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

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

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

In order to simulate the impact of sloshing inside the hull due to damage, a flooded tank is generated inside the mesh of the S-175 container ship as shown in Figure 8. The volume of liquid in the tank is 48.49% of the volume displacement of the undamaged ship on a draft equal to $T = 12.01667$ m, or 32.74% of the volume displacement of the damaged ship that has vertically immersed for $\Delta T = 3.483$ m.
Fluid inside the tank was first treated as the ship added displacement mass but without affecting the global motions and loads. After that, the seakeeping characteristics of the ship were calculated considering also the sea water inside the tank in order to determine the impact of motion and loads of liquid in the tank on global ship motions at a defined sea state. The calculation is based on the Tabain wave spectrum as a modification of JONSWAP (Joint North Sea Wave Project) spectrum for the Adriatic Sea to assess the safety of towing the ship in the sea conditions characteristic for the Adriatic Sea.

The Tabain wave spectrum is defined as follows:

$$S_{\xi}(\omega) = 0.862 \frac{0.0135 g^2}{\omega^3} e^{\left[\frac{-5.186}{\omega^2 H_{\xi}}\right]^{1.63}}$$

where

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

$$\omega_m = 0.32 + \frac{1.8}{H^{1/3} + 0.60}$$

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

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

$$\overline{R_{AW}} = 2 \int_0^\infty \frac{R_{AW}(\omega)}{\xi_a^2} (\omega) S_{\xi}(\omega) d\omega$$

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

Relation between the mean added resistance and the regular wave amplitude squared is linear, and the wave spectrum is assumed to be ideal. The spectral function is obtained by multiplying drift force at certain frequency and the corresponding ordinate of the Tabain
spectrum. The first spectral moment is determined by the trapezoidal rule of integration. The mean values of the added resistance at two defined sea states determined by using the equation (18) are based on the absolute values of the drift forces obtained using the hydrodynamic software. For a certain frequency range, these values are shown in dependence on the ship speed in Figures 9 and 10.

![Figure 9](image9)

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

![Figure 10](image10)

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

Since the second-order wave load or quadratic transfer function QTF was approximated using the Newman formulation with the zeroth-order term only, the ship added resistance is thus underestimated. In order to compare the calculated results with the data from the literature and to substantiate this assumption, the calculation of added resistance coefficients of the S-175 container ship for Froude number $Fr = 0.275$ was conducted. A comparison is made between the calculated results and the data obtained experimentally [11] and numerically using the three-dimensional nonlinear numerical panel method based on the Rankine potential (the Green function) that allows modelling of the nonlinear steady flow in the frequency domain [12]. The results are shown in Figure 11.
It can be concluded from Figure 11 that the curve form of the calculated values of dimensionless added resistance follows the trend of the experimental and numerical data from the literature. Since in this paper added resistance is based on the potential theory, there are certain deviations in relation to the experimental data. It may also be noted that the extreme value of the resulting dimensionless function of added resistance in dependence on the ratio \( \lambda/L \) occurs at a lower \( \lambda/L \) ratio than in the case of the numerical and experimental data from the literature.

6. Conclusion

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

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

It is necessary to adjust the boundary conditions and carefully determine the damping factor which will prevent infinite responses especially in the case of sloshing inside the ship.
Despite numerous simplifications, the linear potential flow theory gives satisfactory results and greatly simplifies the calculation of the interaction of the ship and incoming waves.

**AKNOWLEDGMENTS**

This work has been supported in part by the Croatian Science Foundation under the project 8658.

**REFERENCES**


Submitted: 12.01.2015. Ivana Martić
Nastia Degiuli, nastia.degiuli@fsb.hr

Accepted: 14.04.2015. Ivan Ćatipović
University of Zagreb,
Faculty of Mechanical Engineering and Naval Architecture,
Ivana Lučića 5, 10000 Zagreb, Croatia
THE PROCEDURE FOR CALCULATION OF THE OPTIMAL CARRYING CAPACITY OF PUSHED CONVOY BASED ON PARAMETERS OBTAINED BY EXPERIMENTS IN ACTUAL NAVIGATING CONDITIONS

UDC 629.55:629.561.2:629.5.013.4
Original scientific paper

Summary

Exploitation and technical parameters of a ship are very important qualitative characteristics which determine the efficiency of the use of any kind of transportation vessels both for pushboats and for pushed convoys as a whole. Some of the most important exploitation and technical parameters are: parameter of transport efficiency, tonnage quality parameter, and thrust output of propeller. The complexity of these parameters can be seen from the fact that they present the values of achieved payload-distance during transportation of cargo in the unit of time per unit of installed (or effective) power of propelling engines of ships.

The most reliable means for determining the transport efficiency is the method of testing in actual navigating conditions, which is conducted in order to determine exact technical characteristics of propelling engines of pushboats, and to determine drawbacks in their work and to increase their thrust and speed characteristics. This paper will present the process of choosing the size, shape, and number of barges in the convoy based on experiments conducted on a pushboat whose propelling engines have installed power $3 \times 809.6$ kW ($3 \times 1,100$ HP). Obtained results are based on presumption that the total resistance of a pushed convoy is equal to the total thrust achieved by the ship’s propelling complex.

Keywords: pushboat; parameter of transport efficiency; thrust; thrust output of propeller; pushed convoy; convoy; optimal speed of navigation

1. Introduction

As an objective criterion of accuracy of the theoretical calculations during determination of navigational characteristics of ships-pushboats, experiments are conducted in actual navigating conditions, especially dynamometric and torque measurements together with measurements of speed of navigation.

These experimental measurements are conducted for the highest draught of barges for different working regimes of propelling engines and different shapes of pushed convoys, as well as for stationary measurements. In order to conduct the measurements of already built
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions 

ships in actual conditions it is necessary to choose a measuring track (measuring mile) of substantial length which needs to meet the following conditions [7]:

1. that the speed of river flow is negligible;
2. that the depth in the measuring track (H) is chosen from the simultaneous fulfilment of the following equations \( \frac{H}{T} \leq 7 \) and \( \frac{v}{\sqrt{g \cdot H}} \leq 0.5 \) where \( H \) is the depth of water in measuring track (m), \( T \), draught of vessels-barges during experiments (m);
3. length of measuring track needs to be long enough for measurements to take place and it also needs to have approaching sections on both ends of the track.

The basis for this paper is the presumption that the force needed to overcome the produced resistance of a pushed convoy, called thrust \( (F_t) \), is equal to the total measured resistance \( (R_{tot}) \) shown in Figure 1.

The previously mentioned measurements are performed for the highest possible number of different pushed convoys, for example, a pushboat and one barge (P+1), or a pushboat and two barges (P+1+1), a pushboat and four barges (P+2+2), etc. The experiment ends in such manner that the pushboat works “stationary”, or when the speed of navigation is \( v=0 \), pushing against a river bank. In this case it encounters resistance equal to infinity \( R_{\infty} \), which is equal to navigation with a convoy of endless number of barges.

The thrust created by each of the propellers as a function of power on propeller shafts, the number of rotations of propeller shafts and the speed of navigation in calm water \( F_t=f(P_D; N_p; v) \) are determined while the pushboat is working with each of convoys and for each working regime of the propelling engines. The total thrust achieved by the pushboat \( \Sigma F_t \) is equal to the sum of thrusts created by all pushboat’s propellers. Based on achieved values of the total thrust it is possible to construct a diagram which shows the change \( \Sigma F_t=f(v) \) in a matching coordinate system. It is understood that during the process of thrust calculation, besides the main dimensions of the pushboat, it is necessary to know the constructive characteristics of the propeller and the nozzle and it is necessary to use corresponding propeller diagrams.

Product of multiplication of pushboat’s propeller thrust \( \Sigma F_t \) and the measured speed of navigation in relation to water \( (v) \) for each of the convoys and for all the different working regimes of the propelling engines gives thrust output, \( A_t \). Based on the calculated values a diagram of change of thrust output \( A_t=f(F_t; v) \) can be constructed in a corresponding coordinate system (Figure 2). The purpose of constructing such a diagram of the thrust output \( A_t \) is to create a curve which for one, the optimal speed, for the current technical condition of the pushboat achieves its maximal value and can be described with a mathematical function. The diagram of thrust output starts at the coordinate system origin – since it is the speed of navigation \( v=0 \), and \( A_t(v)=0 \), it ends in the point where the speed of the pushboat is at its maximum \( (v=v_{max}) \) while the pushboat is navigating without a convoy, when thrust of a convoy is \( \Sigma F_t(v)=0 \), because of which \( A_t(v)=0 \).
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions

\[ \sum F_t \cdot v \]

where \( \Sigma F_t \) is total thrust of all the propellers of one pushboat, (N).

The optimal speed of convoy is considered to be the one which achieves the highest thrust output of the pushboat. Based on this claim the following can be stated: \( v_{opt}=f(F_t) \), or \( v_{opt}=f(A_t) \).
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions

The highest thrust output is achieved in the point when the derivative of the function that presents the thrust output \( A_t \) per speed \( v \) is equalized to zero, or \( \frac{dA_t}{dv} = 0 \)

3. Basic characteristics of the K series pushboats

The pushboats of the K series include the Kumanovo, Kragujevac and Kadinjača pushboats. Their basic characteristics are shown in Table 1.

Table 1 Basic characteristics of K series pushboats [18]

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Abbreviation</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length over all</td>
<td>( L_{oal} )</td>
<td>m</td>
<td>34.32</td>
</tr>
<tr>
<td>Length at the constructive water line</td>
<td>( L_{CWL} )</td>
<td>m</td>
<td>33.20</td>
</tr>
<tr>
<td>Beam</td>
<td>( B )</td>
<td>m</td>
<td>11.00</td>
</tr>
<tr>
<td>Maximal drought</td>
<td>( T_{max} )</td>
<td>m</td>
<td>1.80</td>
</tr>
<tr>
<td>Displacement</td>
<td>( D )</td>
<td>t</td>
<td>505.50</td>
</tr>
<tr>
<td>Block coefficient</td>
<td>( \delta )</td>
<td>-</td>
<td>0.76898</td>
</tr>
<tr>
<td>Total power of propelling engines</td>
<td>( P )</td>
<td>kW</td>
<td>3×809.6</td>
</tr>
</tbody>
</table>

The characteristics of the propelling engines of the K series pushboats are the following: manufacturer: MAK; type of the engine: 8М 281 АК – four stroke; number of cylinders: 8; total displacement of the cylinders: 101.4 l; permanent power of the engine: 809.6 kW (1,100 HP) at 750 rpm; reduction rate 2.526:1.

The characteristics of the propeller-nozzle complex of the K series pushboats:
- propeller diameter \( D=1800 \text{ mm} \); number of propeller blades: \( z=3 \); expanded blade area ratio: \( a_E=0.661 \); diameter at the entrance of the nozzle: \( D_u=2030 \text{ mm} \); diameter at the exit of the nozzle: \( D_i=1900 \text{ mm} \); length of the nozzle: \( l_H=1200 \text{ mm} \); shape coefficient of the nozzle: \( \alpha_H=1.244 \); expansion coefficient of the nozzle: \( \beta_H=1.089 \); relative length of the nozzle: \( \tilde{l}_H = 0.66 \).

Names and symbols of variables, where possible, have been matched to the ITTC 2008 recommendations [15].

The pushboats of the K series, shown in Figure 3, have been designed to work with homogenous pushed convoys made of 12 barges in formation P+4+4+4 of total carrying capacity 19,200 t at a navigation speed of \( v=12 \text{ km/h} \) in calm water [13].
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions

IVAN ŠKILJAIĆA, ILIJA TANACKOV, VLADISLAV MARAŠ

4. Basic characteristics of the barges

For determination of the thrust of the K series pushboats propellers, the barges and towed units presented in Table 2 were used.

Table 2 Basic characteristics of barges and towed units [14] [18]

<table>
<thead>
<tr>
<th>Basic dimensions</th>
<th>$L_{oa}$ (m)</th>
<th>$L_{CWL}$ (m)</th>
<th>$B$ (m)</th>
<th>$T_{max}$ (m)</th>
<th>$D$ (t)</th>
<th>$\delta$</th>
<th>$Q_r$ (t)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Barge type</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>JRB71300</td>
<td>66.95</td>
<td>65.65</td>
<td>10.2</td>
<td>2.3</td>
<td>1497.56</td>
<td>0.972</td>
<td>1267.56</td>
</tr>
<tr>
<td>JRB81200</td>
<td>67.03</td>
<td>65.65</td>
<td>10.2</td>
<td>2.3</td>
<td>1454</td>
<td>0.944</td>
<td>1224</td>
</tr>
<tr>
<td>JRB81500</td>
<td>77.02</td>
<td>75.4</td>
<td>11</td>
<td>2.56</td>
<td>1991.22</td>
<td>0.9378</td>
<td>1707.89</td>
</tr>
<tr>
<td><strong>Towed units</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>01008</td>
<td>75.12</td>
<td>71.90</td>
<td>10.01</td>
<td>1.95</td>
<td>1184.00</td>
<td>0.843</td>
<td>949.11</td>
</tr>
<tr>
<td>01009</td>
<td>75.12</td>
<td>71.90</td>
<td>10.01</td>
<td>1.95</td>
<td>1184.00</td>
<td>0.843</td>
<td>955.95</td>
</tr>
<tr>
<td>19702</td>
<td>72.90</td>
<td>69.15</td>
<td>9.04</td>
<td>2.20</td>
<td>1165</td>
<td>0.847</td>
<td>915.32</td>
</tr>
<tr>
<td>26773</td>
<td>59.62</td>
<td>58</td>
<td>8.07</td>
<td>2.10</td>
<td>822.76</td>
<td>0.837</td>
<td>676.89</td>
</tr>
<tr>
<td>26808</td>
<td>57.90</td>
<td>56</td>
<td>8.20</td>
<td>2.15</td>
<td>805.2</td>
<td>0.815</td>
<td>664.75</td>
</tr>
</tbody>
</table>

The barges used in the experiments with the pushboats of the K series are asymmetrical and they are presented in Figures 4 and 5. The JRB71300 barges are used for dry bulk cargo without a lid, while the JRB81200 and JBR81500 barges are intended for transport of crude oil and its derivates.
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions

For better understanding of the problem, it is necessary to make difference between the terms ‘convoy’ and ‘pushed convoy’. A convoy is made up only of barges of the same or different types, while a pushed convoy is made up of a convoy of barges together with a pushboat. This is shown in Figure 6.

5. Procedure of determination of technical and exploitation parameters of the K series pushboat

Determination of technical and exploitation parameters of the K series pushboat is conducted under the condition of the full use of the power of propelling engines and gaining the highest value of the thrust known for the characteristics of the propeller-nozzle complex. Torsion meters (devices for measuring torque, Maihak type) and electric contact counters are placed on all three propeller shafts just in front of the propeller shaft post. Determination of thrust characteristics of pushboats is based on precise determination of the power delivered to the propeller ($P_D$) and the number of rotations per minute ($N_p$) for each propeller shaft for work with different barge convoys, together with simultaneous measurement of the speed of navigation for such pushed convoys in calm water.
The procedure for calculation of the optimal carrying capacity of a pushed convoy based on parameters obtained by experiments in actual navigating conditions

The assumption is made that the torque (used to calculate power) and the number of rotations in front of the propeller shaft post is equal to the power and number of rotations given to the propeller. It is considered that the losses in the propeller shaft post are negligible.

The measuring procedure consists of measuring the characteristics of the working regime of propelling engines for the highest number of rotations of engine shafts which achieve approximately 100% of nominal power.

Measurement of speed in relation to water is done by hydrometric wing which is positioned at approximately 5 meters in front of the front row of barges at a depth of 0.8 to 1.0 meters.

During the testing of the K series pushboat in each case the barges that formed the convoy were loaded to the maximum draught so their coefficient of the use of carrying capacity was \( \epsilon = \frac{Q_e}{Q_r} \approx 1.0 \), where \( Q_e \) is the really loaded amount of cargo in a barge and \( Q_r \) is the registered (maximal) carrying capacity of the barge. The shapes of pushed convoys for which the measurements of thrust characteristics were done are marked with letters A, B, C, D, E, F, G and H and are shown in Figure 7. Experiments with all pushed convoys were done on Djerdap lake (reservoir) where the depth of water during measurements was between 15 and 16 meters. This fact leads to the conclusion that there had not been any negative influences on the results of measurements.

![Figure 7 Shapes of pushed convoys while testing the K series](image)

Technical parameters referring to the pushed convoys A, B, C, D, E and F were determined by the Laboratory for Testing of Ships and Waterways of the Faculty of Transport and Transport Engineering, University of Belgrade, Serbia while the parameters referring to the pushed convoys G and H were obtained by research conducted by Brodarski Institute from Zagreb, Croatia.

6. Parameters measured during testing of pushed convoys

During testing of the considered pushed convoys, the number of rotations of propeller shafts \( (N_p) \) and power on propeller shafts \( (P_D) \) were measured and the values obtained are shown in Table 3.

Simultaneously with the measurements of the number of rotations and torques on propeller shafts, the speed of navigation in relation to water was measured and the obtained values are presented in Table 4.
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions

Ivan Škiljaica, Ilija Tanackov, Vladislav Maraš

Table 3 Measured numbers of rotations and power on propeller shafts [1], [6]

<table>
<thead>
<tr>
<th>Shape of pushed convoy</th>
<th>Measured number of rotations $N_p$ (rpm)</th>
<th>Measured power $P_D$ (HP)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LS</td>
<td>MS</td>
</tr>
<tr>
<td>A</td>
<td>294</td>
<td>293</td>
</tr>
<tr>
<td>B</td>
<td>295</td>
<td>300</td>
</tr>
<tr>
<td>C</td>
<td>288</td>
<td>289</td>
</tr>
<tr>
<td>D</td>
<td>298</td>
<td>294</td>
</tr>
<tr>
<td>E</td>
<td>293</td>
<td>297</td>
</tr>
<tr>
<td>F</td>
<td>295</td>
<td>299</td>
</tr>
<tr>
<td>G</td>
<td>283.5</td>
<td>294.7</td>
</tr>
<tr>
<td>H</td>
<td>280.7</td>
<td>291.3</td>
</tr>
</tbody>
</table>

Notice:
1. During calculations the power on propeller shafts given in HP was converted and presented in kW
2. LS – left shaft; MS – middle shaft; RS – right shaft

Table 4 Measured speeds of navigation in relation to water ($v$)

<table>
<thead>
<tr>
<th>Speed $v$</th>
<th>Shape of pushed convoy</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
</tr>
<tr>
<td>km/h</td>
<td>14.20</td>
</tr>
<tr>
<td>m/s</td>
<td>3.9444</td>
</tr>
</tbody>
</table>

In the procedure of measuring the pushboat parameters, the convoys whose total carrying capacity $\Sigma Q_e$ is presented in Table 5 were used [1], [6].

Table 5 Total carrying capacities ($\Sigma Q_e$) of convoys of barges

<table>
<thead>
<tr>
<th>$\Sigma Q_e$ tons</th>
<th>Shape of pushed convoy</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
</tr>
<tr>
<td>12,685</td>
<td>4,198</td>
</tr>
</tbody>
</table>

7. Presentation of working parameters of the K series pushboat

As mentioned, the thrust of each propeller built into the K series pushboat is calculated according to the methods given in the literature [3], [7], [8], [9], [10] in actual navigating conditions ($v$), ($N_p$) and ($P_D$), while taking into account characteristics conditioned by different shapes of pushed convoys. The graph of the change of total achieved thrust $\Sigma F_t$ is best presented by a curve shaped according to the following expression:

$$ F_t = f(v) = -0.0655 \cdot v^3 + 1.4852 \cdot v^2 - 10.604 \cdot v + 384.6, \ (r^2=0.9939). \quad (2) $$

While the thrust output from Equation 1 for the same shapes of convoys is presented by a curve:

$$ A_t = f(v) = -1.9208 \cdot v^3 + 45.052 \cdot v^2 + 88.765 \cdot v + 256.91, \ (r^2=0.9918). \quad (3) $$
The first derivative of the function of the thrust output by speed $\frac{dA_t}{dv} = 0$ determines the optimal speed of navigation in calm water, for the given technical state of the propelling engines and propulsion complex of the pushboat, which (in this case) equals $v = 16.72 \text{ km/h}$. Graphs of the change of total thrust $\Sigma F_t$ and the thrust output of K series pushboat are obtained based on experiments with pushed convoys having shapes marked as A, B, C, D, E, F, G and H are shown in Figure 8.

![Figure 8](image.png)

**Figure 8** Graph of the change in thrust and thrust output of the K series pushboat

It is known that thrust $F_t$ during constant movement of pushed convoys can be equalized to the total resistance of pushed convoy $R_S$, or in other words $\Sigma F_t = R_S$. Based on this assumption, the curve of the change of resistance of the pushed convoys shown in Figure 7 ($R_S$, kN) for the range of measured speeds ($v$, km/h) has the following shape:

$$R_S = f(v) = 0.160 \cdot v^3 - 8.562 \cdot v^2 + 136.3 \cdot v - 321.8, \ (r^2=0.998). \quad (4)$$

When solving tasks which relate to the choice of the pushboat and pushed convoy it is often useful to use the method of reduced resistances [7], [8], [9], according to the following equation:

$$\bar{R}_S = \frac{R_S}{v^2} \quad (5)$$

Reduced resistance can be described by the following equation:

$$\bar{R}_S = f(v) = -0.011 \cdot v^3 + 0.736 \cdot v^2 - 18.65 \cdot v + 167.3, \ (r^2=0.999). \quad (6)$$

The total resistance of the pushed convoy is usually lower than the sum of all individual resistances of all the barges within the convoy and the resistance of the pushboat, which is especially characteristic when barges form a shape which has a favourable $L/B$ ratio. Calculation of the resistance of the pushed convoy is done according to the following equation (7) [7], [8], [9]:

$$R_S = k_S \left( \sum_{i=1}^{n} R_i + R_T \right) \quad (7)$$
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions, Ivan Škiljaica, Ilija Tanackov, Vladislav Maraš

where:

$k_S$, shape coefficient of the pushed convoy;

$\sum_{i=1}^{n} R_i$, total resistance of all the barges that make the pushed convoy, N;

$n$, number of barges in the pushed convoy;

$R_i$, individual resistance of each individual barge that make the convoy, N;

$R_T$, resistance of the ship (pushboat) in the pushed convoy, N.

Based on the above stated it follows that the shape coefficient of the pushed convoy ($k_S$) is calculated according to the expression:

$$k_S = \frac{R_S}{\sum_{i=1}^{n} R_i + R_T}$$

By using the described procedure for each pushed convoy A, B, C, D, E, F, G and H the coefficient of the shape was calculated. The curve of the change of the coefficient of shape of the pushed convoy as a function of speed of navigation has the following shape:

$$k_S = f(v) = -0.0207 \cdot v^2 - 0.0825 \cdot v + 1.9493, \quad (r^2=0.9996)$$

and can be presented by a graph, as shown in Figure 9.

![Figure 9](image)

Figure 9 Graph of the change of shape coefficient of pushed convoy

Presented results are valid for the range of speeds between 11 km/h and 25 km/h, which is presented by vertical dashed lines in Figures 8 and 9.

Since there are no measurements of resistance for barges within the series JRB71300, JRB81200 and JRB81500 in actual navigating conditions, to calculate the value of coefficient of shape of the tested pushed convoys ($k_S$), the resistance of each barge was determined according to the ITTC-57 method as stated in [2], [3], [4], [7], [8], [9], [10]. In this way, the curve of change of total resistance ($R_i$) was determined for each of the stated barge types for speeds of navigation in calm water up to 6.0 m/s, together with the curve of change of their reduced resistance ($\bar{R}$) for the same speeds, Table 6.
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions

Ivan Škiljaica, Ilija Tanackov, Vladislav Maraš

Table 6 Shapes of the change of curve of resistance \((R_i)\) and reduced resistance \((\overline{R})\) for barges JRB71300, JRB81200 and JRB81500

<table>
<thead>
<tr>
<th>Barge type</th>
<th>Curve of change of barge resistance (R_i(N)) and curve of change of reduced resistance (\overline{R}(N \cdot s^2/m^2))</th>
</tr>
</thead>
<tbody>
<tr>
<td>JRB71300</td>
<td>(R_{71300} = f(v) = 1.7901 \cdot v^2 + 0.0823 \cdot v - 0.0759, (r^2=0.9997)) (\overline{R}_{71300} = f(v) = -2.22 \cdot v^3 + 27.04 \cdot v^2 - 111.4 \cdot v^2 + 175.2 \cdot v + 1723, (r^2=1.0))</td>
</tr>
<tr>
<td>JRB81200</td>
<td>(R_{81200} = f(v) = 1.9835 \cdot v^2 - 1.5969 \cdot v + 0.5592, (r^2=0.9995)) (\overline{R}_{81200} = f(v) = -7.87 \cdot v^3 + 101.9 \cdot v^2 - 371.3 \cdot v + 1980, (r^2=0.997))</td>
</tr>
<tr>
<td>JRB81500</td>
<td>(R_{81500} = f(v) = 2.045 \cdot v^2 - 0.3891 \cdot v + 0.1911, (r^2=0.9998)) (\overline{R}_{81500} = f(v) = 2.672 \cdot v^4 - 33.12 \cdot v^3 + 165.9 \cdot v^2 - 403.0 \cdot v + 2323, (r^2=1.0))</td>
</tr>
</tbody>
</table>

Based on the calculated resistance of each barge type and their total number by types which form convoys, the values of total resistances of all the barges per convoys A, B, C, D, E, F, G and H were calculated, which gives value \(\sum_{i=1}^{n} R_i\). The total resistance of the pushed convoy \((R_S)\) is gained by adding the resistance of the ship, pushboat \((R_T)\) to the previously calculated resistance of all barges in the convoy, \(R_S = \sum_{i=1}^{n} R_i + R_T\).

The resistance of the pushboat is calculated according to the known ITTC-57 method, where the speed of flow of water around the ship’s hull has to be used \((v_T)\), instead of the speed of navigation \(v\). The speed \((v_T)\) which is used for the calculation of the resistance of the pushboat is calculated according to the expression [7], [8], [9]:

\[ v_T = v \cdot (1 - \psi_S) \]  \hspace{1cm} (10)

where \(\psi_S\) is the coefficient of the return flow of water which occurs as a consequence of movement of barges in the convoy and is calculated according to the expression [7], [8], [9]:

\[ \psi_S = c \cdot \delta \cdot \sqrt{\frac{\pi}{L + l}} \]  \hspace{1cm} (11)

where,
- \(c\), coefficient, which is \(c = 0.9\);
- \(\delta\), block coefficient of barges that make the convoy; in the calculation procedure it is taken as the average value of all block coefficients of all the barges in the convoy;
- \(\Omega\), area of the main frame of barges that are tied immediately to the pushboat (the first row of barges), \(m^2\);
- \(L\), length of the stern part of the barge tied to the pushboat, \(m\);
- \(l\), distance between the propeller-nozzle complex and the stern part of the barge that are tied immediately to the pushboat, \(m\).

Function of the change of resistance of the pushboat \((R_T)\) tied to the convoys shaped as A, B, C, D, E, F, G and H according to the stated procedure is best described by the curve of the following shape:

\[ R_T = f(v) = 0.142 \cdot v^2 + 1.440 \cdot v - 0.328, (r^2=0.9998) \]  \hspace{1cm} (12)
The procedure for calculation of the optimal carrying capacity of
pushed convoy based on parameters obtained by experiments in
actual navigating conditions

It is known that on the basis of determined technical parameters the following tasks can be
solved for pushboats:
1. determination of pushed convoy navigation speed for the known working
   parameters of the pushboat and barges that make the pushed convoy;
2. choosing the convoy of barges for known parameters of the pushboat and for
given speed of navigation;
3. choosing the pushboat for known convoy for given speed of navigation.

When forming barge convoys it is important to bear in mind the following two
assumptions:
1. resistance of the pushboat is negligibly low in comparison to total resistance of all
   the barges in a convoy
2. it is not recommended that convoys are formed from barges that significantly differ
   in their shape, and also, barges that have differences in draughts that is 10% or more.

In the case when it is necessary to determine optimal carrying capacity of the pushed
convoy \( Q_{e(opt)} \), for known technical parameters of the K series pushboat and calculated optimal
speed of navigation, the procedure has the following steps:
1. Optimal speed of navigation of a pushboat with a convoy is \( v_{opt}=16.72 \text{ km/h} \) \( v_{opt}=4.644 \)
m/s;
2. Total and reduced resistance of a barge convoy at speed \( v_{opt}=16.72 \text{ km/h} \) \( v_{opt}=4.644 \)
m/s are:
   \[ R_s = f(v) = 0.160 \cdot 16.72^3 - 8.562 \cdot 16.72^2 + 136.3 \cdot 16.72 - 321.8 = 311.431.61 \text{ (N)}; \]
   \[ \bar{R} = f(v) = -0.01 \cdot 16.72^3 + 0.736 \cdot 16.72^2 - 18.65 \cdot 16.72 + 167.3 = 14,484.817 \text{ (N\cdot s^2/m^2)} \]
3. Total and reduced resistances of barges per barge types for the speed \( v_{opt}=16.72 \text{ km/h} \)
\( v_{opt}=4.644 \) m/s, by using expressions given in Table 6, are:
   \[ R_{s1300} = f(v) = 38,912.915 \text{ (N)}; \quad \bar{R}_{s1300} = f(v) = 1,809.735 \text{ (N\cdot s^2/m^2)}; \]
   \[ R_{s1200} = f(v) = 35,920.817 \text{ (N)}; \quad \bar{R}_{s1200} = f(v) = 1,665.106 \text{ (N\cdot s^2/m^2)}; \]
   \[ R_{s1500} = f(v) = 42,488.094 \text{ (N)}; \quad \bar{R}_{s1500} = f(v) = 1,955.036 \text{ (N\cdot s^2/m^2)}; \]
4. Required number of barges in a convoy per types
   \[ n_{s1300} = \frac{R_s}{R_{s1300}} = \frac{311.431.61}{38,912.915} = 8.003 \text{ or } n_{s1300} = \frac{\bar{R}_s}{\bar{R}_{s1300}} = \frac{14,484.817}{1,809.735} = 8.003 \]
   \[ n_{s1200} = \frac{R_s}{R_{s1200}} = \frac{311.431.61}{35,920.817} = 8.669 \text{ or } n_{s1200} = \frac{\bar{R}_s}{\bar{R}_{s1200}} = \frac{14,484.817}{1,665.106} = 8.669 \]
   \[ n_{s1500} = \frac{R_s}{R_{s1500}} = \frac{311.431.61}{42,488.094} = 7.329 \text{ or } n_{s1500} = \frac{\bar{R}_s}{\bar{R}_{s1500}} = \frac{14,484.817}{1,955.036} = 7.408 \]
5. Total thrust of the propellers of the K series pushboat at optimal speed of navigation
\( v_{opt}=16.72 \text{ km/h} \) is:
   \[ F_t = f(v) = -0.0655 \cdot 16.72^3 + 1.4852 \cdot 16.72^2 - 10.604 \cdot 16.72 + 384.6 = 316,340.078 \text{ (N)} \]
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions

Ivan Škiljaica, Ilija Tanackov, Vladislav Maraš

8. Conclusion

The importance of experiments on pushboats in actual navigating conditions with different barge types and different convoy shapes is unquestionable. Based on this research the real (actual) technical condition of propelling engines and propeller-nozzle complex can be determined.

Described experiments were done on a river deep enough with a low speed of water flow (water reservoir) which gave specific results which are applicable only for waterways with sufficient depths. To determine the influence of different shapes of pushed convoys it is necessary to conduct identical research in a waterway with a limited depth in which case different shapes of curves for thrust and thrust output are to be expected.

Calculations based on experiments show that the value of thrust $F_t$ almost has the identical value as the total resistance of pushed convoy $R_S$ (the difference is 1.57%), which is the reason why the claim of equality $\Sigma F_t = R_S$ can be assumed.

None of the eight pushed convoys that the measurements have been made for suits the designed shape P+4+4+4, which makes it hard to comment on the data obtained during the measurements of the working parameters of the pushboat of the K series in actual navigating conditions.

Based on the calculated number of barges per type and by comparing it to the optimal solution in optimal conditions, symmetrical pushed convoys are recommended with a favourable relation between the length and the width $L/B$, as for example P+4+4 or P+3+3+3. Such pushed convoys achieve speeds of navigation close to optimal, while at the same time their shape coefficients $k_S$ have most favourable values. From all of the tested pushed convoys the closest solution to the optimal has the convoy marked with letter A.

It can be also seen from the tested convoys that those marked with letters D, E and F are significantly worse than the optimal solution. This is because those convoys have irregular shape formed from two different barge types together with tow units, which makes it impossible to form regular shapes of convoys, which can be also seen from their shape coefficient of the pushed convoys which are: $k_S(D)=1.425; k_S(E)=1.443; k_S(F)=1.447$.

In order to determine the real optimum for the tested ship-pushboat of known characteristics it is also necessary to conduct experiments with the same barge types (since previous experiments involved different barge types) and with symmetrical barges (since previous experiments involved only asymmetrical barges. Also, it is important to use as many different convoy shapes as possible.

ACKNOWLEDGMENTS

This paper is based on research done within the following projects:

• Models of integration of transport system, project number 36024 for the period 2011÷2014, by the Ministry of Science and Technological Development of the Republic of Serbia;

• Models of sustainable development of traffic in Vojvodina, for the period 2011÷2014, by the Secretariat for Science and Technological Development of the Government of Vojvodina.
The procedure for calculation of the optimal carrying capacity of pushed convoy based on parameters obtained by experiments in actual navigating conditions

REFERENCES


Submitted: 14.11.2014. Ivan Škiljaica, MSc, shkiljaica@gmail.com (Corresponding Author)

Accepted: 18.04.2015.

Faculty of Technical Sciences, University of Novi Sad
IlijaTanackov, PhD, Faculty of Technical Sciences, University of Novi Sad
VladislavMaraš, PhD, Faculty of Transport and Traffic Engineering, University of Belgrade
AN EXPERIMENTAL AND NUMERICAL PREDICTION OF MARINE PROPELLER NOISE UNDER CAVITATING AND NON-CAVITATING CONDITIONS

UDC 629.5.024.71:6295.5035:629.5.018.15:6295.5.5.015.6
Original scientific paper

Summary

In this study, the hydrodynamics and noise prediction of a five blade marine propeller were analyzed through numerical and experimental methods under variety operational conditions. The hydrodynamics of the propeller was studied and the characteristic curves were presented in both numerical and experimental methods. Inception and development of sheet cavitation conditions are obtained in both numerical and experimental methods. The cavitation was started and developed by either increasing the propeller rotational speed in constant pressure or decreasing pressure, while the velocity was kept constant. Good agreements are observed between numerical and experimental results, qualitatively and quantitatively. The noise of the propeller was analyzed through Computational fluid dynamics (CFD) method, based on the formulation of Ffowcs Williams and Hawkins (FW-H). Similarly, the experimental results collected from hydrophones were compared with numerical simulations. Finally, the effects of reflection in cavitation tunnel were obtained by considering overall sound pressure levels in numerical and experimental results.

Key words: CFD; FW-H; Cavitation tunnel; Propeller hydrodynamics; Propeller noise

1. Introduction

Three major sources of underwater noise, produced by underwater and surface vehicles, are the machinery, propeller and the flow noise. There are four mechanisms to produce the pressure waves by the propeller [1, 2]. The cavitation noise is the major source of noise of the propeller, it should be thoroughly investigated. The low frequency noise is a result of sheet cavitation on the surfaces of the blades [2].

The experimental works of the cavitation noise have been represented for various purposes, including detecting the inception and development of sheet cavitation and predicting cavitating and non-cavitating noise levels [3]. The propeller noise measurement is
cost-effective in the water tank or free water for each full-scale. Therefore, the noise measurement of model propeller is performed using the cavitation tunnel. The net propeller noise measurements for different operating conditions in the cavitation tunnel were reported by the 18th ITTC cavitation committee (1987) [4, 5]. UKON et al., in 1987, studied the acoustic field measurement in a cavitation tunnel [6]. BARK, in 1987, investigated cavitation noise from Sydney Express propeller [7]. ZHU et al., in 1978, studied the effects of wall reflections [8]. Sharma et al., in 1990, investigated some marine propellers in cavitation tunnel [9]. Atlar et al., in 2001, investigated cavitation tunnel tests for propeller noise of a Fisheries Research Vessel (FRV) which their research was carried out in Emerson Cavitation Tunnel (ECT) [10]. In a later research, Wang et al., in 2006, studied an experimental investigation on cavitation and noise characteristics of ocean stream turbines in the ECT [11]. Emin et al., in 2012, studied an experimental study into the effect of foul release coating on the efficiency, noise and cavitation characteristics of a propeller [12]. Park et al., in 2009, studied noise source localization in a cavitation tunnel [3]. As mentioned, many experimental researches have been conducted to measure the propeller noise in the cavitation tunnel. The effects of wall reflections are important in the cavitation tunnel tests. A common way for evaluating wall reflections is to measure the propeller noise both in the cavitation tunnel and free water tests. The difference of the Sound Pressure Levels (SPLs) obtained from these tests is considered as the effects of wall reflections in the cavitation tunnel. Examples of such studies were reported by 18th ITTC cavitation committee [4]. In order to extract propeller net noise; many researches were performed by numerical simulations using FW-H equations.

Seol et al., in 2002 and 2005, presented a study on the non-cavitating and cavitating underwater propeller noise [13, 14]. They described the use of a hybrid method to predict the underwater propeller noise. Their results were presented in one operative condition and in the low frequency range. Caro et al., in 2007, presented a Computational Aero-Acoustics (CAA) formulation based on Lighthill analogy for fan noise using CFD method [15]. Jin-Ming et al., in 2012, investigated the noise of a three-blade propeller and they concluded that the overall spectrum of sound in front of the propeller hub, for same distance, is more than the propeller rotating plane [16]. Ianniello et al., in 2013, investigated noise nonlinear analysis a marine propeller base on FW-H equations [17]. PAN et al., in 2013, evaluated marine propeller noise in non-uniform flow by FW-H equation [18].

In the experimental part of this study, the hydrodynamics and noise of marine propeller are investigated in a cavitation tunnel. The effects of pressure drop and propeller rotational speed incensement are studied on the cavitation inception and extent. Also, the effect of cavitation extent was studied on the overall noise of propeller. The aim of the numerical section is to obtain the acoustic field and hydrodynamic analysis of a marine propeller in the uniform inflow. Moreover, the various parameters such as: input velocity, propeller rotational speed, and etc., are investigated to extract the conditions of the cavitation inception and were used in the experimental tests. Also, all the propeller operational curves were optimized using grid study. The flow analysis results are used as the noise source in the equations for obtaining the overall SPLs. Therefore, the results from hydrodynamics analysis are compared to and verified against the experimental findings from the cavitation tunnel. In the present study, the sheet cavitation effect on increasing the overall SPLs is investigated as the most important sound source for cavitation in low frequencies.

2. Methodology of flow and acoustic analysis

The basic equation for sound propagation is the Lighthill equation obtained from the continuity and momentum equations [19]. The FW-H is a solution developed from the
Lighthill equation. In present work, the FW-H formulation is used to extract overall SPLs in the far field by CFD method. The FW-H formulation is represented by Equation (1) [20].

\[
\frac{1}{c_0^2} \frac{\partial^2 p'}{\partial t^2} - \nabla^2 p' = \frac{1}{c_0^2} \frac{\partial^2 p'}{\partial x_i x_j} \left[ T_{ij} H(f) \right] - \frac{\partial}{\partial x_i} \left[ \left( P_{ij} n_j + \rho u_i (u_n - v_n) \right) \delta(f) \right] + \frac{\partial}{\partial t} \left( \rho_0 v_n + \rho (u_n - v_n) \right) \delta(f)
\]

(1)

The terms in the right side of Equation (1) are called quadruple, dipole and monopole sources, respectively. \( p' \) is the sound pressure at the far-field. Setting \( f = 0 \) introduces a surface that embeds the external flow effect (\( f > 0 \)), while \( c_0 \) is the far-field sound speed and \( T_{ij} \) is the Lighthill stress tensor. \( H(f) \) and \( \delta(f) \) are Heaviside and Dirac delta functions, respectively. Farassat proposed a formulation for solving the FW-H equation in time domain [21]. In Farassat formulation, the pressure field is defined by Equations (2) to (4).

\[
p'(\vec{x}, t) = p'_T(\vec{x}, t) + p'_r(\vec{x}, t)
\]

(2)

\[
4\pi p'_T(\vec{x}, t) = \int_{f=0} \left[ \frac{\rho_0 v_n}{r(1-M_r)^2} \right] \, dS
\]

(3)

\[
4\pi p'_r(\vec{x}, t) = \frac{1}{c_0} \int_{f=0} \left[ \frac{l_i r_i}{r(1-M_r)^2} \right] \, dS + \int_{f=0} \left[ \frac{l_i - l M_i}{r^2(1-M_r)^2} \right] \, dS + \frac{1}{c_0} \int_{f=0} \left[ \frac{l_i (r M_i r_i + c_0 M_i - c_0 M_i^2)}{r^2(1-M_r)^3} \right] \, dS
\]

(4)

Where \( p' \) is the acoustic pressure; \( p'_T \) and \( p'_r \) describe the acoustic pressure field resulting from thickness and loading, corresponding to the monopole and the dipole sources, e.g. blade rotation and unsteady sheet cavitation on blades are defined as monopole sources and fluctuation pressure on the blade surface is defined as a dipole source. \( r = |x(t) - y(\tau)| \) is the distance between receiver and source; \( x \) and \( t \) are the sound receiver position and time, respectively. Also, \( y \) and \( \tau \) are the source position and the time, respectively. \( M \) is the Mach number; \( M_r = M \hat{r}_i / r \) is the component of the Mach number vector in the direction of the receiver, and \( \hat{r}_i = r_i / r \) defines the unit vector in the radiation direction. \( l_i \) is the local force per unit area in direction \( i \) while \( l \) is equal to \( l_i r_i \). \( v \) is the local normal velocity of the blade surface. \( c_0 = 1500 \text{m/s} \) and \( \rho_0 = 1025 \text{kg/m}^3 \) are the sound speed and water density.

In the present study, the flow around the object is obtained to determine the source of the noise. Flow effects as blade rotation, unsteady sheet cavitation and fluctuation pressure on the blade surface are used as the input for the noise analysis. The flow field of the propeller is obtained using CFD by solution Reynolds-averaged Navier–Stokes equations (RANS equations). The FW-H acoustics model in CFD code allows you to select multiple source surfaces and receivers. In this work, the surfaces of the propeller blades are selected by
integral surfaces, $f=0$, in Equations (3) and (4). The main objective of the cavitation physical is to extract the mass fraction of vapor and liquid phases. In this study, a multi-phase method [22] is used in order to extract vapor volume fraction. Therefore, in the hydrodynamic analysis of the flow, the flow field was predicted by solving the continuity and momentum equations. The basic equation for sound propagation is the Lighthill equation which is obtained from combination between the continuity and momentum equations. The three source terms on the right-hand side of Equation (1) are the monopole, dipole and quadrupole terms which are obtained from flow field results by solving the Reynolds- Averaged Navier-Stoks (RANS). Noise prediction can be represented as the solution of the wave equation if the distribution of sources on the moving boundary (the blade surface) and in the flow field is known. Setting $f=0$ in Equation (3) and (4) describes a surface that embeds the external flow ($f>0$) effect. In this research, $f=0$ is the blade surfaces of propeller which obtained flow field, including velocity magnitude and pressure distribution, is considered on it as sound sources. Therefore, the flow around propeller is solved using RANS equations and then flow data are used as the input for FW-H equation to predict the far-field acoustics.

3. Numerical Analysis, Model Geometry and Grid Generation

In this paper, a five-blade propeller model is used which has $D=0.15\text{m}$ and $A_e/A_0=0.7$. This model was designed at the Center of Excellence in Hydrodynamics and Dynamics of Marine Vehicles (CEHDMV) and is a research model with high application at CEHDMV. Figure 1 shows several information and quantities, such as the geometries, surface grids on the blade and hub surfaces, the computational domains of the model, and the boundary conditions. The solution of the Unsteady RANS equations for utilizing the $Re$-Normalization Group (RNG) $k-\varepsilon$ turbulence model and the FW-H sound equation is performed by the CFD. The RNG $k-\varepsilon$ model is based on the standard $k-\varepsilon$ model but has many advantages [23]. Type of the grid, size of the meshes, and quality are the main contributing factors in the accuracy of numerical simulation of any geometry, since their compositions affect the convergence/divergence of the solution to a great extent. Here, convective terms are discretized using the second order accurate upwind scheme, while the velocity-pressure coupling and the overall solution procedure are based on the Semi-Implicit Method for Pressure Linked Equations-Consistent (SIMPLEC) type. The blade surface is meshed with triangles grids, while the regions around the root, tip and blade edges are meshed with smaller triangles, i.e. with sides of approximately $0.001D$. The remaining region in the domain is then filled with hexahedron cells.
We also considered zones, named rotating zones, which contained the flow around the propeller, and stationary zones which contained the flow around the moving zone. A cylindrical shape is assumed for the rotating zone, with a diameter of $1.3D$ and a length of $1.3L$, where $L$ is the length of propeller hub. The rotating zone is solved via Moving Reference Frame (MRF). The inlet is situated in $4D$ distance in the upstream, while the outlet is located at $10D$ downstream and the outer boundary is at $5D$ from the shaft axis. In order to simulate the flow around the rotating propeller where the inlet boundary is located, we had imposed the velocity components for a uniform stream with a given inflow speed. At the blade and hub surface, a wall condition had imposed, while a wall boundary condition along with constant pressure conditions are imposed at the lateral and outlet boundaries, respectively.

It is important to keep the cell thickness along the body thinner than the boundary layer. The value of coefficient $y^{+}$ was the main criterion for setting the mesh resolution. The coefficient should be in a range of $30 < y^{+} < 500$ [23, 24] in order to model properly the turbulent boundary layer and obtain correct pressure distributions on the propeller blade surfaces for the $k-\varepsilon$ model. The $y^{+}$ value along the propeller surface was around +27 to 110, see Figure 2.
At first, the number of meshes is considered 1.5 million. By this number of meshes, the vapor volume fraction was not occurred on the blade surface despite their occurrence in the experiments. Therefore, in order to observe vapor volume fraction on the blade surface, the number of cells increased from 1.5 to 3.5 million. In this condition, cavitation or vapor volume fraction happened on the blade surfaces in \( J = 0.2 \). In order to consider the grid independency, trust and torque coefficients were considered for addition three sets of grids, grid 1, 2 and 3 contained 1.5, 3.5 and 4 million meshes, respectively. In these sets of grids, the edge and tip of blades were mostly refined which is important in cavitation simulation of propeller. \( K_T \) and \( K_Q \) have been calculated and compared to experiment. Table 1 shows trust and torque coefficients in different grids for \( J = 0.4, 0.6 \) and 0.8, respectively. As seen, the total thrust and torque values increase with grid refinement from 1.5 to 3.5 grids, and tend to the experimental values. The coefficients are not changed for this model when the number of grids increased from 3.5 to 4 million meshes. Finally, the number of 3.5 million meshes selected and the results are presented for this number of meshes.

### Table 1 Comparison of grid study and experiment results

<table>
<thead>
<tr>
<th></th>
<th>( J = 0.4 )</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( K_T )</td>
<td>( K_Q )</td>
<td>Error ( K_T ) [%]</td>
<td>Error ( K_Q ) [%]</td>
<td></td>
</tr>
<tr>
<td>Experiment</td>
<td>0.3454</td>
<td>0.05298</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Grid 1</td>
<td>0.3101</td>
<td>0.04905</td>
<td>10.22</td>
<td>6.53</td>
<td></td>
</tr>
<tr>
<td>Grid 2</td>
<td>0.3396</td>
<td>0.05174</td>
<td>1.67</td>
<td>2.34</td>
<td></td>
</tr>
<tr>
<td>Grid 3</td>
<td>0.3391</td>
<td>0.05168</td>
<td>1.82</td>
<td>2.45</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>( J = 0.6 )</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( K_T )</td>
<td>( K_Q )</td>
<td>Error ( K_T ) [%]</td>
<td>Error ( K_Q ) [%]</td>
<td></td>
</tr>
<tr>
<td>Experiment</td>
<td>0.2495</td>
<td>0.04088</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Grid 1</td>
<td>0.2203</td>
<td>0.03793</td>
<td>11.70</td>
<td>7.21</td>
<td></td>
</tr>
<tr>
<td>Grid 2</td>
<td>0.2359</td>
<td>0.03897</td>
<td>5.45</td>
<td>4.67</td>
<td></td>
</tr>
<tr>
<td>Grid 3</td>
<td>0.2348</td>
<td>0.03834</td>
<td>5.89</td>
<td>6.21</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>( J = 0.8 )</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( K_T )</td>
<td>( K_Q )</td>
<td>Error ( K_T ) [%]</td>
<td>Error ( K_Q ) [%]</td>
<td></td>
</tr>
<tr>
<td>Experiment</td>
<td>0.1451</td>
<td>0.0270</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Grid 1</td>
<td>0.1359</td>
<td>0.0238</td>
<td>6.34</td>
<td>11.85</td>
<td></td>
</tr>
<tr>
<td>Grid 2</td>
<td>0.1447</td>
<td>0.0259</td>
<td>1.89</td>
<td>4.07</td>
<td></td>
</tr>
<tr>
<td>Grid 3</td>
<td>0.1447</td>
<td>0.0259</td>
<td>1.89</td>
<td>4.07</td>
<td></td>
</tr>
</tbody>
</table>

4. **Test setup and testing laboratory**

Open water tests are carried out according to the ITTC procedure using K23 cavitation tunnel. The K23 cavitation tunnel, located at the CEHDMV in Sharif University is a recirculation tunnel with a rectangular measuring section, 2300 mm long, 650 mm wide and 350 mm deep. The cavitation tunnel test section can be seen in Figure 3.
To obtain the characteristic curves in the hydrodynamics tests, the propeller rotational speed is kept constant, and the flow speed is varied in the allowable range of cavitation tunnel, at 0-3.6 m/s. In acoustic tests, the flow velocity is kept constant and the rotational speed is varied in the range of 300-1600 rpm to investigate the effects of increasing rotational speed on the propeller overall noise.

In the numerical study, cavitation and vapor volume fraction appearance is initiated in advance coefficient $J=0.2$, which is corresponding to $N=1400$ rpm and $V=0.7$ m/s. In the experimental test, the cavitation phenomenon is exactly started in the range predicted by the numerical results. The tunnel flow speed is kept constant at 0.7 m/s during cavitation inception measurements. Two procedures are generated for cavitation development conditions in the tunnel. In the first method, the flow velocity and ambient pressure are kept at 0.7 m/s and 90 kPa, respectively and then the rotational speed of propeller is increased from 1400 rpm to 1600 rpm. In the second method, the rotational speed of propeller and flow velocity are fixed in $N=1400$ rpm and $V=0.7$ m/s, respectively and then ambient pressure in tunnel is decreased from 90 kPa to 70 kPa. A Sony alpha SLT-α 33 video camera, with an appropriate electronic shutter, used for video recording and image capturing of inception and development cavitation. The qualitative comparison between the numerical and experimental results for the cavitation inception and development is presented in the results section.

The same test setup considered for noise measurements according to the 18th ITTC cavitation committee recommendations [4]. In order to measure the propeller noise in K23 cavitation tunnel, two B&k 8103 hydrophones were used. The position of hydrophones shows in Figure 4 and Table 2. The hydrophones are well fixed in positions on Plexiglas section in the cavitation tunnel.
The following steps are carried out to extract net propeller noise:

- Measuring flow noise in tunnel when dynamometer is off.
- Measuring flow and dynamometer noise when propeller is not installed in the tunnel.
- Measuring total noise in tunnel when the propeller rotates.

For the above steps, each test is repeated three times and the uncertainty measurements have been obtained ±3dB.

5. Results and discussions

5.1 Numerical and Experimental Results of the Hydrodynamics Analysis

In numerical simulation, the cavitation phenomenon was well investigated. Cavitation and vapor volume fraction appears in \( J = 0.2 \), which is equivalent to \( N = 1400 \) rpm and \( V = 0.7 \) m/s. Figure 5 presents the vapor volume fraction for two \( J = 0.2 \) and 0.17. Also, this figure shows the cavity growing on the blades of model. In this figure the vapor volume fraction on the surfaces of blades dramatically increases, when the rotational speed is increased from 1400 rpm to 1600 rpm. The results of cavitation numbers, \( \sigma \), at different points of the blade, S, are presented in Table 4.

<table>
<thead>
<tr>
<th>Hydrophone</th>
<th>( X ) (m)</th>
<th>( Y ) (m)</th>
<th>( Z ) (m)</th>
<th>( \theta ) (x-z)</th>
<th>Numerical</th>
<th>Experimental</th>
</tr>
</thead>
<tbody>
<tr>
<td>H1</td>
<td>0</td>
<td>0</td>
<td>0.225</td>
<td>90°</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>H2</td>
<td>0.25</td>
<td>0</td>
<td>0.225</td>
<td>90°</td>
<td>✓</td>
<td>-</td>
</tr>
<tr>
<td>H3</td>
<td>0.50</td>
<td>0</td>
<td>0.225</td>
<td>90°</td>
<td>✓</td>
<td>✓</td>
</tr>
<tr>
<td>H4</td>
<td>0.75</td>
<td>0</td>
<td>0</td>
<td>0°</td>
<td>✓</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 2 Coordinates of hydrophones and their applications in numerical and experimental method

![Fig. 5](image_url) The vapor volume fraction and extent of sheet cavity on the blade surfaces in numerical simulation (a) \( J = 0.2 \) and (b) \( J = 0.17 \)

Table 3 Cavitation numbers in different points of blade
Figure 6 presents the thrust, torque and efficiency coefficients for the various grids in numerical simulation and compared with experimental results. In Figure 6, a good agreement observes between the numerical and experimental results.

![Comparison of numerical simulations with experimental results for trust, torque and efficiency coefficient](image)

**Fig. 6** Comparison of numerical simulations with experimental results for trust, torque and efficiency coefficient

The experimental investigation also indicates that cavitation phenomenon is extended by the pressure drop in the cavitation tunnel. Figure 7 shows that the cavitation inception and development were for constant rotational speed, $N=1400\text{rpm}$, and flow velocity, $V=0.7\text{ m/s}$, for pressure drop of 70 and 90 kPa in two numerical and experimental methods. Cavitation value and vapor volume fraction on blade surface is found to increase with pressure drop in cavitation tunnel. As observed in Figure 7, sheet and tip cavitation increase in blade tip which is seen as pale halation in experimental tests while in the numerical simulation it has been marked by green color. It is observed that the cavity pattern well agrees with in the experiment tests.

![Qualitative comparison of the cavitation inception and development in experimental tests together with numerical simulation](image)

**Fig. 7** Qualitative comparison of the cavitation inception and development in experimental tests together with numerical simulation (a: $J=0.2$ and $p=70\text{kPa}$, b: $J=0.2$ and $p=90\text{kPa}$)
5.2 Numerical and Experimental Results of the Acoustics Analysis

5.2.1 Noise Numerical Results

The overall SPLs are calculated using the FW-H equation and by CFD which assume infinite perimeter with no reflections. Therefore, the outcomes can be considered the propeller net noise in free-field. In order to obtain the overall SPLs at different distances from the propeller’s hub for both cavitating and non-cavitating cases. Four hydrophones are used in numerical study which their positions are shown in Table 3 and Figure 3. The reference level in the solid body is considered as a source of the sound where it is the blade surfaces. Hydrophone 1 is placed in rotation plane of propeller. Figure 8 shows the overall SPLs in hydrophone 1 for \( J = 0.2 \) and 0.3. The maximum of overall SPLs observed at the blade passing frequency (BPF), see Figure 8.

![Fig. 8 The overall SPLs of H1 for \( J = 0.3 \) and 0.2](image)

The overall SPLs under non-cavitation, cavitation inception and development conditions for hydrophones 2 to 4 are presented in Figures 9 to 11. The overall SPLs include all sound sources which are obtained from the flow solution results. As seen in Figures 9 to 11 the overall SPL reduces with increasing the distance from the sound source according to the inverse square of the distance law.

![Fig. 9 The overall SPLs from numerical simulations for \( N=900 \text{rpm} \) (\( J = 0.3 \)) and \( P=90 \text{kPa} \)](image)
The propeller noise is proportionally increased when the rotational speed of propeller is increased; see Figures 12 to 14. These figures show comparison between overall SPLs for non-cavitating, cavitating inception and development states.

Fig. 10 The overall SPLs from numerical simulations for $N=1400\text{rpm (}J=0.2\text{)}$ and $p=90\text{kPa}$

Fig. 11 The overall SPLs from numerical simulations for $N=1600\text{rpm (}J=0.17\text{)}$ and $p=90\text{kPa}$

Fig. 12 The comparison of overall SPLs for H2 in $J=0.3$, 0.2 and 0.17
The difference between overall SPL in non-cavitation and cavitation inception conditions is approximately in the range of 5 to 20 dB for each frequency. This range of difference is related to the increasing of propeller rotational speed from 900 to 1400 rpm. In comparison overall SPLs under cavitation development condition with both other conditions, the difference between the overall SPL is in the range of 10 to 30 dB. This difference depict that cavitation sheet source has much contribution in the increasing of overall SPLs in low frequency. Also, the level range is confirmed by comparing with the results in [9].

As mentioned already, the FW-H equation is solved assuming infinite field and there is not reflections from surrounding environment. Therefore, numerical results in this work can be considered as net propeller noise measurement results in free-field. As a novelty of the present work, the SPL of cavitation tunnel wall reflection has been calculated. For this purpose, it is necessary to make a discussion on the results of the experimental section which led to measuring the propeller noise in the cavitation tunnel.

5.2.2 Noise Experimental Results

The model scale measurements and procedures have been submitted to the ITTC committee and reported in the Specialist Committee on Hydrodynamic Noise for the 27th ITTC [25]. The total SPLs of the propeller at wideband are first recorded at frequencies ranging from 20 Hz to 20 kHz, which included both the total noise in tunnel (propeller noise, equipment vibration, dynamometer noise and circulation noise of flow), and the noise generated by the background noise. In order to calculate the noise generated in the tunnel, the background noise is measured separately and subtracted logarithmically from the total measured noise.
In this study, the results of net propeller noise presented in 1/3 octave band for each center frequencies [26]. The measured values of SPLs in each one-third octave band to an equivalent 1Hz bandwidth by means of the correction formula as follows [10]:

\[ SPL = SPL_n - 10 \log \Delta f \]  

(5)

Where, \( SPL_n \) and \( \Delta f \) are measured SPL at each center frequency and bandwidth for each 1/3 octave band filter in 1Hz, respectively. The net SPL (\( SPL_N \)) is calculated at each center frequency using Equation (6).

\[ SPL_N = 10 \log \left[ 10^{SPL_T/10} - 10^{SPL_B/10} \right] \]  

(6)

Here \( SPL_T \) and \( SPL_B \) are total and background SPLs, respectively, measured at an equivalent 1Hz bandwidth and 1m. Time history signals are transformed to the frequency domain using Fast Fourier Transform (FFT) utility in Matlab code. Numerical method simulates only the sheet cavitation which happens on blade surfaces while in experiments, cloud cavitation and edge vortex exist in downstream and close to hydrophone 3. This effects increase the SPL in downstream region especially at hydrophone 3.

The experimental results for the SPLs for two hydrophones 1 and 3 in the constant pressure, 90 kPa, three rotational speeds \( N= 900, 1400, 1600 \) rpm and constant inlet velocity, \( V= 0.7 \) m/s, are shown in Figure 15. As seen in this Figure, the SPLs rise with the increasing in rotational speed of the propeller. Figure 16 presents the experimental results of the SPLs in two hydrophone for pressure drop from 90 to 70 kPa in \( J= 0.20 \). It is observed that SPLs gradual increase in low frequency range as pressure decreases.
Generally, vortex cavitation and small bubbles detaching from the sheet cavitation cause increasing noise level in high frequency range. However in low frequencies, large noises are due to the increasing volumes of sheet cavitation on propeller blade surfaces which act like large vibrating bubbles. Therefore, it can be concluded that fully developed cavitation is extended at pressures under 80 kPa for $J=0.2$. Of course, if cavitation development conditions happened in higher rotational speeds of propeller, then fully developed cavitation is occurred at pressure upper 80 kPa. The difference of SPLs for three assumed pressures is 3-6 dB for fully developed cavitation, in frequency range lower than 500 Hz. However, it is 4-12 dB for frequency range upper than 500 Hz, see Figure 16.

The SPLs of the received signal of hydrophone 3 are larger than those of the hydrophone 1, which are mainly because the distance between the dynamometer and the hydrophone 3 is relatively short, and the non-linear effects, e.g. cloud cavitation, occur in downstream flow close to the hydrophone 3. The SPLs obtained in this study are almost in the same range and behaviour as in the previous studies for other types of propellers [10-18].

5.3 Analysis of cavitation tunnel effects

FW-H equation is used for far-field and it does not include reflection effects of computational domain. Authors assured that numerical SPLs results of this equation can be used as free-field results which have been validated in Figures 6 and 7. But cavitation tunnel is a full reverberation environment; and wall reflections affect on the net noise results of propeller. The difference of SPLs between experimental and numerical results is introduced as effect of tunnel reflections and nonlinear terms as cloud cavitation. Zhu and et al., [8] studied the measurement of tunnel wall effects by experimental method. In present work, the tunnel wall reflection coefficient, $k(f)$, has been defined by Equation (7).

$$k(f) = 20\log\left(\frac{P_1(f)}{P_2(f)}\right)$$  

(7)
An Experimental and Numerical Prediction of Marine Propeller Noise under Cavitating and Non-Cavitating Conditions

Mohammad Reza Bagheri, Hamid Mehdigholi, Mohammad Said Seif, etc.

Where $P_d(f)$ and $P_t(f)$ are Fourier transforms of $P_d(t)$ signal in free-field and $P_t(t)$ signal in cavitation tunnel, respectively. The Equation (7) can be reviewed as Equation (8).

$$k(f) = 20\log \left( \frac{P_t(f)}{P_d(f)} \right) = 20 \left[ \log \left( \frac{P_t(f)}{P_{ref}(f)} \right) - \log \left( \frac{P_{d}(f)}{P_{ref}(f)} \right) \right] =$$

$$SPL_d - SPL_t = SPL_{Experimental} - SPL_{Numerical} \tag{8}$$

According to the results in previous sections, Figures 8, 13, 15a and 15b, and using Equation (8), $k(f)$ can be calculated in each center frequency. Figure 17 depicts the results of approximation effects of cavitation tunnel reflections. It is seen that the $k(f)$ amount in low frequency range is more than that in high frequency range which is due to the higher absorptions of sound in higher frequencies for most materials. The $k(f)$ have low amount in cavitation development conditions especially in frequency range upper than 500 Hz.

Parameter $k(f)$ can be defined as appropriate approximation for the reflection effects of cavitation tunnel walls. In fact this parameter presents the overall effects in cavitation tunnel, as cloud and vortex cavitation and reflections effects of tunnel walls. This method can be applied to extract approximate effects of wall reflections when there is not appropriate equipment for noise measurements of full-scale propeller in the free-field.
6. Conclusion

In this research, a complete parametric study investigated on pressure drop and rotational speed in order to find out cavitation inception and development and their effects on the propeller noise. Cavitation inception occurred in the range of N=1200 rpm to 1400 rpm for a five blade propeller. It is found that the pressure drop in the constant rotational speed is more effective on the cavitation inception compared to the increase in the rotational speed. Fully developed cavitation is extended at pressures under 80 kPa for J=0.2. The difference of SPLs for different pressures from 90 to 70 kPa is 3-6 dB for fully developed cavitation in frequencies lower than 500 Hz, but it is 4-12 dB for frequencies upper than 500 Hz. The difference between overall SPLs in non-cavitation and cavitation inception conditions are approximately in range 5 to 20 dB in each frequency which is related to increase of propeller rotational speed from 900 to 1400 rpm. In comparison overall SPLs under cavitation development condition with two other conditions, the difference between the overall SPL is in range 10 to 30 dB.

The difference in the overall SPL, obtained from the numerical and experimental approaches, is attributed to the reflections from the effect of tunnel. Therefore, the numerical results can be used as the results for free-field. As an important contribution of this research, the tunnel cavitation effects predicted approximately. It is seen that these effects are more effective in low frequency range than that in high frequency range. It is suggested that these effects are introduced from difference between experimental and numerical SPLs. Of interesting results that of comparison between numerical and experimental results are observed, this subject is that with increasing rotational speed of propeller and fully cavitation development on blade surface in each frequency, reflection coefficient is decreased.

REFERENCES

An Experimental and Numerical Prediction of Marine Propeller Noise under Cavitating and Non-Cavitating

Mohammad Reza Bagheri, Hamid Mehdigholi
Mohammad Said Seif, etc.


EXPERIMENTAL STUDY OF TWO LARGE-SCALE MODELS’
SEAKEEPING PERFORMANCE IN COASTAL WAVES

UDC 629.5.018.71:629.5.017.2
Original scientific paper

Summary

Actual sea waves and vessel motion are an unsteady nonlinear random process. The currently adopted test to simulate wave impact of vessel models in tank can't fully reveal the impact of real sea waves on vessel swing motion. In this paper the buoy wave height meter is adopted to carry out measurements and analyses of the coastal wave environment. The correlation between the coastal wave spectra and the ocean wave spectra is analyzed. The test system is established for remote control and telemetry self-propelled vessel models suitable for the experiment conducted in the coastal areas. The seakeeping performance test is conducted for the same tonnage of round bilge vessel model and the deep-V hybrid monohull of large-scale vessel model under the coastal wave conditions. The experimental results are compared with the test results of small-scale vessel model in the towing tank. The experimental results show that the seakeeping performance of the deep-V hybrid monohull is improved by a wide margin in contrast to that of the round bilge model, and there is a marked difference between the motion characteristics of large-scale vessel models in the coastal wave environment and that of small-scale vessel models in tank.

Key words: Large-scale model test; seakeeping performance; remote control and telemetry; coastal waves

1. Introduction

It has already been a tendency to conduct the physical simulation test for large-scale vessel models in natural environment, and the research work in this field is rather sophisticated. This test has been introduced as a new item of testing technology in the reports of ITTC (2008, 2012) in the last few years. Many researchers carried out the simulation tests of the large-scale vessel model or pilot boat for some newly-developed vessels. Loukakis (2005) gave an introduction of large scale model test at sea during the Proceeding of ITTC 2005. Sun (2010) built up the large-scale model system and studied the seakeeping performance of the model at sea. Jacobi and Thomas (2014) studied the slamming behaviour of large high-speed catamarans through full-scale measurements. Suebyiw (2013) studied the application of radio-controlled model testing in coastal waves for the design of a high-speed craft. Coraddu (2013) studied twin screw ships' asymmetric propeller behaviour by means of free running model tests in a lake.
Shu-zheng Sun, Hui-long Ren, Xiao-dong Zhao, Ji-de Li

Experimental Study of Two Large-scale Models' Seakeeping Performance in Coastal Waves

The research content of this paper focuses mainly on the seakeeping performance test. Because it is conducted under the natural sea conditions, the sea wave conditions encountered are three-dimensional non-linear actual sea waves, and the model size is no longer subject to the impact of the testing ground. Thereby, the influence of the model size effect can be reduced greatly, and the motion response of the model can better approach the non-linear motion response of a real vessel under the real sea wave conditions.

According to the model size and the requirement of the model test in the actual sea wave conditions, this paper chooses to conduct the test study under the coastal stormy wave conditions. First of all, measurements and analyses of the sea wave environment are conducted in different coastal areas in China to understand the basic characteristics and rules of the coastal wave environment, which lay a foundation for proceeding with the test study on the motion property of large-scale vessel models. This paper has established a testing system of remote control and telemetry self-propelled model.

Before the study of large-scale model test the seakeeping performance experiment of two small-scale models in towing tank had been carried out. The sections of the two models are given in Figure 1, they are a round bilge hull and a deep-V hybrid monohull which are close in the principal dimensions and displacement. The hybrid monohull has a built-up appendage fixed under the bow which can reduce the longitudinal motion amplitude of the hull. The results of model test in tank indicated that the seakeeping performance of hybrid monohull was better than that of round bilge monohull. For the validation of seakeeping performance especially nonlinear hydrodynamic performance of hybrid monohull in real ocean waves, we started the experimental study of the two large-scale models’ seakeeping performance in coastal waves. Besides, the testing results of large-scale models were compared with that of the small-scale models in towing tank to analyse the difference between the two testing methods.

2. Experimental procedures

2.1 Measurement and analysis of the coastal wave conditions

To decide whether the large-scale model test under the actual coastal wave conditions is viable or not, one of the key techniques is whether the testing conditions match those of the ocean wave conditions. For this purpose, the research team goes to Huludao Harbour sea area, Xiaoheishi sea area and Xiaoping Island sea area in Dalian respectively to investigate and measure the sea wave conditions in these three coastal areas. The wave data are measured by a buoy wave height meter (see Figure 2). The shape of the buoy is spherical, and the bottom is fitted with the iron chain to lower the center of gravity of the wave height meter. The lower part of the ball is immersed in water, and there is a slopping flap in the middle part of the ball to reduce the swing motion of the buoy. There is an acceleration sensor fitted near the center of gravity within the buoy ball to measure the acceleration time duration of wave surface heaving.
Experimental Study of Two Large-scale Models' Shu-zheng Sun, Hui-long Ren, Seakeeping Performance in Coastal Waves Xiao-dong Zhao, Ji-de Li

The time interval for collecting data is 0.05s, and each collecting time lasts about 10 minutes. See Figure 3 for the actually recorded acceleration time duration of wave surface heaving. By solving the equation, the correlation function of the acceleration time duration is derived:

\[ R(\tau) = \lim_{T \to \infty} \frac{1}{T} \int_{0}^{T} \zeta(t) \zeta(t+\tau) dt \]  

(1)

In the formula: \( \tau \) stands for the time interval, and \( T \) stands for the limited time of enough length. Then carry out Fourier transform of the above formula, which is further processed by "Hamming smoothing", from which spectral density function of wave surface heaving acceleration is derived:

\[ S_\zeta(\omega) = \frac{2}{\pi} \int_{0}^{\infty} R(\tau) \cos \omega \tau d\tau \]  

(2)

The wave surface heaving spectra are derived by two quadratures of the acceleration spectra:

\[ S_\zeta(\omega) = S_\zeta(\omega) / \omega^4 \]  

(3)

By computing the area beneath the heaving spectra curve of the wave surface \( m_0 \) (namely, the variance of the wave surface heaving), we can further obtain parameters such as the significant wave height of the wave (\( H_{1/3} \)) and mean period from spectral analysis (\( T_s \)).

The measurements and analyses results of sea waves at different tidal periods and with different wind directions at Huludao sea area and Xiaoping Island sea area in Bohai Sea are performed. The measuring points are all over 1 nautical mile off the coasts and the water depth at the measuring waters is about 10 meters. The by measurements and analyses, it is found that there is relatively great influence of the tide and wind direction on the sea wave. During the flow phase period and the high-tide period, when the wind blows from the sea to the coast, the wave spectra of the measured waters are basically similar to the ocean wave spectra.

Figure 4 shows the wave spectra of Huludao sea area at the high water period. On the very day when the test is carried out at Huludao Harbour (40° 43′ N, 121° 00′ E), the parameters of the tide, wind and water depth, and etc., are as follows:

The time of tide: 01:05 with a height of 76cm; 9:39, 286cm; 16:02, 80cm. The tidal datum plane is 158cm beneath the average sea level, in the time zone: GMT+8;

There is a southerly wind above the sea, with a wind speed of about 5m/s; and a water depth of about 8 meters in the test sea area.

The significant wave heights (\( H_{1/3} \)) of the three groups of wave data recorded are respectively 0.14m, 0.153m and 0.161m.
Experimental Study of Two Large-scale Models' Seakeeping Performance in Coastal Waves

Figure 4 Wave spectra of Huludao sea area

Figure 5 indicates the wave spectra of the coastal flow phase at Xiaoping Island sea area in Dalian City. On the very day when the test is carried out at Xiaoping Island (38° 49′ N, 121° 12′ E), the parameters of the tide, wind and water depth, and etc., are as follows:

The time of tide: 6:33 with a height of 51 cm; 11:57, 323 cm; 18:33, 46 cm. The tidal datum plane is 163 cm beneath the average sea level, in the time zone: GMT+8; with a southerly wind above the sea, and a wind speed of about 4 m/s; and with a water depth of about 10 meters in the tested sea area waters.

The statistical significant wave heights (H$_{1/3}$) are respectively 0.279 m, 0.264 m, and 0.251 m.

Comparison of the measured wave spectra with the ocean wave spectra is given in Figure 6.

There are usually phenomena of double peaks and multi-peaks for the sea wave spectra during the period of falling tide to that of low tide. Through nondimensional comparison of the off-shore sea wave spectra with the ocean wave spectra, it is seen that the main features of the sea wave spectra during the period of flow phase and that of high-tide are similar to those of the ocean wave spectra in that the central frequencies are basically the same, and the frequency scope and energy distribution are rather wide, closer to those of the actual ocean wave spectra. However, the wave spectra of Huludao sea area is more similar to ITTC Single Spectrum, and the wave spectra of Xiaoping Island is more similar to China Sea Spectrum.
2.2 Establishment of Testing System for Large-scale Vessel Model

For the seakeeping performance test of the large-scale model under the actual wave conditions, the model is the remote control self-propelled one, and the wave is measured by the buoy wave height meter. The equipment and respective characteristics are shown in Table 1. The roll and pitch motion were measured by angular rate top, the course angle of ship model was measured by course top, the vertical acceleration of model bow 1 section, middle mid-ship position and tail 19 section were measured by acceleration sensor; the arrangement of each device in the model is shown in Figure 7.

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Type</th>
<th>Range</th>
<th>Precision</th>
<th>Motions</th>
<th>Sampling Frequency/Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Angular rate top</td>
<td>MEMSIC VG440</td>
<td>Roll: ±45° Pitch: ±30°</td>
<td>±0.015°</td>
<td>Roll, Pitch</td>
<td>20</td>
</tr>
<tr>
<td>Course top</td>
<td>KVH C100</td>
<td>±180°</td>
<td>±0.05°</td>
<td>Course angle</td>
<td>20</td>
</tr>
<tr>
<td>Acceleration sensor</td>
<td>Kister 8315</td>
<td>±5g</td>
<td>±0.1%</td>
<td>Vertical acceleration</td>
<td>20</td>
</tr>
</tbody>
</table>

The main parameters of round bilge vessel of four thousand tons and the deep-V hybrid monohull models, with a scale ratio as 1:19, are shown in Table 2. The models have a remote-controlled distance of more than 1km, using 10 storage batteries to provide power. The capability of one battery is 40Ah, and the voltage is 12V. Since the power of the electromotor is 1KW, the batteries could make the model running for more than 3 hours continuously. Model test data was collected and recorded by internal inner test system and external telemetering system at the same time, and a video system installed in the model is used to observe model bow motion and green water condition. Ship model was made of glass steel, and Figure 8 shows pictures of round bilge vessel model and deep-V hybrid monohull model respectively. The model is propelled by screw propellers, with twin rudder to control course, as shown in Figure 9.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>LWL</th>
<th>BWL</th>
<th>T</th>
<th>CB</th>
<th>CG</th>
<th>D</th>
<th>Ixx</th>
<th>Iyy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Units</td>
<td>(m)</td>
<td>(m)</td>
<td>(m)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Round bilge monohull</td>
<td>6.6</td>
<td>0.75</td>
<td>0.22</td>
<td>0.45</td>
<td>0.33</td>
<td>490</td>
<td>1325.53</td>
<td>33.76</td>
</tr>
<tr>
<td>Hybrid monohull</td>
<td>6.6</td>
<td>0.8</td>
<td>0.23</td>
<td>0.41</td>
<td>0.33</td>
<td>498</td>
<td>1347.17</td>
<td>39.04</td>
</tr>
</tbody>
</table>

Tab.2 Parameters of large scale models

Fig.8 Large scale models
The debugging test of the large-scale model testing system is carried out in calm water in the Songhua River, as shown in Figure 10. Dynamical and static force of the model are adjusted, power system is debugged, and remote control and telemetering system, video system, data inner test system etc. are tested for the model. By benchmarking method, the relationship between model's navigational speed and main engine's rotate speed is measured. The test range pole is set in the shore-side when speed is measured, with the separation distance as 200m. The debugging model with direct route condition entered into velocity measurement area, so that the relationship between navigational speed and rotate speed is measured, the ship model's navigational speeds under seven main engines' rotate speeds are all measured, the measurement results are matched and the relationship curve between the main engine's rotate speed and ship model's navigational speed is obtained, as shown in Figure 11.

2.3 Seakeeping performance Test of Large-scale Ship Model

The two models are tested respectively, for totally 4 days. During the test period, internal wind direction blew from sea surface to coastal side, and tests were conducted between rising tide period and high tide period. Significant wave height compared with actual ship was Class 6 sea condition in the test area after wave height instrument's measurement.
Experimental Study of Two Large-scale Models' Seakeeping Performance in Coastal Waves Shu-zheng Sun, Hui-long Ren, Xiao-dong Zhao, Ji-de Li

Because near-shore wave was not stable and had strong randomness, test area and wave height instrument should keep proper distance and within 100m of distance to ship model to ensure the motion response of model ship and wave motivation have real-time synchronization, to make wave's data and data measured by wave height instrument coincide when model sailing, and to easily test model's motion of different course angles in the wave in the process of seakeeping performance test. The two models are tested respectively for their motions in different navigational speeds, different wave directions under the equivalently actual ship class 6 sea state ($H_{1/3}=5~6m$), as shown in Figures 12 and 13 which were test video screenshots.

3. Results and discussion

3.1 Test Data Processing for Large Scale Model Test

The test data were measured and recorded by model's internal inner test system and telemetering system in the measurement ship at the same time. This paper adopts correlation function method as described above, in which correlation function is calculated by model motion data, which then is processed by Fourier transform, and the swing angular velocity spectrum of model and fixed-point vertical acceleration spectrum can be obtained, in which the swing angular spectrum can be got after angular velocity via one time integral.

Figure 14 shows the time history of pitch and roll angular velocity of large scale test in actual sea wave environment of the round bilge vessel model; Figure 15 shows the spectrum of pitch, roll and bow vertical acceleration of large scale models for speed of 18 knots and 24 knots, whose motion significant value is shown in Table 3.
Experimental Study of Two Large-scale Models’ Seakeeping Performance in Coastal Waves

Fig. 14 Test data of large-scale model test on the sea

Tab. 3 The significant motion amplitude of large scale model test on 6th class wave ($H_{1/3} = 5m$)

<table>
<thead>
<tr>
<th>Ships</th>
<th>Speed (kn)</th>
<th>$\theta_{1/3}$ (deg)</th>
<th>$A_{bow1/3}$ (m/s²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Round bilge monohull</td>
<td>18</td>
<td>1.547</td>
<td>2.535</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>1.6306</td>
<td>3.649</td>
</tr>
<tr>
<td>Hybrid monohull</td>
<td>18</td>
<td>1.154</td>
<td>1.825</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>1.18</td>
<td>2.35</td>
</tr>
</tbody>
</table>
Experimental Study of Two Large-scale Models’ Seakeeping Performance in Coastal Waves

Comparing the motions of hybrid hull model and round bilge vessel model under Class 6 sea condition: for 18 knots, the peak value of pitching angle decreases by 30.02%, and the peak value of head accelerated speed deduces by 34.69%; while for 24 kn, the peak value of pitching angle decreases by 29.52%, and the peak value of head accelerated speed deduces by 44.13%. For 18 knots, the significant value of pitching decreases by 25.4%, and the significant value of head accelerated speed reduces by 28%; while for 24 kn, the significant value of pitching decreases by 27.6%, and the significant value of head accelerated speed reduces by 35.6%.

It is seen from the results of large scale model test that the seakeeping performance of deep-V hybrid hull has been improved greatly as compared with that of round bilge vessel model.

3.2 Comparison of Test Results with Tank Model

Finally the results of large-scale model are compared against result of tank small-scale model (scale ratio 1: 40). The test was conducted in the model ship towing tank of Harbin Engineering University (HEU), and the size of towing tank is 108*7*3.5m. The speed of the tow truck is 0.1~6.5m/s. Wave was made by the rocker flap wave generator, and the model motion of irregular wave height class 6 sea state (significant wave height H_{1/3}=5m) was measured by 4DOF seakeeping instrument, the fixed-point acceleration of model was measured by acceleration sensor, and the sensor's position was the same as that in the large-scale model. The parameters of the measurement equipment are shown in Table 4 and Figure16.

<table>
<thead>
<tr>
<th>Equipment</th>
<th>Range</th>
<th>Precision</th>
<th>Motions</th>
<th>Sampling Frequency/Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wave generator</td>
<td>T: 0.4~4s</td>
<td>±0.05°</td>
<td>Course angle</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>H_{max}: 0.4m</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ultrasonic wave height meter</td>
<td>±200mm</td>
<td>±0.1%</td>
<td>-</td>
<td>20</td>
</tr>
<tr>
<td>4DOF seakeeping instrument</td>
<td>Heave: 0~±200mm</td>
<td>±0.1%</td>
<td>Heave, Pitch, Roll</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>Pitch angle: 0~±15°</td>
<td>±0.1%</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Roll angle: 0~±50°</td>
<td>±0.1%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Acceleration sensor</td>
<td>±5g</td>
<td>±0.1%</td>
<td>Vertical acceleration</td>
<td>20</td>
</tr>
</tbody>
</table>

The parameters of small-scale models are shown in Table 5. Figure 17 shows the seakeeping performance in irregular waves of hybrid monohull. Figure 18 shows the time history of pitch amplitude in head seas in tank test. Figure 18 shows the spectrum of pitch and bow vertical acceleration of small-scale models in tank test for speed of 18 knots and 24 knots of actual ship under Class 6 sea condition.
Tab.5 Parameters of small-scale models

<table>
<thead>
<tr>
<th>Parameters</th>
<th>$L_{WL}$</th>
<th>$B_{WL}$</th>
<th>$T$</th>
<th>$CB$</th>
<th>$CG$</th>
<th>$D$</th>
<th>$I_{xx}$</th>
<th>$I_{yy}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Units</td>
<td>(m)</td>
<td>(m)</td>
<td>(m)</td>
<td>-</td>
<td>(m)</td>
<td>(kg)</td>
<td>(kgm$^2$)</td>
<td>(kgm$^2$)</td>
</tr>
<tr>
<td>Round bilge monohull</td>
<td>3.125</td>
<td>0.356</td>
<td>0.105</td>
<td>0.45</td>
<td>0.157</td>
<td>52.51</td>
<td>32.05</td>
<td>0.816</td>
</tr>
<tr>
<td>Hybrid monohull</td>
<td>3.125</td>
<td>0.381</td>
<td>0.113</td>
<td>0.41</td>
<td>0.157</td>
<td>53.37</td>
<td>32.57</td>
<td>0.944</td>
</tr>
</tbody>
</table>

Fig.17 Picture of seakeeping performance in irregular waves of hybrid monohull

Fig.18 Time history of pitch amplitude in head seas in tank test

a. Pitch spectrum in tank at 18kn
b. Bow acceleration spectrum in tank at 18kn

c. Pitch spectrum in tank at 24kn
d. Bow acceleration spectrum in tank at 24kn

Fig.19 Spectrum results of model test in tank
In this paper, nondimensional comparison is conducted between the test result of tank model and test result of large scale model, and Figure 20 and Figure 21 show the comparisons on pitching scale between the test result of tank model and test result of large-scale model under Class 6 sea condition ($H_{1/3}=5m$).

Because the actual wave environment is three dimensional and the large-scale model is in the free sailing status when navigating, which is the same as the actual ship; therefore, six coupled degrees of freedom of motions are generated. Compared with tank model test, large scale model test adds the pitching motion, whose six coupled degrees of free motions may be close to that of the actual ship. With dispersion of energy in each free action, the response value of pitching motion and head accelerated speed would appear to be low, which reflects the difference between large scale model test and tank test. However, the wave of vessel model tank is two-dimensional long-crested wave, and the model is sailed under restrictions, which limits the head and pitching motion of vessel model. As a result, the energy is concentrated on certain free motions, and the motion characteristics thereof are inevitably different from those of large scale model test.

Besides, the difference on green water frequency and wave physical phenomenon are also relatively great. From the model test in towing tank we find that the bow deck of hybrid monohull is too narrow to make the wave can climb up to the deck easily. So there have been observed many green water events. In order to reduce green water events of hybrid monohull the bow deck of large-scale model was broadened, as shown in Figure 22, and it became much less than that of small-scale model in tank. However, the two models of round bilge hull are totally the same, so we can see the difference of green water between large-scale model test and small-scale model test from the results of round bilge monohull. Table 6 shows the comparison of green water frequency between tank model test and large-scale model test of actual wave environment under Class 6 sea condition ($H_{1/3}=5m$), while Figure 23 shows the printscreen of green water video of large scale model test and Figure 24 shows the printscreen of green water video of small-scale model test in tank.
Experimental Study of Two Large-scale Models' Seakeeping Performance in Coastal Waves

Fig. 22 Sketch of the difference between the two models’ decks of hybrid monohull

Tab. 6 The comparison of green water frequency

<table>
<thead>
<tr>
<th>Program</th>
<th>Speed/kn</th>
<th>Frequency/Times per minute</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Round bilge monohull</td>
</tr>
<tr>
<td>Small-scale model test in tank</td>
<td>18</td>
<td>3.34</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>10.62</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>12.18</td>
</tr>
<tr>
<td>Large-scale model test at sea</td>
<td>18</td>
<td>2.6</td>
</tr>
<tr>
<td></td>
<td>24</td>
<td>6.1</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>9.9</td>
</tr>
</tbody>
</table>

Fig. 23 Picture of green water of large-scale model recorded by videos

Fig. 24 Picture of green water of small-scale model in tank recorded by video

From the above green water frequency and video, it is found that as compared with the small-scale model test in tank, the coupled motion characteristics of large-scale model under three-dimensional wind waving is more obvious. Therefore, its wave frequency is low and its wave physics phenomenon is closer to that of the actual vessel, which has more obvious nonlinear characteristics.
4. Conclusion

The following conclusions could be drawn by the above researches:

(1) By analyzing the measurement of coastal waves in several sea waters, we can find that the coastal wave parameter has relationships with tide, wind direction, water depth. When the wave floods to its climax with the wind blowing from the sea surface to the coastal, and the water depth is more than 8m, the wave spectrum shape measured is basically similar to the ocean spectrum, and wave height frequency density complies with the Rayleigh distribution generally. The test system for large-scale remote control and telemetry self-propelled vessel models under actual wave condition established in this paper is stable in operation and reliable in measuring data.

(2) By comparing the large-scale model test result of seakeeping performance property for round bilge vessel model and hybrid monohull model, we find that in coastal waves the seakeeping performance of hybrid monohull has also been improved dramatically compared with that of round bilge vessel. By comparing and analyzing the data collected in the coastal large-scale model test and tank small-scale model test, we find that the response amplitude of pitching motion and bow acceleration is relatively low and its green water frequency is smaller, which shows the difference between large-scale model test in coastal and model test in tank.

(3) For the motions like yaw and sway of small-scale models in tank are restricted, the energy would be concentrated on the other 4 degrees of freedom, while the motions of large-scale models in coastal have 6 degrees of freedom. This is a source of the difference between large-scale model test in coastal and model test in tank. Besides, the different testing environment and measurement instruments can also cause the difference between the two testing methods. However, the major part of the differences maybe due to uncertainty of measurements, especially in case of large scale models. So the measurement uncertainty of large-scale model test should be studied in the future work.

Acknowledgment

The author thanks the anonymous reviewers for their valuable remarks and comments. This work is supported by National Natural Science Fund of China (Grant No. 51209054), Basic Research Foundation of HEU(HEUCFR1201).

REFERENCES

Shu-zheng Sun, Hui-long Ren, Xiao-dong Zhao, Ji-de Li  
Experimental Study of Two Large-scale Models' Seakeeping Performance in Coastal Waves


Submitted: 20.11.2014.  Shu-zheng Sun, E-mail: sunshuzheng@hrbeu.edu.cn
Hui-long Ren
Accepted: 21.05.2015.  Xiao-dong Zhao
Ji-de Li
College of Shipbuilding Engineering, Harbin Engineering University
Nantong Street No.145, Harbin City, China
The need for green energy sources without or with low emissions in addition to improve the using efficiency of current fossil fuels in the marine field makes it important to replace or improve current fossil-fuelled engines. The replacement process should work on narrowing the gap between the most scientific innovative clean energy technologies and the concepts of feasibility and cost-effective solutions. Early expectations of very low emissions and relatively high efficiencies have been met in marine power plants using fuel cell. In this study, 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. The benefits of these two different reforming options can be assessed using computer predictions incorporating chemical flow sheeting software. 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.

Key words: Solid oxide fuel cell; Polymer electrolyte membrane fuel cell; reforming options, marine applications.

1. Introduction

Use of the conventional marine fuels as a source of energy for traditional marine power plant have been faced a lot of barriers due to its demerits especially regarding to the environmental and economic point of views. Environmentally, it was shown by [1] that in 2000, 15% of all global NOx emissions and 4-9% of global SO2 emissions have been emitted around by ocean-going ships. In addition, IMO revealed in 2009 that ships had emitted about 25 million tons of NOx during the year 2007. To eliminate the harmful NOx emissions; the IMO regulations had planned to achieve NOx emissions reduction through three Tiers which could require reduction in nitrogen oxides by 85% onboard ships [2]. Regarding eliminating of dangerous SOx emissions, IMO planed using of marine fuel containing 0.1% sulfur onboard ships beginning by January 2015[3, 4, and 5]. In addition economically, the prices of the conventional marine fuels showed a great fluctuation through the past few years and still present the main part of ship operating budget. To rise above, the shipping industry started search for alternative fuels that are also price competitive comparing to typical marine fuels.
like marine diesel oil (MDO). After over 20 years of preparatory steps, Liquefied Natural Gas (LNG) can be used as an alternative fuel in the shipping industry [6]. LNG-propelled ships will be particularly attractive in case the vessel will enter Emission Control Areas (ECAs) since they can meet Tier III emission levels and the SO\textsubscript{X} requirements without any treatment of the exhaust gas. It is estimated that almost 70% of the world fleet will be entering ECAs in the near future [7, 8].

Some guidelines were set for installing of fuel-cell systems (FC systems) permanently on ships and boats in different classification societies rules [9]. It described the technical requirements for the safe operation of FC systems. One of these guidelines belong to Det Norske Veritas (DNV) and Germanischer Lloyd (G.L) classification society, which revealed some consideration that should be taken into account regarding safety functions, installation, fuel transfer system, fuel storage, fuel conditioning, and fuel distribution [10, 11]. Lloyd’s Register in its guidelines [9] showed the importance of study safety of the ship, safety of personnel, and safety of machinery in case of use fuel cell onboard ships to be comply with standards of Safety of Live at Sea ( SOLAS) convention. There are five main types of fuel cells being developed in the stationary fuel cell market. These cells are referred to the type of electrolyte used within the system. Two of those are considered candidates for shipboard use, the Solid Oxide Fuel Cell (SOFC) and the Proton Exchange Membrane fuel cell (PEM) as they are available in the market size, most of materials used in their manufacturing are available, and the development of their efficiency is high [12, 13].

Steam Reforming (SREF), Partial Oxidation (POX), Auto-thermal Reforming (ATR), SOFC are the four major hydrocarbon-reforming technologies for PEM fuel cells. Steam reforming and auto thermal reforming appear to be the most competitive fuel processing options in terms of fuel processing efficiencies. Partial Oxidation (POX) method has a less fuel processing efficiency than steam reforming method [14, 15]. The primary purpose of this paper is to identify favorable operating conditions at which natural gas fuel is converted to hydrogen rich gas mixtures via SREF and SOFC internal reforming processes at reasonable fuel reforming efficiencies to be used in PEMFCs. Natural gas is selected for hydrogen rich gas production for PEMFC due to its favorable composition from lower molecular weight compounds compared with other fossil fuels. In addition, a simulation study of the different components of the selected two reforming options for PEM fuel cell will be conducted. Also, some of the performance parameters of the total system operated with natural gas fuel will be discussed as a proposed marine power plant for ships.

2. Simulation steps for PEM fuel cell

PEM fuel cells require a high purity hydrogen source for operation. Hence, the projected commercialization of PEMFC powered ships requires a readily available hydrogen source, which is either used directly or is produced in an on-board fuel processor. Hydrogen can be produced by reforming a hydrocarbon fuel into a hydrogen rich gas gases. The reformed fuel often contains other gases such as carbon monoxide (CO) that are detrimental to PEMFC operation. The CO contained in the reformat must be further reduced to capacity 10 ppm prior to feeding to the PEM fuel cell [16, 17, and 18]. The investigated PEM fuel cell system consists of three sections and their components as shown in Fig. 1. The first section is fuel processing and clean-up section. It includes steam reforming or SOFC with internal reforming (two cases investigated), high and low temperature shift reactors (HTS and LTS), and preferential oxidation reactor (PROX). The second section is PEM fuel cell which contains fuel cell stack and DC/AC converter. The third section is the auxiliary units like pumps, compressor, expander, heat exchangers, heaters, coolers, and burner. For all sections, all reactors are simulated to operate under equilibrium conditions.
Steam and SOFC based reforming options of PEM fuel cells for marine applications

SOFC based reforming option for PEM fuel cell is a combined cycle in which both SOFC and PEM fuel cell are combined. The system can be shown in Fig. 2 and it works as follows: the internal reforming SOFC is run under conditions that give low fuel utilization. This enables a high power output for a relatively low stack size. Unused fuel from SOFC appears in the anode exhaust where it undergoes shift reaction, followed by a process stage when the final traces of carbon monoxide are removed. At this stage, the gas comprises mainly hydrogen and carbon dioxide, with some steam. This gas, once it is cooled, is suitable for use as a fuel in the PEM stack [19, 20, 21, and 22].

The main difference in SOFC stack cost compared to PEMFC cost relates to the simpler system configuration of the SOFC system. This is mainly due to the fact that SOFC stacks do not contain the high-cost precious metals that PEFCs contain. In addition, the cost of SOFC balance of plant is low by comparison to the PEMFC [19]. However, the cost of the recuperating heat exchangers partially offsets that. This is offset in part by the relatively complex manufacturing process required for the SOFC electrolyte plates and by the lower power density in SOFC systems [23, 24].

2.1 Brief description of fuel cell processing and clean up sections

The fuel processing efficiency covers the section from the hydrocarbon feed section to the fuel cell including all reforming and clean-up reactors and auxiliary equipment.
The pressure is kept constant at 3 bars in this study. The operating parameters of reactors are changed parametrically to determine the best operating conditions. The limitations set by the catalysts and hydrocarbons involved are also considered. The simulation code is capable to calculate the steady state product compositions taking into account the incoming stream compositions under the defined operation conditions.

The aim is to convert as much hydrogen in the fuel into hydrogen gas at acceptable yields in an efficient manner while decreasing CO and CH₄ formation. Lower SC ratios favor soot and coke formation, which is not desired in catalytic steam and auto thermal reforming processes.

2.2 Chemical reaction scheme

The fuel processor is simplified to a steam reformer or SOFC reactor, two water gas shift reactors and a preferential oxidation reactor for the modeling purpose. Steam reforming is a method of hydrogen production used on a large scale industrially, most notably in the production of ammonia. Steam reforming involves both the reforming reaction Eqs. (1) and (2) and the water-gas shift reaction Eq. (3). These are carried out at elevated temperatures over a supported nickel catalyst.

\[
\begin{align*}
\text{CH}_4 + \text{H}_2\text{O} & \rightarrow \text{CO} + 3\text{H}_2 \\
\text{CH}_4 + 2\text{H}_2\text{O} & \rightarrow \text{CO}_2 + 4\text{H}_2 \\
\text{CO} + \text{H}_2\text{O} & \leftrightarrow \text{CO}_2 + \text{H}_2
\end{align*}
\]

In case of SOFC internal reforming (SOFC-IR), the electrolyte, which divides the stack into two electrodes, acts as an electronic barrier and avoids the direct chemical reaction of the fuel at the anode with the oxygen at the cathode. At the cathode, molecular oxygen combines with electrons and is reduced to negatively charged ions (O²⁻) with the aid of a catalyst [19, 23].

The degree to which an anode supports direct oxidation will then impact the degree of the reforming of the fuel that is required, which in turn typically impacts the balance of plant complexity and cost [23, 25]. The net cell reaction is thus written as:

\[
\text{CH}_4,\text{anode} + 2\text{O}_2,\text{cathode} \Leftrightarrow 2\text{H}_2\text{O,anode} + \text{CO}_2,\text{anode}
\]

In order to reduce the CO concentration out of the LTS, the preferential oxidation reaction (PROX) is performed.

\[
\begin{align*}
\text{CO} + 1/2 \text{O}_2 & \rightarrow \text{CO}_2 \\
\text{H}_2 + 1/2 \text{O}_2 & \rightarrow \text{H}_2\text{O}
\end{align*}
\]

2.3 Simulation of the steam reformer and SOFC internal reforming

Steam reforming is a method for hydrogen production from hydrocarbon fuels such as natural gas. This is achieved in a processing device called a reformer which reacts steam at high temperature with the natural gas fuel. In this study, both steam reforming and SOFC internal reforming reactors are modelled using HYSYS conversion reactors.
2.4 Simulation of water gas shift reactor

The CO content can be reduced to about 0.5% by reacting it with water at lower temperatures to produce additional hydrogen according to the WGS reaction (Eq. (3)). Commercial hydrogen plants generally perform the WGS in two stages: (i) High-temperature shift at 300-450°C using an oxide catalyst, and (ii) low-temperature shift at 200 – 250 °C using copper zinc oxide. Heat exchangers are required between shift reactors to provide cooling, and the conversion in an adiabatic reactor is limited because the reaction is exothermic and the temperature increases as the reaction proceeds. In this study, WGS reactors are modeled using equilibrium reactor. By using equilibrium reactor, HYSYS will determine the composition of the outlet stream given the stoichiometry of all reactions occurring and the value of equilibrium constant for each reaction.

2.5 Simulation of preferential oxidation reactor

Carbon monoxide is a poison to the precious metal catalyst in the anode of the PEM fuel cell. Preferential oxidation (PROX) is a reactive approach to destroy CO in the reformat composition. PROX of CO is typically used to reduce CO to the part per million levels required for the PEM fuel cell. The catalyst and conditions must be selected to minimize the oxidation of hydrogen. For the overall process model heat and material balance, 50% selectivity to CO oxidation is assumed, with the remainder of the oxygen reacting with hydrogen to form water. The PROX reactor was modeled in HYSYS as a conversion reactor based on two reactions to oxidize CO as shown in Eqs. (5) and (6).

3. The present simulation

Figs. 3 and 4 show the two reforming cases of PEM fuel cell system scheme simulation studied by Aspen- HYSYS 3.2 taking into account selected balance plant of plant equipment. The hydrocarbon fuel is first pressurized (2), and then vaporized (5). The vaporized hydrocarbon fuel is divided into two streams: One stream (6) is directed to the burner where it is combusted to provide the necessary process heat, the other stream (7) is mixed in the air-fuel mixer (AFM) with the hot compressed air (12) from the compressor. The air fuel mixture (13) is heated with the hot combustion gases (40) from the combustor up to the required PRE-SR temperature (35).
All of the chemical reactions are assumed to occur adiabatically under equilibrium conditions. The gases leaving SOFC reactor (14) are cooled (16) prior to entering the HTS reactor. The gases are further processed in LTS and PROX. The exit gases from the PROX (23) are fed to PEM fuel cell after cooling (25).

It is desired to maximize hydrogen concentration and to minimize carbon monoxide (CO) content considering the requirements of PEM fuel cells. The high and low temperature water- gas shift reactors (HTS and LTS) and the preferential oxidation (PROX) are used to decrease the CO concentration level of the SREF or SOFC reactors exit gas to the desired values.

Compressed air is divided into 4 streams in case of SOFC based reforming PEM fuel cell system: one stream is directed to the SOFC (8) as SOFC reactant; another stream is used in PROX (9); the third stream (10) supplies the cathode air of PEM fuel cell; the fourth air stream is the combustion air (11). Pressurized water (3) is converted to steam (4) to be used in SOFC. Water is circulated (41-42) to cool down the PEM fuel cell.

Anode and cathode off-gases (26) of the PEM fuel cell are combusted together with the hydrocarbon fuel (6). The combustor off- gases are expanded after exchanging heat with the hydrocarbon fuels to heat them up prior to SREF-SOFC entrance to produce additional power. The final burner exit gases (40) are above 500°C. The fuel cell stack is assumed to run under constant temperature and pressure, namely 70°C and 3bars. The PEM fuel cell characteristics are presented in Table 1.

<table>
<thead>
<tr>
<th>Table 1</th>
<th>The PEM fuel cell characteristics (e$: Electron)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2H₂ → 4H$^+$ + 4e$^-$</td>
<td>Anode Reaction</td>
</tr>
<tr>
<td>O₂ + 4H$^+$ + 4e$^-$→2H₂O</td>
<td>Cathode Reaction</td>
</tr>
<tr>
<td>65</td>
<td>Fuel utilization ( %)</td>
</tr>
<tr>
<td>70</td>
<td>Fuel cell outlet temperature (°C)</td>
</tr>
<tr>
<td>3</td>
<td>Pressure (bar)</td>
</tr>
<tr>
<td>800</td>
<td>Average cell voltage (mV)</td>
</tr>
<tr>
<td>Water</td>
<td>Stack cooling media</td>
</tr>
</tbody>
</table>

Table 2 summarizes the assumed data of different auxiliary system components utilized in the simulation studies. They are based on commercially available units [17, 26].
Steam and SOFC based reforming options of PEM fuel cells for marine applications

<table>
<thead>
<tr>
<th>Component</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel pump</td>
<td>Adiabatic efficiency (%)</td>
<td>75</td>
</tr>
<tr>
<td>Water pump</td>
<td>Adiabatic efficiency (%)</td>
<td>75</td>
</tr>
<tr>
<td>Cooling water pump</td>
<td>Adiabatic efficiency (%)</td>
<td>75</td>
</tr>
<tr>
<td>Compressor</td>
<td>Adiabatic efficiency (%)</td>
<td>70</td>
</tr>
<tr>
<td>Expander</td>
<td>Adiabatic efficiency (%)</td>
<td>75</td>
</tr>
<tr>
<td>DC/AC Converter</td>
<td>Conversion efficiency (%)</td>
<td>98</td>
</tr>
</tbody>
</table>

The thermal efficiencies of the REF, HTS, LTS and PROX reactors are $\eta_{\text{REF}}$, $\eta_{\text{HTS}}$, $\eta_{\text{LTS}}$ and $\eta_{\text{PROX}}$, respectively. They are defined as the ratio of the heating values and mass flows of the exit and inlet streams, Eqs. (7) to (11). The heating value of a stream is calculated by the multiplication of its lower heating value (LHV) with its mass flow rate in kg per hour. $\eta_1$ presents the fraction of the REF inlet stream heating value to the heating value of the total fuel feed to the system. The remainder is fed to the burner. The total fuel processing efficiency is the product of $\eta_1$, $\eta_{\text{REF}}$, $\eta_{\text{HTS}}$, $\eta_{\text{LTS}}$ and $\eta_{\text{PROX}}$ as shown in Eq. (12).

$$\eta_1 = \frac{\dot{m}_7 \times \text{LHV}_7}{\dot{m}_F \times \text{LHV}_F}$$  \hspace{1cm} (7)

$$\eta_{\text{REF}} = \frac{\dot{m}_{16} \times \text{LHV}_{16}}{\dot{m}_{35} \times \text{LHV}_{35}}$$  \hspace{1cm} (8)

$$\eta_{\text{HTS}} = \frac{\dot{m}_{19} \times \text{LHV}_{19}}{\dot{m}_{16} \times \text{LHV}_{16}}$$  \hspace{1cm} (9)

$$\eta_{\text{LTS}} = \frac{\dot{m}_{22} \times \text{LHV}_{22}}{\dot{m}_{19} \times \text{LHV}_{19}}$$  \hspace{1cm} (10)

$$\eta_{\text{PROX}} = \frac{\dot{m}_{25} \times \text{LHV}_{25}}{\dot{m}_{22} \times \text{LHV}_{22}}$$  \hspace{1cm} (11)

$$\eta_{\text{FP}} = \eta_1 \times \eta_{\text{REF}} \times \eta_{\text{HTS}} \times \eta_{\text{LTS}} \times \eta_{\text{PROX}}$$  \hspace{1cm} (12)

The PEM fuel cell module has been simulated using the PEM fuel cell characteristics presented in Table 2. The SOFC and PEM fuel cell stack efficiencies depend on fuel utilization coefficient ($U_f$), stack voltage, and DC/AC conversion efficiencies. Fuel cell voltage ($V_{\text{cell}}$) is the difference between cell voltage at no load, which can be called open circuit voltage and the specific fuel cell irreversibility or voltage drop. The following Eq. (13) shows the operating voltage of a fuel cell at a current density ($i_{\text{den}}$) [19, 27].

$$V_{\text{cell}} = E_o - (i_{\text{den}} \times r) - A \times \ln(i_{\text{den}}) + m \times e^{(n \times i_{\text{den}})}$$  \hspace{1cm} (13)

$$\eta_{\text{stack voltage}} = \frac{V_{\text{cell}} \times U_f}{E_o}$$  \hspace{1cm} (14)

$$\eta_{\text{DC/AC}} = 0.98$$  \hspace{1cm} (15)

$$\eta_{\text{FC}} = \eta_{\text{stack voltage}} \times \eta_{\text{DC/AC}}$$  \hspace{1cm} (16)
In Eq. (13), $E_o$ is the open circuit voltage, $i_n$ internal current density, 'A' is slope of Tafel curve, 'm' and 'n' are constants, 'r' is specific resistance. Typical values of these constants for a SOFC and PEM fuel cell systems are given in Table 3.

<table>
<thead>
<tr>
<th>Constant</th>
<th>SOFC</th>
<th>PEMFC</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_o$ (V)</td>
<td>1.01</td>
<td>1.031</td>
</tr>
<tr>
<td>r (kΩcm$^{-2}$)</td>
<td>2.0×10$^{-3}$</td>
<td>2.45×10$^{-4}$</td>
</tr>
<tr>
<td>A (V)</td>
<td>0.002</td>
<td>0.003</td>
</tr>
<tr>
<td>m (V)</td>
<td>1.0×10$^{-4}$</td>
<td>2.11×10$^{-5}$</td>
</tr>
<tr>
<td>n (cm$^2$mA$^{-1}$)</td>
<td>8×10$^{-3}$</td>
<td>8×10$^{-3}$</td>
</tr>
</tbody>
</table>

Auxiliary units comprise pumps, compressor, expander, heat exchangers, heaters, coolers, and burner. The auxiliary system efficiency ($\eta_{Aux}$) is calculated as follows:

\[
\eta_{motor} = 0.9 \\
\eta_{Aux} = 1 + \frac{(P_E - P_{par})}{P_{PEM,AC}}
\]

Extensive heat integration is sought within the study to achieve acceptable overall system efficiency levels. The overall system efficiency ($\eta_{net.el}$) is calculated as the product of fuel processing ($\eta_{FP}$), both SOFC and PEM fuel cell ($\eta_{FC}$) and auxiliary ($\eta_{Aux}$) system efficiencies.

\[
\eta_{net.el} = \eta_{FP} \times (\eta_{PEMFC} + \eta_{SOFC}) \times \eta_{Aux}
\]

4. Results and discussion

The performance of both SOFC and PEM fuel cells can be described by the polarization curve, which relates the cell voltage to its current density. This polarization curve is affected by the losses of the fuel cell. Fig. 5 shows the polarization curves of both SOFC and PEM fuel cells. As the cell current density increases, there will be a drop of the output voltage of the fuel cell. This drop of the cell voltage will be higher in SOFC than that of PEMFC. As the current density reaches its maximum value, the SOFC voltage drops sharply to zero before than PEMFC voltages does.
Fig. 5 SOFC and PEMFC polarization curves

In the following, the results obtained for the reforming of natural gas using SREF and SOFC-PEM shown in Fig.s 3 and 4 are presented. The components results for the selected reforming options for PEM fuel cell. The operating points for the SOFC and PEM fuel cells are at cell output current density of 250 mA/cm², cell voltage and fuel utilization coefficient of (0.804 volts, 65%) and (0.499 volts, 45%) for PEM and SOFC respectively. Moreover, AF and SC ratios are (5, 8.25) and (7.75, 8.5) for SREF and SOFC reforming options respectively. In addition, the percent of the fuel supplied to the burner is 1.4% from the supplied fuel to the system. Tables 4 and 5 show the results of selected system points calculated under the prescribed operating conditions applied in the two cases of reforming.

Table 4 Simulation results for selected system points calculated under the prescribed operating conditions applied in SREF-PEM

<table>
<thead>
<tr>
<th>Stream</th>
<th>Fuel</th>
<th>Air</th>
<th>Water</th>
<th>5</th>
<th>6</th>
<th>13</th>
<th>8</th>
<th>10</th>
<th>11</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>25</td>
<td>20</td>
<td>25</td>
<td>200</td>
<td>200</td>
<td>495</td>
<td>206</td>
<td>206</td>
<td>206</td>
</tr>
<tr>
<td>Pressure (kPa)</td>
<td>120</td>
<td>100</td>
<td>170</td>
<td>300</td>
<td>300</td>
<td>300</td>
<td>400</td>
<td>400</td>
<td>400</td>
</tr>
<tr>
<td>Mass flow (kg/hr)</td>
<td>40</td>
<td>200</td>
<td>330</td>
<td>40</td>
<td>0.55</td>
<td>369</td>
<td>1.5</td>
<td>55</td>
<td>143</td>
</tr>
<tr>
<td>Stream</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>500</td>
<td>850</td>
<td>350</td>
<td>200</td>
<td>250</td>
<td>120</td>
<td>150</td>
<td>70</td>
<td>513</td>
</tr>
<tr>
<td>Pressure (kPa)</td>
<td>300</td>
<td>300</td>
<td>168</td>
<td>170</td>
<td>170</td>
<td>163</td>
<td>163</td>
<td>300</td>
<td>300</td>
</tr>
<tr>
<td>Mass flow (kg/hr)</td>
<td>369</td>
<td>369</td>
<td>369</td>
<td>25</td>
<td>25</td>
<td>25</td>
<td>25</td>
<td>25</td>
<td>25</td>
</tr>
</tbody>
</table>

Table 5 Simulation results for selected system points calculated under the prescribed operating conditions applied in SOFC-PEM

<table>
<thead>
<tr>
<th>Stream</th>
<th>Fuel</th>
<th>Air</th>
<th>Water</th>
<th>5</th>
<th>6</th>
<th>13</th>
<th>8</th>
<th>10</th>
<th>11</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>25</td>
<td>20</td>
<td>25</td>
<td>200</td>
<td>200</td>
<td>455</td>
<td>162</td>
<td>162</td>
<td>162</td>
</tr>
<tr>
<td>Pressure (kPa)</td>
<td>120</td>
<td>100</td>
<td>170</td>
<td>300</td>
<td>300</td>
<td>300</td>
<td>300</td>
<td>300</td>
<td>300</td>
</tr>
<tr>
<td>Mass flow (kg/hr)</td>
<td>40</td>
<td>310</td>
<td>340</td>
<td>40</td>
<td>0.55</td>
<td>379</td>
<td>105</td>
<td>65</td>
<td>173</td>
</tr>
<tr>
<td>Stream</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>500</td>
<td>850</td>
<td>350</td>
<td>100</td>
<td>250</td>
<td>120</td>
<td>150</td>
<td>70</td>
<td>536</td>
</tr>
<tr>
<td>Pressure (kPa)</td>
<td>300</td>
<td>300</td>
<td>168</td>
<td>170</td>
<td>170</td>
<td>163</td>
<td>163</td>
<td>300</td>
<td>300</td>
</tr>
<tr>
<td>Mass flow (kg/hr)</td>
<td>379</td>
<td>450</td>
<td>450</td>
<td>450</td>
<td>450</td>
<td>450</td>
<td>451</td>
<td>451</td>
<td>960</td>
</tr>
</tbody>
</table>
The product compositions and LHV results for SREF and SOFC cases at the operating points can be shown in the following Figs. 6 and 7. With the developed system models which are implemented in the HYSYS 3.2 process simulator, effluents from all reactors are simulated. A considerably wide SC and AF ratios has been changed to see its effect on hydrogen yield and CO formation. The selected operating point achieves high reformer efficiency and acceptable CO content for PEM. CO content in the product steam changes from 2.3% and 3.3% to 0.0 for steam and SOFC based reforming options before entering PEMFC.

![Fig. 6 Product compositions and LHV values of steam reforming based fuel preparation reactors.](image)

![Fig. 7 Product compositions and LHV values of SOFC based fuel preparation reactors.](image)

Fig. 8 shows the molar fractions and product lower heating values of all components in the effluent of the two reformer reactors of the natural gas fuel processor system. In SREF case, 100% methane is converted to produce 30.8% hydrogen, 6.0% CO2 and 2.4% CO. In addition, under these conditions, oxygen is 100% consumed. Simultaneously, in the case of SOFC-PEM, 100% methane is converted to produce 22.6% hydrogen, 4.2% CO2 and 3.2% CO.
The operating parameters of the reforming process are of utmost importance to achieve the desired high hydrogen and low CO content product gases along with acceptable fuel conversion efficiency level. The total air to fuel ratio range studied is between 3.0 and 5.0 for the SREF-PEM as shown in Fig. 9. Similar approach has been adopted for SOFC-PEM; in this case air to fuel consumption \( (m_A/m_f) \) has been changed between 1.0 and 3.0 as shown in Fig. 10. It can be noticed that, in the case of SREF-PEM, the reformer efficiency as well as the hydrogen content of the product gases steadily decreases as the air to fuel ratio decreases. In addition, increasing the steam flow rate increases the reformer efficiency but this increase will affect the following reformers like PROX and burner exhaust temperature. In the other case, the reformer efficiency decreases as the reformer air to fuel ratio decreases at low steam to carbon ratios. In contrast, at high steam to carbon ratios the reformer efficiency increases as the air to fuel ratio decreases.

![Fig. 8 The molar compositions of the two cases of reformer products](image)

![Fig. 9 Reformer efficiency as a function of steam flow rate and total system AF ratio (SREF-PEM)](image)
The two reforming options based fuel-processing, fuel cell, auxiliary and overall system efficiencies of the investigated natural gas fuel are presented in Table 6. The values indicate that reforming of natural gas either by steam reforming or by SOFC based reforming option achieves high fuel processing efficiency. Moreover, natural gas based systems do not require the pre-reformer unit compared with liquid fuel systems due to their high lower molecular weight hydrocarbon, namely CH$_4$ content [32, 33]. In addition, both fuel cell efficiency and net electrical efficiency depends on fuel utilization coefficient fuel cell output current density.

### Table 6 SREF and SOFC based fuel processing options for PEM fuel cell system

<table>
<thead>
<tr>
<th>System</th>
<th>$U_f$</th>
<th>$\eta_{FP}$</th>
<th>$\eta_{FC}$</th>
<th>$\eta_{Aux}$</th>
<th>$\eta_{net.el}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reformer-PEM</td>
<td>0.65</td>
<td>96.48</td>
<td>50.71</td>
<td>-----</td>
<td>90.7</td>
</tr>
<tr>
<td>SOFC-PEM</td>
<td>0.45</td>
<td>(SOFC)-0.65(PEMFC)</td>
<td>95.27</td>
<td>50.71</td>
<td>22.26</td>
</tr>
</tbody>
</table>

It can be noticed from Table 6 that, heat integration within reforming, cleanup sections and PEM fuel cell components are the most important factor to achieve high PEMFC efficiency levels. These efficiency levels will be necessary to achieve the aims of the international emission regulations and to improve the total efficiency of marine power plants.

The obtained net electrical efficiency levels are at 44% and 62% for SREF and SOFC based reforming options for PEM fuel cells. These efficiency levels are higher than those of Otto engines. Therefore, both Steam reforming and SOFC internal reforming options of natural gas offers an efficient, and can be widely used for hydrogen production, and for near and mid-term energy provide with a good environmental benefits. Natural gas is a convenient, easy to handle, hydrogen feedstock with a high hydrogen to-carbon ratio. The cost of hydrogen produced by methane is acutely dependent on natural gas prices.

### 5. Conclusions

- PEM fuel cells generate electrical power from air and from hydrogen or hydrogen rich gas mixtures. Therefore, there is an increasing interest in converting current hydrocarbon based transportation fuels such as Natural gas into hydrogen rich gases acceptable by PEM fuel cells on board ships. In addition PEM fuel cell fuelled by natural gas is an attractive option to limit the environmental impact of the marine sector in order to satisfy the requirements of international regulations and to achieve high efficiency.
As a model for application, a SREF and SOFC based reforming options for PEM fuel cell system at cell output current density of 250 mA/cm², and fuel utilization coefficients of 65% and 45% for PEM and SOFC respectively has been carried out. Among the conditions studied, the highest fuel processing efficiency is achieved at about 96% with AF= 5.0 and SC=8.25 using SREF-PEM fuel cell system. Also, SOFC-PEM fuel cell system resulted in high fuel processing efficiency at 95% incorporating AF and SC ratios of 7.75 and 8.5 respectively.

The simulation results of SREF-PEM showed that, the obtained net electrical efficiency level is at 44% which is higher than those of Otto engines and competitive to that of diesel engines. In SOFC-PEM fuel cell, the advantages of each type of fuel cell are enhanced by operating in synergy. It is found that a high overall efficiency approaching 60% can be achieved using SOFC–PEM systems which are significantly better than a SREF-PEM system or an SOFC only system.

Finally, high PEM fuel cell system efficiency levels can be achieved only with intensive heat integration within the PEMFC systems. Hence, heat integration within PEMFC components system studies along with the development of reforming and clean-up systems are of utmost importance if hydrogen production is desired on-board ships.

ACKNOWLEDGEMENT

This work was funded by the Deanship of Scientific Research (DSR), King Abdulaziz University, Jeddah, under grant no. (980-563-D1435). The authors, therefore, acknowledge with thanks DSR technical and financial support.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Slope of Tafel curve</td>
<td>Volt</td>
</tr>
<tr>
<td>E₀</td>
<td>Open circuit voltage</td>
<td>Volt</td>
</tr>
<tr>
<td>i_dcm</td>
<td>Load current density</td>
<td>mA/cm²</td>
</tr>
<tr>
<td>LHV</td>
<td>Lower heating value</td>
<td>MJ/Kg.mole</td>
</tr>
<tr>
<td>m</td>
<td>Fuel cell voltage constant</td>
<td>Volt</td>
</tr>
<tr>
<td>ṁ</td>
<td>Mass flow rate</td>
<td>kg/s</td>
</tr>
<tr>
<td>n</td>
<td>Fuel cell voltage constant</td>
<td>cm²/ mA⁻¹</td>
</tr>
<tr>
<td>P_C</td>
<td>Power of compressor</td>
<td>kW</td>
</tr>
<tr>
<td>P_E</td>
<td>Total power of expanders</td>
<td>kW</td>
</tr>
<tr>
<td>P_par</td>
<td>Parasitic power</td>
<td>kW</td>
</tr>
<tr>
<td>P_P1</td>
<td>Feed water pump power</td>
<td>kW</td>
</tr>
<tr>
<td>P_P2</td>
<td>Fuel cell cooling water pump power</td>
<td>kW</td>
</tr>
<tr>
<td>P_PEM,AC</td>
<td>Fuel cell output AC power</td>
<td>kW</td>
</tr>
<tr>
<td>r</td>
<td>Specific resistance</td>
<td>kΩ*cm²</td>
</tr>
<tr>
<td>V_cell</td>
<td>Fuel cell voltage</td>
<td>Volt</td>
</tr>
</tbody>
</table>
Greek symbols

- $\eta_{\text{Aux}}$: Auxiliary systems efficiency
- $\eta_{\text{DC/AC}}$: DC / AC conversion efficiency
- $\eta_{\text{FC}}$: Fuel cell efficiency
- $\eta_{\text{FP}}$: Fuel processing efficiency
- $\eta_{\text{HTS}}$: High temperature shift reactor efficiency
- $\eta_{\text{LTS}}$: Low temperature shift reactor efficiency
- $\eta_{\text{motor}}$: Electric motor efficiency
- $\eta_{\text{net.el}}$: Net electric efficiency
- $\eta_{\text{PEMFC}}$: Proton exchange membrane fuel cell efficiency
- $\eta_{\text{PROX}}$: Preferential oxidation efficiency
- $\eta_{\text{REF}}$: Reforming section efficiency
- $\eta_{\text{SOFC}}$: Solid oxide fuel cell efficiency
- $\eta_{\text{Stack voltage}}$: Stack voltage efficiency
- $\eta_{\text{1POX inlet to total fuel feed}}$: POX inlet to total fuel feed efficiency

Abbreviations

- AC: Alternating current
- AF: Air to fuel ratio
- DC: Direct current
- ECAs: Emission Control Areas
- HTS: High temperature shift reactor
- H$_2$O: Water vapor
- IMO: International maritime organization
- LNG: Liquefied natural gas
- LTS: Low temperature shift reactor
- NO$_x$: Nitrogen oxides emissions
- PEMFC: Proton exchange membrane fuel cell
- PROX: Preferential oxidation reactor
- ppm: Part per millions
- REF: Reforming
- SC: Steam to carbon ratio
- SREF: Steam reforming
- SOFC: Solid oxide fuel cell
- SO$_x$: Sulfur oxides emissions
- SREF-PEM: Steam reforming based proton exchange membrane fuel cell
- SOFC-PEM: SOFC based reforming option for PEMFC
- $U_f$: Fuel utilization coefficient
- WGS: Water gas shift reactor
REFERENCES


SHIPBUILDING PRODUCTION PROCESS DESIGN METHODOLOGY USING COMPUTER SIMULATION

UDC 629.5.081
Professional paper

Summary

In this research a shipbuilding production process design methodology, using computer simulation, is suggested. It is expected from suggested methodology to give better and more efficient tool for complex shipbuilding production processes design procedure. Within the first part of this research existing practice for production process design in shipbuilding was discussed, its shortcomings and problem were emphasized. In continuing, discrete event simulation modelling method, as basis of suggested methodology, is investigated and described regarding its special characteristics, advantages and reasons for application, especially in shipbuilding production process. Furthermore, simulation modeling basics were described as well as suggested methodology for production process procedure. Case study of suggested methodology application for designing a robotized profile fabrication production process line is demonstrated. Selected design solution, acquired with suggested methodology was evaluated through comparison with robotized profile cutting production line installation in a specific shipyard production process. Based on obtained data from real production the simulation model was further enhanced. Finally, on grounds of this research, results and droved conclusions, directions for further research are suggested.

Key words: shipbuilding; production process design; decision making; computer simulation

1. Introduction

On today's market, shipyard continuously has to invest in improvement in their production process and technology so to increase productivity and profit. Therefore, shipyard management is often conducting significant actions in their production process, especially in terms of implementing new technologies into the existing production process, which is a complex task. Design of the new production process is a task that is often based on various
assumption within known existing limitations, furthermore, solution is necessary the result of interaction between dependent decision making variables [1]. Regarding these issues, the author has analyzed existing design methods, techniques and tools for designing production processes, and the shipbuilding process in particular [2]. Following perceived shortcoming of existing method, the need for a new scientifically founded methodology for shipbuilding process design is identified. Such method should provide a better support within implementation of shipyards new technologies, within managing and improving of existing ones, and within decision making process overall. Therefore, in this paper a methodology for shipbuilding production processes design based on simulation modeling method and chosen operation research methods will be presented. Suggested methodology and designed computer simulation model was tested through case study of particular shipyards production process design and was confirmed after the production line installation. Model was further enhanced with real process data and confirmed against several different production scenarios and as such it has potential to be used in real production process for scheduling, conducting what – if scenarios, optimization, planning, control, etc.

2. Problem discussion

Shipyard production process, regarding its characteristics, is one of the most complex business and production systems. This complexity is the result of the complexity of its final products - ship, individual product of high capital value, which are mainly of different types and sizes. Such a complex product requires equally complex shipbuilding process with following fundamental characteristics [3]: large number of intermediate products; significant interaction and interdependence of processes; mostly it is about non-repeating processes with different durations; process input contains a large number of components, but with small number of different output final products; processes are conducted in many parallel sub processes, with greater or lesser time overlaps; processes are technologically different, using different means of work; production process has both, a "movement of products through the process," as well as "the process moving through the product". On figure 1, a simplified scheme of a shipbuilding production process is given. Within conducted research, various existing methods, techniques and tools for production process design were investigated and shortcomings of such methods are identified, especially in terms of above mentioned complexity of the shipyard production process [2, 4]. In general, with a conventional approach a design solution is commonly defined based on comparison with other shipyards which already have similar technology. Such solution in particular cases can be satisfactory, however not necessarily optimally adapted to the observed shipyard [5, 6]. For that matter, the application of the scientific methods for process design and improvement is more widely accepted, i.e. relevant methods of mathematical modeling [7, 8]. However authors identifies shortcomings of conventional mathematical modeling and analytical approaches for designing complex production processes, such as shipbuilding, which makes the application of conventional mathematical methods with certain limiting factors, such as [9]: real production process, elements and their relations are often insufficiently known and can’t be mathematically defined; real problems are often very complex, which makes its analytical definitions very difficult; with conventional mathematical modeling it is difficult to render dynamics of observed process. Following identified issues, within this research, a new methodology for shipbuilding production processes design, based on simulation modeling method, and chosen operation research methods and tools is presented.
3. **Problem solving methodology**

Based on conducted analysis and identified shortcomings a new methodology for shipbuilding production process design is developed and suggested, with discrete event simulation modeling as its basic method.

3.1 **Discrete event simulation modeling**

The term simulation modeling expresses a complex activity that involves three elements: the actual system, model and the computer. Simulation can be defined as the process of establishing a dynamic model of the actual dynamic system, within the defined requirements and limits, for the purpose of understanding the behavior of the real system and evaluation of different design and/or production alternatives for the design of new system or to improve of the existing one [10]. Within proposed methodology, an object oriented SimTalk language, within discrete event simulation modelling software *eM-Plant*, is used. Discrete event simulation is used because the system of the production process researched in this work is mainly recognized as discrete event system. In such system each event occurs at a particular instant in time and marks a change of state in the system, between consecutive events, no
change in the system is assumed to occur [11]. A computer simulation model, compared to traditional analytic model, is more descriptive, more manageable and it allows designers to verify various decisions alternatives on computer, fast and in early design stages. [12]. Furthermore, such approach makes the final decision more reliable and better adapted to the observed shipyard because it provides a lot of relevant and timely information enabling more reliable and lower risk decisions with solution better adapted to the particular shipyard prior to the line installation. In general, some of the most significant reasons why simulation modeling method is suggested as basic method for production process design are. [13]:

- computer simulation model can be used for evaluating different design alternatives (what-if scenarios) prior to the final investment;
- computer simulation model can be used for experimenting with certain critical equipment parameters without influencing the real process;
- using computer simulation model, it is possible to spot process bottleneck on its computer model before they occur in the real process; using process simulation computer model could improve process productivity; using process simulation computer model could improve scheduling policy; using process simulation computer model could reduce production costs and improve quality, etc.

On the other hand, one should be aware that simulation modeling process could be time consuming and costly so it should not be used if for example: problem can be solved faster and easier analytically; problem can be solved using classic experiment; developing simulation model costs more than potential benefits; there is no time which developing simulation model requires; simulation model results can not be confirmed; behavior and characteristics of modeled system is too complex and unknown.

3.2 Proposed methodology description

In this work, of particular interest will be the case of shipbuilding production process and its computer simulation model. Methodology itself is structured through seven phases as follows:

Phase 1: Problem and project goal definition; Within this phase existing process should be analyzed and problems, goals and deadlines should be defined using methods and tools such as graphic process flow, cause effect diagram, pareto chart, benchmarking (SWOT, comparison tables, expert survey, potential analysis) etc. Main tasks of this phase are as follows: define problems and its causes, and what has to be improved; project goals should be clearly defined; responsibilities and deadlines should be defined.

Phase 2: Definition of input data and conceptualization of simulation model; The main goal of this phase is to gather required input data, establish preliminary new design solution and it simulation model using methods and tools such as cause effect diagram, CAD tools, process flow chart, simulation object programming language, etc. Main tasks of the phase 2 are: definition of input data and preliminary new design solution (defining equipment CAD drawings, process flowchart, cause effect chart. etc.); conceptualization simulation model (simulation model of new production process should be conceptually defined).

Phase 3: Computer simulation model development; The main goal of this phase is to develop functional computer simulation model of new production process design using primarily discrete event simulation model method and tools such as regression analysis, statistic analysis, simulation, etc. Main tasks of this phase are: organization and systematization of gathered data (understands overview of available data and identification of missing ones); definition of input production data (input production data as basis for simulation model should be defined); developing of computer simulation model (computer simulation model of new design is developed within discrete simulation software).

Phase 4: Verification of simulation model; The main goal of this phase is verifications of developed simulation model and confirm it for further analysis, to establish confidence in
functionality and logic of developed simulation model. Methods used are mainly benchmarking (comparison tables) and expert survey. For that matter verification of model understands removing logical mistakes from model and insuring full functionality of the model.

Phase 5: Production scenarios analysis and improvement of simulation model; The main goal of this phase is to evaluate simulation model of design solution and its potential improvement. This phase should result with definition of line parameters as to satisfy project goals. Main task of this phase are: analysis and validation of simulated design solution (design solution should be analyzed against project goals as to find if the goals of the project are satisfied. If not, solution should be further analyzed and improved); analysis and improvement of simulated design solution. Hereby suggested methods and tools used are: for validation of design solution, material flow analysis and production line load analysis simulation method is used; with sensitivity analysis result is tested against changes of line parameters and most influence one are identified.

Phase 6: Results documenting; Main task of this phase is to document project procedures and results on clear and understandable manner.

Phase 7: Implementation of design solutions; the main goal of this phase is implementation of suggested design solution into the real shipyard production process. Main task of this phase are: implementation of the final design solution into the real shipyard process; improvement of simulation model (simulation model is further improved based on gathered data from real production process). Such improved model can be used for continuous production improvement and production planning.

In Table 1, a condense presentation of methodology phases and associated main tasks is shown.

Table 1 Condense presentation of methodology phases and associated main tasks

<table>
<thead>
<tr>
<th>Phase No.</th>
<th>Main tasks</th>
</tr>
</thead>
</table>
| 1.        | - Problem definition  
- Project goal and deadline definition |
| 2.        | - Input data definition  
- Simulation model conceptualization |
| 3.        | - Definition of input product mix  
- Computer simulation model development |
| 4.        | - Simulation model verification |
| 5.        | - Analysis of simulated design  
- Improvement of simulated design |
| 6.        | - Documenting procedures and results |
| 7.        | - Implementation of final design solution |

4. Methodology application case study

Developed methodology was applied and tested through case study of designing shipyards robotized profile fabrication production line. Methodology was conducted following the defined procedure explained in this section.
4.1 Identifying the goal of the new robotized profile fabrication line design

Existing profile fabrication line, in observed shipyard is obsolete and have inadequate throughput rate, occupies too large production area and workers. Therefore, the shipyards major goal to design a new, robotized profile fabrication line, which will require less space, be more efficient and have larger throughput rate. Method used in this phase, for defining initial design, was mainly benchmarking through comparison with similar shipyard already having such production line, where shipyard and chosen equipment manufacturer had suggested an initial production process design. Line throughput was initially estimated using average profile production time which was provided by equipment manufacturer. However, such solution, based on average profile, was not fully satisfactory. It than was required to test suggested solution with a production data from typical ship sections of several ship types in order to minimize the decision making risk and to be more certain that suggested production line will comply the required throughput. Therefore, it was decided to develop a simulation model of initially suggested production line solution. Such model will be tested with selected production mix of chosen ship types as to evaluated if suggested solution fulfills required throughput. If not, the line will be further analyzed and improved in order to achieve required throughput. Such conclusion could be than communicated to the equipment manufacturer with requirement to improve initially suggested solutions to the particular demands. In this way, decision making involves much less risk with the final solution more adapted to the particular shipyard and expectably with reduced overall cost. To summarize, the major goal for application of developed methodology is: based on developed computer simulation model, it have to be tested if manufacturer suggested design solution of new robotized line match the minimum throughput requirement, \( T_{\text{min}} \); if not, it should be suggested how to improve line characteristics and its parameters.

4.2 Computer simulation model conceptualization and development

Based on initially suggested design solution of the new robotized profile cutting line, conceptual cause effect chart (Figure 2) and production process flow chart (Figure 3) are created. Furthermore, preliminary technical characteristics of the line, operation and material flow characteristics and input production data are defined. This data are partially accepted from equipment supplier and partially from shipyards experts survey method. Most important input parameters from observed production line, as input data for conceptual simulation model, are presented in table 2. Based on conducted analysis, gathered data and defined production process, computer simulation model is conceptually defined regarding its structure, logic, functionality and organization.
Shipbuilding production process design methodology using computer simulation

Marko Hadjina, Nikša Fafandjel, Tin Matulja

Fig. 2 Robotized profile fabrication line cause effect chart

Fig. 3 Robotized profile fabrication line conceptual production process flow chart

Table 2 Production line elements input parameters for simulation

<table>
<thead>
<tr>
<th>Production line element</th>
<th>Description</th>
<th>Defined parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conveyer VT₁</td>
<td>Conveyer profile transport from prefabrication to profile fabrication production line</td>
<td>Length/width, Speed range, Capacity</td>
</tr>
<tr>
<td>Rotating conv. RT</td>
<td>Rotation conveyer</td>
<td>Length/width, Speed range, Rotation speed, Capacity</td>
</tr>
<tr>
<td>Conveyer VT₂</td>
<td>Conveyer profile side transport to buffer storage</td>
<td>Length/width, Speed range, Capacity</td>
</tr>
<tr>
<td>Bankata 1-5</td>
<td>Standard package of profiles</td>
<td>Capacity</td>
</tr>
<tr>
<td>Component</td>
<td>Description</td>
<td>Dimensions/Capacity</td>
</tr>
<tr>
<td>-------------------</td>
<td>------------------------------------------------------------------------------</td>
<td>----------------------------------------------------------</td>
</tr>
<tr>
<td>Buffer MS₁</td>
<td>Buffer storage</td>
<td></td>
</tr>
<tr>
<td>Conveyer VT₃</td>
<td>Conveyer profile transport from buffer to blasting station</td>
<td>Length/width, Speed range, Capacity</td>
</tr>
<tr>
<td>Conveyer VT₄</td>
<td>Conveyer profile side transport to buffer storage and robot cutting station</td>
<td>Length/width, Speed range, Capacity</td>
</tr>
<tr>
<td>Buffer MS₂</td>
<td>Buffer storage before robot cutting station</td>
<td>Dimensions, Capacity</td>
</tr>
<tr>
<td>Robotized cutting station</td>
<td>Profile cutting station</td>
<td>Cutting speed</td>
</tr>
<tr>
<td>Conveyer VT₅</td>
<td>Conveyer profile transport from cutting station</td>
<td>Length/width, Speed range, Capacity</td>
</tr>
<tr>
<td>Buffer MS₃</td>
<td>Buffer storage before profile sorting</td>
<td>Dimensions, Capacity</td>
</tr>
<tr>
<td>Pallet Pca</td>
<td>Pallet with profiles for automatic subassembly line</td>
<td>Capacity</td>
</tr>
<tr>
<td>Pallet Pcr</td>
<td>Pallet with profiles for robotized subassembly line</td>
<td>Capacity</td>
</tr>
<tr>
<td>Pallet Pmp</td>
<td>Pallet with profiles for subassembly line</td>
<td>Capacity</td>
</tr>
<tr>
<td>Conveyer VT₆</td>
<td>Conveyer profile transport to panel line buffer storage</td>
<td>Length/width, Speed range, Capacity</td>
</tr>
<tr>
<td>Profiles</td>
<td>Each profile from profile stockyard is defined and imported from external database</td>
<td>Type, Dimensions, Number of cuts, Type of cuts, Cutting length, Scrap</td>
</tr>
</tbody>
</table>

Based on defined production process cause effect chart, process flow chart, technical characteristics of the line, the simulation model of the new robotized profile fabrication cutting line has been developed in specialized discrete simulation software, (Figure 4). Hereby, it is important to distinguishing between relevant and irrelevant facts and making the right assumptions is of essential importance for the development of a quality model as the basis for simulation modeling process. Therefore, it is important to: determine what should be included in the model and the level of details in the model; distinguish basic resources in the process and operations that are performed; identify and define any limitations on the system in terms of spatial or temporal constraints; agree and define the conditions and methods for verification and confirmation of the model; determine what is expected as the final result as well as deadlines.
Further, input material specification as production mix for a simulation model was defined (Table 3), where product mix specific configuration was defined through shipyard expert surveying method. Sample contains profiles and flat bars from double bottom of ship for chemical products and profiles and flat bars from bottom of ship for asphalt.

**Table 3 Input material specification**

<table>
<thead>
<tr>
<th>Sample mix of profiles (HP) and flat bars (FB)</th>
<th>Height, [mm]</th>
<th>Thickness, [mm]</th>
<th>Number of pieces</th>
</tr>
</thead>
<tbody>
<tr>
<td>HP 220 x 11.5</td>
<td>220</td>
<td>11.5</td>
<td>30</td>
</tr>
<tr>
<td>HP 240 x 10</td>
<td>240</td>
<td>10</td>
<td>22</td>
</tr>
<tr>
<td>HP 280 x 11</td>
<td>280</td>
<td>11</td>
<td>2</td>
</tr>
<tr>
<td>HP 340 x 12</td>
<td>340</td>
<td>12</td>
<td>20</td>
</tr>
<tr>
<td>HP 340 x 14</td>
<td>340</td>
<td>14</td>
<td>60</td>
</tr>
<tr>
<td>HP 370 x 13</td>
<td>370</td>
<td>13</td>
<td>54</td>
</tr>
<tr>
<td>HP 370 x 15</td>
<td>370</td>
<td>15</td>
<td>4</td>
</tr>
<tr>
<td>HP 400 x 14</td>
<td>400</td>
<td>14</td>
<td>2</td>
</tr>
<tr>
<td>FB 80 x 15</td>
<td>80</td>
<td>15</td>
<td>12</td>
</tr>
<tr>
<td>FB 120 x 13</td>
<td>120</td>
<td>13</td>
<td>2</td>
</tr>
<tr>
<td>FB 130 x 12</td>
<td>130</td>
<td>12</td>
<td>108</td>
</tr>
<tr>
<td>FB 150 x 12</td>
<td>150</td>
<td>12</td>
<td>96</td>
</tr>
<tr>
<td>FB 150 x 13</td>
<td>150</td>
<td>13</td>
<td>521</td>
</tr>
<tr>
<td>FB 150 x 15</td>
<td>150</td>
<td>15</td>
<td>1205</td>
</tr>
<tr>
<td>FB 200 x 10</td>
<td>200</td>
<td>10</td>
<td>5</td>
</tr>
<tr>
<td>FB 200 x 15</td>
<td>200</td>
<td>15</td>
<td>6</td>
</tr>
<tr>
<td>FB 200 x 20</td>
<td>200</td>
<td>20</td>
<td>14</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td></td>
<td></td>
<td><strong>2136</strong></td>
</tr>
</tbody>
</table>
4.3 Simulation model verification, analysis and improvement

Initial model verification is conducted in cooperation with shipyards experts. Such verification is primarily involving testing process model logic, functionality, behavior and results mainly based on shipyard expert experience and known data. If required, model is fine tuned in several iterations, until final confirmations. Such confirmed model can be used to evaluate if suggested design is fulfilling the project goal which is: simulated fabrication time, \( F_{tsim} \), of initially suggested design solution for defined characteristic input production mix of profiles and flat bars, should be less than minimum fabricated time, \( F_{tmin} \), based on defined goal throughput rate of profiles – \( T_{min} \), achieved in one month and two shifts. Therefore:

\[
F_{tsim} < F_{tmin} \quad (1)
\]

and

\[
F_{tmin} = \frac{N_p}{T_{min}} \cdot N_{wd} \cdot N_s \cdot N_{whs} \quad [h] \quad (2)
\]

Where \( F_{tsim} \) is simulated fabrication time for chosen input production data of initially suggested design solution; \( F_{tmin} \) is minimum required fabrication time for chosen input production data; \( N_p \) is number of profiles and flat bars in chosen product mix; \( T_{min} \) is number of profiles and flat bars in targeted shipyard month production; \( N_{wd} \) is number of working days in month; \( N_s \) is number of working shifts in day; \( N_{whs} \) is number of working hours in shift. With simulation modeling, it is determined that simulated fabrication time, \( F_{tsim} \), of initially suggested design solution for selected characteristic input data, takes approximately 20% longer time than minimally required fabrication time, \( F_{tmin} \):

\[
F_{tsim} = 1.2 \cdot F_{tmin} \quad (3)
\]

Above does not comply with project goal. Therefore, suggested design solution has to be further analyzed to determine the cause. Major topics of further analysis are as follows: material flow analysis; production line loads analysis and identification of potential line bottlenecks; identification of the most influence line parameters on the goal function with the sensitivity analysis method. Within sensitivity analysis method, line characteristics where changed within range of 10% over the initial input values, and all scenario combinations were simulated. In particular, parameters changed were a robot cutting head speed; crane movement and lifting speed; blasting station speed; the size of buffers and speed of conveyers. Range of parameters variations were defined according to shipyard expert’s survey method. It has been identified that primary bottleneck and most influence line element on the cutting time, of the suggested design solution is the performance of sorting crane (Area G on figure 4) which is sorting out cut profiles at the exit of the profile cutting robot station, (figure 5).
Legend: Sr – Robot cutting station; Ds – Sorting crane; Srub – Blasting station; M51 – Buffer 1; M53 – Buffer 3; Vt1 – Conveyer 1; Rt1 – Rotating conveyer; Vt2 – conveyer 2; Vt3 – conveyer 3; Vt4 – conveyer 4

**Fig. 5** Results of sensitivity analysis on the total process duration time depending on the changing values of line elements

Since the operation performance of that crane is insufficient, robot cutting station is blocked more than 35% of the time, which is unacceptable, and has to be improved. Therefore, more simulations of various production scenarios have been conducted, simultaneously varying crane and robot cutting stations parameters. Results are shown in table 4, where improved simulated fabrication time, $F_{tsimp}$, time is presented in comparison with targeted minimum fabrication and simulated fabrication time of initially suggested solution. Also, improvement over each is presented.

**Table 4** Fabrication time results and improvement against initially suggested solution

<table>
<thead>
<tr>
<th>Targeted min. fabrication time for required throughput, $T_{tmin}$</th>
<th>Simulated time of a initially suggested design solution, $F_{tsim}$</th>
<th>Improved simulated model fabrication time, $T_{tsimp}$</th>
<th>Improvement over the initially suggested design solution</th>
</tr>
</thead>
<tbody>
<tr>
<td>69 h</td>
<td>77 h</td>
<td>67 h</td>
<td>13 %</td>
</tr>
</tbody>
</table>

Where $T_{tmin}$ is calculated according to (2), based on following particular shipyards production data: $N_p$ – 2136 profile and flat bar parts; $T_{tmin}$ – 11000 profile and flat bars per month; $N_{wd}$ – 24 working days in a month; $N_s$ – 2 working shifts in a day; $N_{whs}$ – 7.5 working hours in a shift. From table 4, it is evident that improved simulation model solution, over the initially suggested design solution by the manufacturer, achieved improvement of 13%. Also, such improved solution met the objectives regarding the required monthly throughput, which the initial proposed solution was not achieving. Furthermore, suggested methodology provides the possibility to make comments and communicate to the manufacturer, regarded perceived issues, which would result in a decision that better meets the requirements of the particular shipyard. Otherwise, problems (regarding inadequate throughput performance) would have been noticed only after the installation of the line, when the changes would be extremely expensive, if at all feasible. Such improvement requirements would most probably increase the overall cost of investment, but this cost will be much lower than a cost potentially
emerging from repairing production line after installation or from using inadequate production line during the exploitation. In the following section, further enhancement of developed simulation model will be explained, based on comparisons with the actual process.

4.4 Simulation model confirmation, enhancement and further research

Simulation model based methodology and computer model itself, was tested against real production process data after project realization and implementation in the shipyard production process, (Figure 6). Based on comparison data with a real production process, computer simulation model was evaluated in terms of its representation of real production process. Certain differences, with real production process were perceived. For the several scenarios, using the four production mix samples of different ships and ship sections the real production process time was different, in average, for approximately 7.25%, (Table 5). In general product mix are consisted of sections from three different ships as follows: Product mix 1 is composed primarily from flat bars and profiles from product carrier double bottom section from ship mid section; Product mix 2 is composed from flat bars and profiles, also from product carrier ship type, but from consequent double bottom section from ship mid section; Product mix 3 is composed from flat bars and profiles, from product carrier ship type, mainly from double hull section from ship mid section. Product mix 4 is composed from flat bars and profiles, from asphalt carrier ship type, primarily from bottom section of ship mid section. Differences in measured times between product mixes are mainly because of different structure characteristics between those ships and used section within ship structure, as average profile thickness, profile dimensions, profile ending preparation, profile treatment type and characteristics, etc. Also, the size and type of product mixes was similar to initial one used for simulation modeling and are related to the particular shipyard sorting and assembly strategies.

<table>
<thead>
<tr>
<th>Product mix</th>
<th>Computer simulated time, min</th>
<th>Measured time in real production process, min</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Product mix 1</td>
<td>4130</td>
<td>4460</td>
<td>8%</td>
</tr>
<tr>
<td>Product mix 2</td>
<td>3850</td>
<td>4081</td>
<td>6%</td>
</tr>
<tr>
<td>Product mix 3</td>
<td>4528</td>
<td>4845</td>
<td>7%</td>
</tr>
<tr>
<td>Product mix 4</td>
<td>4601</td>
<td>4969</td>
<td>8%</td>
</tr>
<tr>
<td><strong>Average</strong>:</td>
<td></td>
<td></td>
<td><strong>7.25%</strong></td>
</tr>
</tbody>
</table>
Differences, presented in table 5, should be assessed taking into account the following facts: actual measured processing time includes failures, maintenance, human factors; it was determine that the specific parameters of the robot cutting head regarding positioning of the robot cutting head, marking and cutting, are different from what they were used initially in the simulation model. Therefore, the computer simulation model was enhanced including the following: failures were included based on gathered real process statistical data within several months; robot characteristic values were updated (based on real production data); maintenance intervals were included based on manufactured requirements. After such model enhancement, the model was again tested based on previously used several different production mix scenarios. Difference between computer model simulation and real production time were now between 3-5%, depending on ship type, production sample mix used, etc, (Table 6)

<table>
<thead>
<tr>
<th>Product mix</th>
<th>Product mix 1</th>
<th>Product mix 2</th>
<th>Product mix 3</th>
<th>Product mix 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sim. time diff.</td>
<td>+5%</td>
<td>-2%</td>
<td>+3%</td>
<td>-3%</td>
</tr>
</tbody>
</table>

Still persisting difference can be discussed because certain human factor issues, unexpected failures or jams are not included in simulation model and which can be included in the model as statistical and probability variables, based on gathered data from production process over the longer period of time. Furthermore, there is still some fine tuning that can be done regarding robot specifications, sorting crane specifications and sorting pallet specifications. Still, this version of model was accepted as sufficiently accurate, because, as stated before, it is not purpose of computer simulated model to be exact perfect copy of the real system but to be adequate, logic, technologically accurate and sufficiently precise presentation of a real system. Furthermore such model also could be used for: planning and evaluation of required working hours for certain production mix; spotting and predicting problems and bottlenecks in production before they occur; continuous measurement and analysis of the production process depending on various conditions so that he could continually be adapted and
improved, etc. Regarding model future application potentials, the author’s research is primarily continuing as to implement this computer model into shipyards CAD/CAM system, to integrate it with relevant software and production data to potentially enable real time functionality.

5. CONCLUSION

In this research, analysis of the existing methods and techniques for production process design has been conducted with emphasis on shipbuilding production process. Based on this analysis, the shortcomings of conventional approach and traditional mathematical modeling with analytic solution for complex production processes design have been perceived. Furthermore, regarding identified problems, suitability of discrete event simulation modeling method application for designing shipyard processes in particular, has been determined. Therefore, a methodology for designing shipbuilding production processes, based on discrete event simulation modeling method and selected operations research methods and tools, has been developed and presented. Developed methodology was applied and presented over the case study of designing the shipbuilding production process of robotized profile fabrication line. Design solution accomplished with suggested methodology, had fabrication time improved over the initially suggested design using conventional approach. Also, such solution fulfilled the objectives of the shipyard regarding the required monthly throughput, what the initial proposed solution was not. Furthermore, it is possible, during the early design stage, to make quality comments to the manufacturer, regarding production line setup and parameters, which will provide solution more adapted to the particular shipyard, which is much feasible and less expensive than making changes after line installation. Simulation model was tested against real production process data, after production line installation, and model was further enhanced and updated. Finally, the primarily contribution of developed methodology is improvement in shipbuilding production processes design practice. Such methodology provides shipyard management with efficient tool for validating design alternatives in early design stage, efficient tool for production process planning and control and also enables management to make decisions with lower risk level.

ACKNOWLEDGMENT

This work has been supported by the University of Rijeka under the project number 13.09.1.1.06.

REFERENCES


**NOMENCLATURE**

VT\textsubscript{1} – conveyer number 1
RT – rotating conveyer
VT\textsubscript{2} – conveyer number 2
MS\textsubscript{1} – buffer storage number 1
VT\textsubscript{3} – conveyer number 3
VT\textsubscript{4} – conveyer number 4
MS\textsubscript{2} – buffer storage number 2
VT\textsubscript{5} – conveyer number 5
MS\textsubscript{3} – buffer storage number 3
P\textsubscript{ca} – pallet for automatic subassemby line
P\textsubscript{cr} – pallet for robotic subassemby line
P\textsubscript{mp} – pallet for subassembly line
VT\textsubscript{6} – conveyer number 6
F_{\text{sim}} – simulated fabrication time of initially suggested design solution
F_{\text{min}} - minimum required fabrication time for chosen input production data
N\textsubscript{p} – number of profiles and flat bars in chosen product mix;
T_{\text{min}} – number of profiles and flat bars in targeted shipyard month production;
N_{\text{wd}} – number of working days in month;
N_{s} – number of working shifts in day;
N_{whs} – number of working hours in shift.
T_{\text{simp}} - improved fabrication time

Submitted: 16.03.2015.   Marko Hadjina, E-mail: hadjina@riteh.hr. Tel.: 051 651 449, Assistant Professor
Nikša Fafandjel, E-mail: niksaf@riteh.hr. Tel.: 051 651 455, Professor
Chair of Technology and Organization sector of the Naval Architecture and Ocean Engineering Department,
Tin Matulja, E-mail: tin.matulja@riteh.hr. Tel.: 051 651 454
Assistant Professor
Naval Architecture and Ocean Engineering Department, University of Rijeka – Faculty of Engineering, Vukovarska 58, 51000 Rijeka, Croatia

Accepted: 31.03.2015.
Table of contents Volume 66 Number 3 2015.

RESIDUAL HULL GIRDER ULTIMATE STRENGTH OF DOUBLE HULL OIL TANKERS (str.1-13)
Jerolim Andrić, Stanislav Kitarović, Karlo Pirić
Izvorni znanstveni članak

ANALYTICAL SOLUTION OF BASIC SHIP HYDROSTATICS INTEGRALS USING POLYNOMIAL RADIAL BASIS FUNCTIONS (str.15-37)
Dario Ban, Josip Bašić
Izvorni znanstveni članak

NEW ALGORITHM FOR OPTIMAL DIELECTRIC MATERIAL SELECTION IN MARINE ENVIRONMENT (str.39-48)
Igor Vujović, Zlatan Kulenović, Ivica Kuzmanić
Prethodno priopćenje

ANALYSIS OF THE ENERGY EFFICIENCY DESIGN INDEX WITH A PROPOSAL FOR IMPROVEMENT (str.49-59)
Predrag Čudina
Prethodno priopćenje

SWOT ANALYSIS OF DEFICIENCIES ON SHIP COMPONENTS IDENTIFIED BY PORT STATE CONTROL INSPECTIONS WITH THE AIM TO IMPROVE THE SAFETY OF MARITIME NAVIGATION (str.61-72)
Miroslav Randić, Dario Matika, Darko Možnik
Pregledni rad

CONSIDERING ANTI-PIRACY SHIP SECURITY: CITADEL DESIGN AND USE (str.75-90)
L. Carral, Carlos Fernández Garrido, José J. de Troya, José Ángel Fraguela
Stručni rad

AN ENHANCED EQUATION FOR VIBRATION PREDICTION OF NEW TYPES OF SHIPS (str.91-117)
Valter Cergol, Peter Vidmar
Izvorni znanstveni članak

[https://www.fsb.unizg.hr/brodogradnja/](https://www.fsb.unizg.hr/brodogradnja/)
Within the scope of the presented work, a hull girder ultimate strength analysis of the double hull oil tanker structures damaged by the collision or grounding is performed. An incremental-iterative progressive collapse analysis method prescribed by the forthcoming IACS Harmonized Common Structural Rules (H-CSR) is used for determination of the ultimate vertical bending moment and collapse sequence of the considered structures. Three characteristic variants of the oil tanker main frame cross sections of a different geometry and size (Aframax, Suezmax, and VLCC) are considered. The position of a ship’s side and/or bottom damage is defined in accordance with the IACS H-CSR. Proposed analytical formulations of the relationship between reduction of the hull girder ultimate vertical bending moment (with respect to the undamaged state) and damage size are based on the results of a systematic variation of a ship’s side or bottom damage size. Finally, comparison of the collapse sequences determined for the undamaged and damaged state in upright position (defined by IACS H-CSR) of the considered structure of the Aframax ship example is performed.

Key words: Damaged ships; hull girder ultimate strength; grounding; collision; double hull tanker structure, residual strength.

1. Introduction

A large number of ship accidents resulting in a loss of cargo, pollution of the environment and a loss of human life still occur, despite the advancements in ship design, production, and navigation procedures. Accident scenarios typically include collision, grounding, fire, and explosion. In that respect, it is of great importance to ensure acceptable safety level for ships damaged in those accidents. When faced with any of these accidental situations, the ship operator’s need to take rapid decisions regarding the salvage actions and further steps should be based on evaluation of the damage effects on the ships safety using the residual strength assessment procedure. Adequate hull girder strength in intact condition does not necessarily guarantee an acceptable safety margin in damaged conditions [1].
A draft of the IACS Harmonized Common Structural Rules (H-CSR) [2] has been released for the industry review in April 2013. In comparison to the IACS CSR currently in force [3] IACS H-CSR contains additional requirement regarding the residual strength of tankers and bulk carriers, i.e. the hull girder ultimate strength in prescribed damaged conditions. According to the IACS H-CSR, the residual strength is evaluated for the two specific accident scenarios: collision and grounding. A similar approach can be found in ABS Guide to accessing hull girder residual strength for tankers [4] but with the different request regarding extent of damage.

Among a number of the contemporary methods for the hull girder strength evaluation, various incremental-iterative progressive collapse analysis method based on Smith’s approach [5] are arguably the most widespread. Furthermore, rules of many classification societies, including IACS CSR and IACS H-CSR, prescribe utilization of incremental-iterative procedures based on Smith’s approach for evaluation of the longitudinal ultimate load-capacity of ship structures both in intact and damaged condition. Overview of various existing methods for the hull girder ultimate strength calculation in intact condition can be found in [6-10], while the critical review of their accuracy can be found in [11, 12]. Recently, the residual hull girder strength has been investigated through two different approaches: nonlinear FEM [13, 14 and 15], and progressive collapse methods (PCM) based on Smith’s approach using incremental-iterative or pure incremental procedures [16-20].

When the cross section is asymmetrically damaged like in a way of collision damaged, the neutral axis (NA) rotates and the problem can be treated as biaxial bending problem. Recently several procedures have been suggested to include NA shift into account. Choung et al. [21] provided two convergence criteria to find translational and rotational locations of the neutral axis plane for intact and damaged vessels. Definition of three types of asymmetries of a ship section was proposed: material-, load-, and geometry-induced asymmetries. Concept of moment plane (MP) was introduced to define the heeling angle of ship section. It is suggested that force equilibrium and force vector equilibrium criteria are both necessary to determine new position of NA due to both translational and rotational shifts. Recently Fujikubo et al. [22] have suggested updated pure incremental method (PCM) to derive the biaxial bending moment-curvature relationship taking into account the rotation and translation of the neutral axis in asymmetrically damaged hull girders, while Makouei et al. [23] further tested the accuracy of method presented by Fujikubo. However this effect is much more expressed for a single skin bulk carrier structures compared to double hull tanker structures examined in the research performed by Fujikubo et al. [22].

Intention of the present study is to investigate the influence of the damage size on the ultimate hull girder capacity of oil tankers for the two characteristic types of accidents: collision and grounding. Proposed analytical formulations of the relationship between the reduction of the hull girder ultimate bending moment (with respect to the undamaged state) and damage size are based on the analysis of the results of a systematic variation of damage extent of ship’s side or bottom.

2. Capacity models of considered hull girder structures

Three characteristic variants of the double hull oil tanker midship sections of a different geometry and size (Aframax, Suezmax and VLCC) are considered. All examined structures are designed according to the pre-CSR requirements of different classification societies. The main particulars of the tanker structures considered by this study are given in Table 1. Examined structures denoted as models M2 and M3 (Suezmax and VLCC tanker) belong to the standard set of the ISSC benchmark examples and all relevant data regarding their material and geometric properties are given in Technical Committee IV.2. [9],and Technical
Committee III.1 [8]. Figures 1-3 illustrates one-bay structural models at midship section of all considered structures in intact condition. Structural model definition, essential for all ultimate bending capacity calculations performed by the co-authors for the purposes of the present paper is done using the computer program MAESTRO [24]. For all models no corrosion deduction has been implemented, so as-built scantlings were used for the study.

Table 1 Main particulars of the examined ships

<table>
<thead>
<tr>
<th></th>
<th>M1 - Aframax tanker</th>
<th>M2 - Suezmax tanker</th>
<th>M3 - VLCC tanker</th>
</tr>
</thead>
<tbody>
<tr>
<td>(L_{BP}) (m)</td>
<td>235</td>
<td>265</td>
<td>320</td>
</tr>
<tr>
<td>(B) (m)</td>
<td>42</td>
<td>46.4</td>
<td>58</td>
</tr>
<tr>
<td>(D) (m)</td>
<td>21</td>
<td>23.2</td>
<td>30</td>
</tr>
<tr>
<td>(C_B) (-)</td>
<td>0.86</td>
<td>0.83</td>
<td>0.82</td>
</tr>
</tbody>
</table>

3. Damage scenarios

The damage due to grounding and collision are the most common reasons of the destruction of ship structures. Ship to ship collision causes the bow of the striking ship to collapse and the side of the struck ship to be damaged. It is the most destructive among all possible damages. Ship grounding on rock(s) results in a cutting or crushing of the bow
bottom [20]. The basic definition of the damage extent in this study was performed according to IACS H-CSR [2] and a specified extent of damage for collision and grounding type of accident is illustrated by Figures 4 and 5, respectively. The hull girder ultimate bending capacity with the specified damage extents is to be checked.

![Diagram of damage extent](image)

**Fig. 4** Damage extent for collision specified by IACS H-CSR [2].

![Diagram of damage extent](image)

**Fig. 5** Damage extent for grounding specified by IACS H-CSR [2].

As stated by Notaro et al. [13] the ultimate capacity in damaged condition is not largely influenced by the shape and the longitudinal extension of the damage. The main factor leading the capacity reduction is the vertical and transversal extent of the damage. With respect to those remarks the performed systematic variation of a damage size is based on the following principles:

- For the collision case depth of the damage penetration is kept constant \((d=B/16)\), as specified by the Rules, while the damage penetration height \(h\) is systematically varied from \(0.1D\) to \(0.8D\), with the step of \(0.1D\). For this case the damage is on one side only and located immediately below the freeboard deck;

- For the grounding case height of the damage penetration is kept constant \((h=\min(B/15, 2))\) as specified by the Rules, while the damage penetration breadth \(b\) is systematically varied from \(0.1B\) to \(0.8B\), with the step of \(0.1B\). For this case the damage is considered to be located symmetrically from the CL on PS and SB side.

Nine different models were generated for the each of three tankers (eight damaged and one intact) and used for each damage case. Several examples of a damaged ship models are presented in Figures 6 and 7 for the collision and grounding case, respectively.
4. Hull girder ultimate strength results

Imminent occurrence of the inter-frame collapse prior to any other feasible global collapse mode ensures that the global structural behaviour of the complex monotonous thin-walled structures submitted to flexure can be idealized in accordance with the beam bending theory during the whole collapse process. This implication represents the fundamental premise of the Smith’s method [5] which is considered to be the first among established progressive collapse analysis methods that incorporate more sophisticated consideration of the structural collapse sequence and structural post-critical response of structural elements. Development of the original method subsequently stimulated proposition of various methods based on Smith’s approach [16, 25, 26]. In shipbuilding practice, rules of many classification societies and their associations [2, 3] prescribe utilization of the incremental-iterative procedures based on Smith’s approach for evaluation of longitudinal ultimate capacity during the structural design synthesis. The ultimate vertical bending moment capacities of the hull girder transverse section, in hogging and sagging conditions, are defined as the maximum values of the curves of the vertical bending moment capacity versus the curvature $\chi$ of the transverse section considered. The curve is obtained through an incremental-iterative approach. Within the framework of this paper, IACS incremental-iterative progressive collapse analysis method is employed, as previously implemented within OCTOPUS [27] computer program. In performed calculations several assumptions were made:

- Calculation procedure for the vertical ultimate bending moment capacities of a damaged section is same as for the intact condition and follows recommendations given in IACS;
• Damaged area, as defined in Ch. 3, carries no loads and is therefore removed from the models;
• Only vertical bending is considered. The effects of the shear force, torsion loading, horizontal bending moment and lateral pressure are neglected;
• The ultimate bending capacity of the damaged transverse cross section is calculated with the model kept in upright position and a neutral axis rotation is not considered. Implication of that assumption, and possibly error can be advocate to be below 10 % due to fact that the same method is defined in H-CRS [2] with prescribed safety factor equal to \(C_{NA}=1.1\) for collision and \(C_{NA}=1.0\) for grounding case. Some suggestions regarding the inclusion of the neutral axis rotation are given in Choung et al. [21] and Fujikubo et al. [22]). Fujikubo et al. [22] also reported that influence of NA rotation on hull girder ultimate strength is very low for tankers in collision case. For bulk carriers the effect is more significant due to single side shell structure and around 8% reduction of ultimate strength can be expected for 70% side damage.

In this study the residual strength index (\(RIF\)), originally introduced by Fang and Das [28] and used by Hussein and Soares [20], as a way to compare the ultimate strength capacity of the damaged hull (\(M_{U,\text{Damage}}\)) with the intact one (\(M_{U,\text{Intact}}\)), is used to systematically investigate the relationship between the ultimate strength capacity and a damage size:

\[
RIF = \frac{M_{U,\text{Damage}}}{M_{U,\text{Intact}}} \tag{1}
\]

Similar approach can be used to compare other relevant sectional characteristics \((A, I_y, W_D, W_B)\) of the damaged and intact hull girder cross sections:

\[
RIF\_A = \frac{A_{\text{Damage}}}{A_{\text{Intact}}}; RIF\_I_y = \frac{I_{y,\text{Damage}}}{I_{y,\text{Intact}}}; RIF\_W_D = \frac{W_{D,\text{Damage}}}{W_{D,\text{Intact}}}; RIF\_W_B = \frac{W_{B,\text{Damage}}}{W_{B,\text{Intact}}} \tag{2}
\]

where \(A_{\text{Damage}}\) and \(A_{\text{Intact}}\) are cross sectional area in damaged and intact condition, respectively; \(I_{y,\text{Damage}}\) and \(I_{y,\text{Intact}}\) are vertical moments of inertia for cross sections in damaged and intact condition, respectively; \(W_{B,\text{Damage}}\) and \(W_{B,\text{Intact}}\) are bottom sectional modulus in damaged and intact condition, respectively; \(W_{D,\text{Damage}}\) and \(W_{D,\text{Intact}}\) are deck sectional modulus in damaged and intact condition, respectively.

4.1. Grounding case

Summary of the obtained results for the grounding case is given in Table 2.

Damage ratio \(\lambda\) for grounding has been specified as breadth of damage area \((b_{\text{damage}})\) divided by the breadth of the ship \((B)\), see Table 2.

From the presented results it can be noted that the reduction of the hull girder ultimate bending moment, expressed through the \(RIF\), is larger in the hogging than in the sagging case for all evaluated tankers. Data given in Table 2 enable easy establishment of the dependence between the reduction of the cross sectional characteristics \((RIF\_A, I_y, W_D, W_B)\) and \(RIF\).

For example, a damage size ratio of \(\lambda = 0.6\) in the grounding case (specified by the IACS (2014) as the requested damage value), cause average reduction of the cross section area by 13.9%. At the same time, the ultimate hogging and sagging moments are reduced in average (for all three models) by 19.7% and 8.3%, respectively. Graphical presentation of the relationship between \(RIF\) and a damage size ratio is presented in Figure 8. From the data
presented in Table 2 and Figure 8, a linear equations are proposed to describe the relationship between the $RIF$ and a damage size ratio ($\lambda = \frac{b_{\text{damage}}}{B}$):

$$RIF_{\text{grounding-SAGG}} = 1.008 - 0.158\lambda$$  \hspace{1cm} (3)

$$RIF_{\text{grounding-HOGG}} = 1.006 - 0.341\lambda$$  \hspace{1cm} (4)

Hussein and Guedes Soares published [15] a similar research and specified a unique expression for the double hull oil tanker structure:

$$RIF_{\text{grounding}} = 1.02 - 0.254\lambda$$  \hspace{1cm} (5)

<table>
<thead>
<tr>
<th>Damage ratio: $\lambda_{\text{damage}}/B$</th>
<th>M1-Aframax tanker</th>
<th>M2-Suezmax tanker</th>
<th>M3-VLCC tanker</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>0.1</td>
<td>0.988</td>
<td>0.966</td>
<td>0.973</td>
</tr>
<tr>
<td>0.2</td>
<td>0.977</td>
<td>0.934</td>
<td>0.952</td>
</tr>
<tr>
<td>0.3</td>
<td>0.966</td>
<td>0.907</td>
<td>0.931</td>
</tr>
<tr>
<td>0.4</td>
<td>0.951</td>
<td>0.867</td>
<td>0.907</td>
</tr>
<tr>
<td>0.5</td>
<td>0.933</td>
<td>0.829</td>
<td>0.877</td>
</tr>
<tr>
<td>0.6(H-CSR request)</td>
<td>0.910</td>
<td>0.789</td>
<td>0.853</td>
</tr>
<tr>
<td>0.7</td>
<td>0.891</td>
<td>0.760</td>
<td>0.833</td>
</tr>
<tr>
<td>0.8</td>
<td>0.859</td>
<td>0.713</td>
<td>0.800</td>
</tr>
<tr>
<td>0.9</td>
<td>0.836</td>
<td>0.682</td>
<td>0.767</td>
</tr>
<tr>
<td>0.6(H-CSR request)</td>
<td>0.922</td>
<td>0.797</td>
<td>0.860</td>
</tr>
<tr>
<td>0.7</td>
<td>0.902</td>
<td>0.757</td>
<td>0.838</td>
</tr>
<tr>
<td>0.8</td>
<td>0.874</td>
<td>0.707</td>
<td>0.807</td>
</tr>
<tr>
<td>0.9</td>
<td>0.849</td>
<td>0.668</td>
<td>0.774</td>
</tr>
<tr>
<td>0.6(H-CSR request)</td>
<td>0.920</td>
<td>0.823</td>
<td>0.870</td>
</tr>
<tr>
<td>0.7</td>
<td>0.899</td>
<td>0.792</td>
<td>0.850</td>
</tr>
<tr>
<td>0.8</td>
<td>0.876</td>
<td>0.757</td>
<td>0.824</td>
</tr>
</tbody>
</table>

Collapse sequences in hogging and sagging are analyzed in detail for undamaged and damaged case ($\lambda = 0.6$) for all three examined cross sections. Vertical bending moment capacity versus the curvature $\chi$ curve is presented for the undamaged and damaged conditions for the Aframax tanker model in hogging, see Figure 9, as an example.
Fig. 8 RIF for grounding in sagging and hogging cases.

Fig. 9 Collapse sequences of Aframax tanker in grounding, hogging case.
Due to the reduced cross section, it can be noted that the damaged section has reduced bending stiffness and reaches the ultimate bending capability faster than the undamaged section. Also, the damaged section reaches the ultimate bending capacity at the lower curvature compared to the undamaged section.

Due to the ineffectiveness of the damaged bottom plating, which does not contribute to the bending stiffness of the cross section, the inner bottom plating is imposed with the higher compressive load. When inner bottom structure collapses due to buckling, the damaged section reaches the ultimate bending capacity. It can be noted that the undamaged section reached its ultimate bending capacity just after the bottom plating collapsed, but without the collapse of the inner bottom plating. Furthermore, it can be also noticed that the deck structure is the structural part that collapses first, due to the high tensile stresses in both cases.

In-house software OCTOPUS [27] used in this study enables identification of the characteristic structural collapse sequence accounting for the load-shedding effect during the progressive load incrementation. This capability can enable determination of more rational distributions of the longitudinally effective material within the process of concept design synthesis, i.e. during the consideration of various topologic variants and/or materially-geometrical properties of the feasible structural cross-sections, since it can point to the more efficient ways of required structural safety level accomplishment. Furthermore, collapse sequence can also be considered as a rational pathfinder during the material reduction process of the initially over-dimensioned cross section (for the case of structural safety criteria over-satisfaction).

4.2. Collision case

Summary of the obtained results for the collision case are given in Table 3.

Damage ratio \( \lambda \) for the collision is specified as the height of the damage area \( (h_{\text{damage}}) \) divided by the depth of the ship \( (D) \), see Table 3.

From the presented results it can be noted that the reduction of the hull girder ultimate bending moment expressed through residual \( RIF \) is larger in sagging than in hogging case for all evaluated tankers. This is the opposite trend with respect to the findings obtained for the grounding case.

Case with damage size ratio of \( \lambda = 0.6 \) (specified by the IACS H-CSRR [2] as requested damage value), causes an average reduction of the cross sectional area by 11.5\%. At the same time, the ultimate hogging and sagging moments are reduced in average (for all three models) by 12.2\% and 18.8\%, respectively.

Graphical presentation of the relationship between the \( RIF \) and a damage size ratio is presented in Figure 10.

From the data presented in Table 3 and Figure 10, a linear equations can be used to represent the relationship between the \( RIF \) and a damage size ratio \( \lambda = h_{\text{damage}}/D \):  

\[
RIF_{\text{collision-SAGG}} = 0.9927 - 0.5802\lambda + 0.4516\lambda^2 \tag{6}
\]

\[
RIF_{\text{collision-HOOGG}} = 0.9948 - 0.3494\lambda + 0.2544\lambda^2 \tag{7}
\]

In [20], Hussein and Guedes Soares proposed a unique expression for the double hull oil tankers:

\[
RIF_{\text{collision}} = 0.98 - 0.084\lambda \tag{8}
\]
Table 3 Residual strength indices for collision.

<table>
<thead>
<tr>
<th>Damage ratio:</th>
<th>M1-Aframax tanker</th>
<th>M2-Suezmax tanker</th>
<th>M3-VLCC tanker</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \lambda = \frac{h_{\text{damaged}}}{D} )</td>
<td>RIF( _M )-sagg</td>
<td>RIF( _M )-hogg</td>
<td>RIF( _A )</td>
</tr>
<tr>
<td>----------------</td>
<td>-------------------</td>
<td>-------------------</td>
<td>----------</td>
</tr>
<tr>
<td>0</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>0.1</td>
<td>0.925</td>
<td>0.951</td>
<td>0.973</td>
</tr>
<tr>
<td>0.2</td>
<td>0.895</td>
<td>0.932</td>
<td>0.958</td>
</tr>
<tr>
<td>0.3</td>
<td>0.848</td>
<td>0.907</td>
<td>0.936</td>
</tr>
<tr>
<td>0.4</td>
<td>0.827</td>
<td>0.894</td>
<td>0.921</td>
</tr>
<tr>
<td>0.5</td>
<td>0.810</td>
<td>0.882</td>
<td>0.902</td>
</tr>
</tbody>
</table>

Collapse sequences in hogging and sagging are analysed in detail for the undamaged and damaged cases (\( \lambda = 0.6 \)), for all three examined cross sections. Vertical bending moment capacity versus the curvature \( \chi \) curves are presented for the undamaged and damaged conditions for Aframax tanker model in sagging, see Figure 11, as an example.

A similar collapse sequences are identified for the damaged and undamaged conditions in the hogging and sagging case. The critical structural part which collapses first is the deck and after the part of the side structure (outer and inner) collapsed, the cross section reached its ultimate bending moment capacity.
Residual hull girder ultimate strength of a double hull oil tankers  J. Andrić, S. Kitarović, K. Pirić

$y = 0.4516x^2 - 0.5802x + 0.9927$
$R^2 = 0.9936$

$y = 0.2544x^2 - 0.3493x + 0.9948$
$R^2 = 0.994$

**Fig. 10** RIF for collision in sagging and hogging case.

**Fig. 11** Collapse sequences of Aframax tanker in collision, sagging case.
5. Conclusions

Intention of the present study was to investigate the influence of the damage size on the ultimate hull girder capacity of the oil tankers for the two characteristic types of accidents: collision and grounding, using an IACS incremental-iterative progressive collapse analysis method.

Influence of the different ship size, structural configuration and damage extent (size and location) in collision and grounding on the hull girder residual ultimate strength has been systematically investigated. Analytical formulations of the relationship between reductions of the hull girder ultimate bending moment (with respect to the undamaged state) and a damage size ratio has been proposed based on the analysis of the results of a systematic variation of damage extent of ship’s side or bottom. Those design equations and associated diagrams can be used for the rapid assessment of the hull girder residual ultimate strength and give first basis for the emergency situation decision making.

In-house software used in this study enables identification of the characteristic structural collapse sequence and can be used for determination of more rational distributions of the longitudinally effective material within the design process.

Future investigation will go a step further with respect to the extension of the employed progressive collapse analysis method regarding the possibility to calculate vertical and horizontal ultimate bending moments and to enable rotation of the cross sectional neutral axis in damaged conditions.

Acknowledgements

This work has been supported in part by Croatian Science Foundation under the project 8658. Thanks are due to our diligent students Ana Zarko and Maja Plavsic for the time and effort spent on realization of the considerable portion of the structural models and numerical simulations performed within the scope of work presented by this article.

REFERENCES

Residual hull girder ultimate strength of a double hull oil tankers
J. Andrić, S. Kitarović, K. Pirić

[27] …: OCTOPUS Software Documentation, University of Zagreb, FSB., Zagreb, 2014.

Submitted: 25.03.2015. Jerolim Andrić, jerolim.andric@fsb.hr
Stanislav Kitarović
Accepted: 15.08.2015. Karlo Pirić
University of Zagreb, Faculty of Mechanical Engineering and Naval Architecture, I. Lučića 5, 10000 Zagreb, Croatia
ANALYTICAL SOLUTION OF BASIC SHIP HYDROSTATICS INTEGRALS USING POLYNOMIAL RADIAL BASIS FUNCTIONS

UDC 629.5.015.1
Original scientific paper

Summary

One of the main tasks of ship's computational geometry is calculation of basic integrals of ship's hydrostatics. In order to enable direct computation of those integrals it is necessary to describe geometry using analytical methods, like description using radial basis functions (RBF) with $L_1$ norm. Moreover, using the composition of cubic and linear Polynomial radial basis functions, it is possible to give analytical solution of general global 2D description of ship geometry with discontinuities in the form of polynomials, thus enabling direct calculation of basic integrals of ship hydrostatics.

Key words: analytical, solution, integrals, intersection, ship, hydrostatics, polynomial, RBF, $L_1$ norm

1 Introduction

From the beginning of naval architecture as science, in the works of its founders Chapman, [1] and Euler, [2] and others, there have been efforts for analytical description of ship geometry using polynomials in order to enable direct, analytical solution of ship's computational geometry problems, like direct solving of intersection problem or direct solving of basic hydrostatic integrals. At the beginning of 20th century, Taylor, [3], described ship's hydrostatic and geometric particulars approximating them with polynomials, thus showing the possibility of their usage in ship computational geometry calculations. Nevertheless, his method did not solve all computational geometry problems like solving belonging geometric and hydrostatic integrals or intersection problem, with possible multiple roots over single ordered pair of function domain values or not finding solutions for nearly parallel lines. That has not been achieved until the advances in the computer technology in recent decades that results in the development of meshless methods based on piecewise radial basis functions.

In dissertation, [4], and paper, [5], it is shown that there exist global solution of description problem of 2D ship geometry with discontinuities using composition of cubic and linear Polynomial radial basis functions with dense description of discontinuities. Except very high accuracy, this solution gives possibility of direct solution of basic integrals of ship
hydrostatics expanded for the integral for the determination of wetted area of immersed part of the ship's hull, which is much faster and more accurate than numerical integration methods.

In order to show the efficiency of the global 2D radial basis functions (RBFs) description method using Polynomial RBFs as the direct solution of basic ship hydrostatic integrals, two test frames are chosen: test frame of car-truck carrier with flat side and camber, and test frame in the shape of semicircle. Former is frame section with discontinuities which is complex from the ship's computational geometry point of view, and latter is theoretical frame section that enables analytical checking of integration results. Those test frames will enable the testing of novel calculation method using Polynomial RBFs, in the calculation of hydrostatic particulars for actual ship frame with discontinuities, and for the calculation of hydrostatic particulars of theoretical frame section in the shape of semicircle.

2 Calculation of ship hydrostatics particulars

There are five basic integrals defined for the calculation of ship hydrostatic particulars, with the assumption of using 2D description methods. Those integrals are all univariant definite integrals whose upper limit depends on the actual waterline position, defined for one of the coordinate axis of the ship coordinate system, Figure 1.

According to the hydrostatic values to be calculated, they can be divided to integrals for the calculation of actual waterline particulars and actual displacement properties.

The ship waterline particulars and their belonging integrals to be determined are:

- Waterplane area $A_{WL}$,
- The centroid of waterplane area $X_{WL} \equiv \{ x_{WL}, y_{WL} \}$, and
- The moment of inertia of waterplane $I_{WL} \equiv \{ I_L, I_B \}$.

The displacement particulars and their belonging integrals are:

- The volume of displacement $\nabla$, and
- The centre of buoyancy $X_B \equiv \{ x_B, y_B, z_B \}$.
After obtaining the results of frame section areas and belonging static moments around y axis, it is necessary to integrate along remaining x axis in order to obtain ship displacement particulars values.

Respective integrals for the calculation of Bonjean curves correspond to the waterplane area integrals with:

- Section area $A$, and
- Section area moment around y axis, $M_y$.

In order to calculate the centre of waterline area, it is necessary to determine belonging area moments as:

$$M_{WL} = A_{WL} \cdot X_{WL}, \quad M_{WL} \equiv \{M_{WL,x}, M_{WL,y}\}$$

Respective displacement volume moments for the calculation of centre of buoyancy are:

$$M_B = \nabla \cdot X_B, \quad M_B \equiv \{M_{B,x}, M_{B,y}, M_{B,z}\}$$

In the case of 3D description of ship geometry, the formulae for the calculation of volume displacement $\nabla$ for some predefined waterline, and known geometry description $y = f(x, z)$ using double integral definition, is:

$$\nabla = \int_{x_1, z_1}^{x_2, z_2(x)} f(x, z) \cdot dx \cdot dz$$

Where $x_1, x_2, z_1(x), z_2(x)$ are integration limits defined by the intersection of actual waterline with ship geometry and ship bottom line.

Accordingly, belonging moment $M_B$ can be also determined using double integral as:

$$M_B = \int_{x_1, z_1(x)}^{x_2, z_2(x)} f_M(x, z) \cdot dx \cdot dz$$

Nevertheless, the 3D RBF description of ship geometry is not available yet, in the form suitable for direct integration of ship hydrostatic particulars. Therefore, only 2D Polynomial RBF direct integration methods will be shown in this paper.

Except above mentioned, the surface area of the immersed ship's hull can be also included in basic hydrostatic particulars, with the calculation of:

- Wetted area $S_W$.

When 2D calculation methods are used, it is first necessary to calculate:

- Frame section curve lengths $L_W$.

After that, it is necessary to integrate along remaining x axis.

In order to solve above stated integrals it is necessary to determine their limits first, by calculating the intersection of some arbitrary waterline with ship geometry, both described by computationally compatible RBF methods that will be shown in the next paragraph.
2.1 Five basic integrals of ship hydrostatics

There are five basic integrals to be solved for the determination of ship's hydrostatics particulars. When defined for general coordinate $x$ and function description $f(x)$, those definite integrals are:

1. $\int_{x_1}^{x_2} f(x) \, dx$  \hspace{1cm} (1)
2. $\int_{x_1}^{x_2} xf(x) \, dx$ \hspace{1cm} (2)
3. $\int_{x_1}^{x_2} [f(x)]^2 \, dx$ \hspace{1cm} (3)
4. $\int_{x_1}^{x_2} [f(x)]^3 \, dx$ \hspace{1cm} (4)
5. $\int_{x_1}^{x_2} x^2 f(x) \, dx$ \hspace{1cm} (5)

The limits of above integrals are defined for the keel of the ship, $x_1$, and the intersection of frame section curve with actual ship waterline, $x_2$.

The descriptions of the ship geometry in general case, contain translated points by elastic shift method and can be rotated for some angle $\varphi$, [1]. Therefore, general definitions of above integrals contain translation corrections and rotation terms, as will be shown in further papers.

2.2 Curve length integral

The curve length integral for 2D calculation of wetted area of the ship is:

$$\int_C \sqrt{d^2 x + d^2 y} = \int_x \sqrt{1 + y'^2} \, dx$$

Using function description $f(x)$:

$$\int_{x_1}^{x_2} \sqrt{1 + [f'(x)]^2} \, dx$$

More generally, basic integral to be solve is:

6. $\int_{x_1}^{x_2} \sqrt{1 + g(x)} \, dx$ \hspace{1cm} (6)
3 Polynomial Radial Basis Functions

3.1 RBF Definition

The main reason for using analytical curve description using functions is to enable direct 2D and 3D integration, as well as solving intersection problem. The possibility of solving integrals for belonging functional curve description depends on computational characteristics of function chosen. In the case of radial basis functions, except their basis type, there is additional problem of their definition with norm as function argument.

Radial basis function networks are generally defined as the linear combination of basis functions, which depend on $L_2$ norm, i.e. the distance $\|x - t\|$ between input data set points, $x$, and the points of centres, $t$, around which the function is developed. Therefore, RBFs as direct feed-forward neural networks, Figure 2, can be represented in the form:

$$y = f(x) = \sum_{i=1}^{O} w_i \Phi_i(\|x - t_i\|), x \in \Omega \subseteq IR^d, y \in IR^l$$

where $\Phi_i$ is radial basis function, $x$ is input variables data points set, $y$ is output variables data points set, $t$ is the set of $O$ centres radial basis functions are developed for, $w$ is the matrix of the weight coefficients, $d$ is the dimension of input data set, and $l$ is the dimension of output data set.

![Figure 2 Single-layered, Feed-forward RBF Neural Network](image)

The main advantage of RBFs, and the reason why they are widely used, is that they are the solution of scattered data interpolation problem, where the number of centres equal the number of input points with $O = N$, i.e. $t = x$. The solution of above interpolation problem can be obtained by determination of weight coefficient vector/matrix $w$, using inversion of interpolating matrix $H$ as:

$$w = H^{-1} \cdot y$$

The problem of the curve or some surface reconstruction is the problem of the determination of the function $f$ for curve description, based on the input data set $X \equiv \{x_j, j = 1, \ldots, N\}$ and output data set points $Y \equiv \{y_j, j = 1, \ldots, N\}$, on the range $[a, b]$. Overall data set is then divided in two sets, one data set $X$ for RBF calculation and one set $X_G$ for generalization of the description, with $X_G \equiv \{x_j, j = 1, \ldots, N_G\}$.
3.2 Polynomial RBF definition

Although widely used, none of standard RBFs defined with $L_2$ norm used for 3D ship’s geometry description are not twice integrable for their argument $\|x - x_i\|$. Besides, it is not possible to solve intersection problem directly for 3D case using $L_2$ norm, [4]. That is one of the reasons for choosing 2D ship geometry description using Polynomial RBFs with $L_1$ norm and argument $|x - x_i|$.

Polynomial radial basis functions for 2D problems can be generally defined as:

$$f(x) = \sum_{j=1}^{N} w_j |x - t_j|^\beta + c, \quad \beta \in IR \setminus 2 \cdot IN$$  \hspace{1cm} (8)

With shape parameter $c$ set outside weighted sum, $c \in IR$, and function exponent $\beta$ defined in the whole space real numbers $IR$ restricted for even integers.

Moreover, it is shown in [4] and [5] that it is possible to solve general 2D description problem of ship geometry with discontinuities using composition of cubic and linear Polynomial RBFs with $L_1$ norm and dense points around discontinuities.

Thus Polynomial RBFs with $L_1$ norm have simple form with odd integer exponents that enable direct integral solutions.

3.3 Polynomial RBFs in general polynomial form

Above mentioned Polynomial RBFs (PRBF) can be defined for general polynomial basis with polynomial coefficients as:

$$f(x) = \sum_{i=1}^{n} C_i x^i$$  \hspace{1cm} (9)

where $C_i, i = 1, \ldots, n$ are polynomial coefficients.

Regarding polynomial degree $n$ of developed Polynomial RBF, it is equal to the function exponent $\beta$, i.e.:

$$n = \beta$$  \hspace{1cm} (10)

Due to limitations of contemporary mathematics, the polynomial degree $n$ is limited to direct univariate polynomial solutions with $n \leq 6$, as will be described in chapter 5 of this paper.

The integration of polynomials is the simplest possible, and the integration of cubic Polynomial radial basis functions will be shown in this paper. Moreover, the solutions of polynomial roots exist also, for some polynomial degrees as will be described later in the text.

4 Precision of the description methods

Except integrability and solvability of the basis function, it is necessary to ensure high precision of the ship geometry description, in order to enable integration procedure.

Global and local measures of the accuracy of the description are defined as RMSE and $\text{Err}_{\text{max}}$, for chosen number of input points $N$. Global error is defined as:
Analytical solution of basic ship hydrostatics integrals using polynomial radial basis functions

Dario BAN
Josip BAŠIĆ

$$\text{RMSE} = \sqrt{\frac{\sum_{i=1}^{N} (f(x_i) - y_i)^2}{N}}$$

Where $N$ is the number of input data set $y_i$, $i = 1, \ldots, N$ is the output data set of points, and $f(x)$ is radial basis function.

Corresponding local error is defined as:

$$\text{Err}_\text{max} = \max \{ \text{Err}_i \}, \text{Err}_i = |y_i - f(x_i)|, i = 1, \ldots, N_G$$

Where $N_G$ is the number of the points used for generalization of the description.

The problem with precision usually occurs for the integration of the interpolating functions with small global precision of the description, where the effect of error grouping in a point occur, shown on Figure 3 for calculation of vertical centre of buoyancy $z_B$, below, with low Root Mean Square Error (RMSE) value.

![Graph](image)

**Figure 3** Error grouping in a point with low precision integration of RMSE = $4.16 \cdot 10^{-2}$

This problem is one of the main reasons why approximation methods, like decomposition methods, fast computing methods and the development into series methods, or approximation methods in general, are not suitable in computational geometry for the calculation of ship hydrostatics particulars using 2D methods.

Opposite to above, Polynomial radial basis functions ensure high precision of the description, even for frame sections with discontinuities. For example, global accuracy of the description for Test frame No. 1 using composition of Polynomial RBFs with $\beta = \{3, 1\}$ is RMSE = $3.5 \cdot 10^{-10}$, while local accuracy is $\text{Err}_\text{max} = 1.168 \cdot 10^{-6}$ (m), as shown on Figure 4.
It is obvious that the accuracy of the description of Test frame No. 1 using composition of Polynomial RBFs with $\beta = \{3, 1\}$ is very high, much higher than required $10^{-4}$ (m).

5 Solution of the intersection problem

One of basic computational geometry tasks, together with solving basic hydrostatic integrals, is solving the problem of ship geometry intersection with actual waterline. In order to enable direct solution of this problem, both ship geometry and surrounding waterplane should be described in the same manner, analytically. The method that ensures such solution is polynomial description of ship geometry with corresponding waterplane, using Polynomial RBFs with $L_1$ norm.

It is known from the theory of polynomials that direct solution of roots of general polynomials are available for degrees lower than seven, only, [6]. Also, their direct solutions always exist for polynomials with exponents lower and including four, and those solutions can be obtained by basic arithmetic operations, adding, subtracting, multiplication and division, according to Abel’s theorem, [7]. The roots of the polynomials for general quintic and sextic cannot always be obtained, and their available solutions are given using hypergeometric functions. The solution of septic equation is not always available too, and when it is, it can be solved using Galois groups, [6], that require hyperelliptic functions for their solution, or by superimposing continuous functions of two variables.

Above degree restrictions of directly solvable polynomials is the reason for the limitation of the polynomial degree for ship geometry description, and therefore it is set to:

$$n \leq 4$$  \hspace{1cm} (11)
Another restriction for polynomial degree comes from the definition of polynomial RBF where only odd integer values are allowed:

$$\beta = 1, 3, 5, \ldots$$ \hspace{1cm} (12)

In order to produce smooth curve, Polynomial RBFs must be at least twice integrable, thus giving yet another restriction with:

$$\beta \geq 3$$ \hspace{1cm} (13)

As mentioned before, belonging polynomial degree $n$ of developed Polynomial RBF is equal to the function exponent $\beta$. It can be concluded from above restrictions that the only Polynomial RBF exponent $\beta$ that satisfies all three requirements is:

$$\beta = 3$$ \hspace{1cm} (14)

As stated before, the solution of the global description of ship geometry with discontinuities is given using cubic-linear Polynomial RBFs, thus fulfilling above exponent conditions, and enabling direct solution of intersection problem. After writing it in the form with polynomial coefficients (9), cubic polynomial can be obtained with direct root solutions are always available.

$$C_3 \cdot x^3 + C_2 \cdot x^2 + C_1 \cdot x + C_0 = 0, \quad C_3 \neq 0$$ \hspace{1cm} (15)

The roots of above cubic polynomial (15) are found in 16th century by Nicolo Tartaglia, as published by Gerolamo Cardano, [8].

General cubic with real coefficients (15) always has at least one real solution thus fulfilling the requirement for the existence of the solution. The solution of cubic polynomials equation is well known, and more detailed description of the cubic roots determination can be found in [4].

### 6 Area integral

#### 6.1 General

General area integral for some ship section area calculation can be written as a plane curve integral along some coordinate axis, as given for basic definite integral (1). The section area of a frame using integration along $z$ axis is than:

$$A = \int_{z_1}^{z_2} y \cdot dz$$ \hspace{1cm} (16)

Where $z_1$ and $z_2$ are the limits of definite integral.

In the case of the functional description of the ship's frame section curve with $y = f(z)$ we have:

$$A = \int_{z_0}^{z_{WL}} f(z) \cdot dz$$ \hspace{1cm} (17)

If we choose the point on the ship keel $z_0$ as the lower integral limit, and the point of intersection with waterline $z_{WL}$ as upper limit of the integral limit, we obtain:
Regarding Polynomial RBFs chosen, there are two integration procedures that can be done: by direct integration, or by transformation of radial basis into polynomial basis, and those two ways will be chosen in the text below.

6.2 Direct Polynomial RBF integration

6.2.1 Integration of compatible cubic-linear PRBF

General form of cubic-linear compatible PRBF for z axis with $\beta = \{3, 1\}$ is:

$$y = \sum_{\beta = 3}^{i} w_i \left( \left| z - z_i \right|^3 + c_1 \right) + \sum_{\beta = 1}^{i} w_i \left( \left| z - z_i \right| + c_2 \right)$$

Belonging integral can be written as:

$$A = \int_{z_1}^{z_2} \left[ \sum_{\beta = 3}^{i} w_i \left( \left| z - z_i \right|^3 + c_1 \right) + \sum_{\beta = 1}^{i} w_i \left( \left| z - z_i \right| + c_2 \right) \right] \, dz$$

The direct solution of above integral is:

$$A = \left[ \sum_{\beta = 3}^{i} w_i \left( \frac{\left| z - z_i \right|^4}{4} + c_1 z \right) + \sum_{\beta = 1}^{i} w_i \left( \frac{(z - z_i)^2}{2} + c_2 \cdot z \right) \right]_{z_1}^{z_2}$$

6.2.2 Integration of Polynomial RBFs with general polynomial basis

Since Polynomial RBFs can be written with polynomial basis, the area integral can be generally written as:

$$A = \int_{z_1}^{z_2} \left( \sum_{k=0}^{n} C_k \cdot z^k \right) \, dz$$

Belonging area integral can be than written as:

$$A = \left( \sum_{k=0}^{n} \frac{1}{k+1} \cdot C_k \cdot z^{k+1} \right)_{z_1}^{z_2}$$

Regarding solution is than:

$$A = \sum_{k=0}^{n} \frac{1}{k+1} \cdot C_k \cdot (z_2^{k+1} - z_1^{k+1})$$
7 Calculation of the static section area moment for y axis

7.1 General integral form

General integral for the calculation of ship’s static section area moment for y axis in (2) can be rewritten as:

\[ M_y = \int \limits_{z_i}^{z_f} z \cdot y \cdot dz \] \hspace{1cm} (24)

For \( y = f(z) \) we obtain integral:

\[ M_y = \int \limits_{z_i}^{z_f} z \cdot f(z) \cdot dz \] \hspace{1cm} (25)

The solutions of above integral will be shown below.

7.2 Integral of compatible cubic-linear Polynomial RBF for y axis

The integral of static section area moment for y axis in (2), using cubic-linear PRBFs, can be written as:

\[ M_y = \int \limits_{z_i}^{z_f} \left[ \sum_{\beta = 3} w_i \left( z - z_i \right)^3 + c_i^3 \right] + \sum_{\beta = 1} w_i \left( z - z_i \right) + c_i \cdot z \cdot dz \] \hspace{1cm} (26)

Above integral can be further separated in two integrals as:

\[ M_y = \int \limits_{z_i}^{z_f} \sum_{\beta = 3} w_i \left( z - z_i \right)^3 + c_i \cdot z \cdot dz + \int \limits_{z_i}^{z_f} \sum_{\beta = 1} w_i \left( z - z_i \right) + c_i \cdot z \cdot dz \]

In order to solve the first integral in above, it is necessary to develop it as:

\[ z \cdot \left( z - z_i \right)^3 = z \cdot \left| z \right|^3 - 3 \cdot z \cdot \left| z \right|^2 \left| z_i \right| + 3 \cdot z \cdot \left| z \right|^2 \left| z_i \right|^2 - z \cdot \left| z_i \right|^3 \]

\[ \int \limits_{z_i}^{z_f} z \cdot \left( z - z_i \right)^3 + c_i \cdot z \cdot dz = \int \limits_{z_i}^{z_f} \left[ z^4 - 3 \cdot z^3 \left| z_i \right| + 3 \cdot z^2 \cdot \left| z_i \right|^2 + z \cdot \left( c_i^3 - \left| z_i \right|^3 \right) \right] dz = \]

\[ = \left[ \frac{z^5}{5} - 3 \cdot \frac{z^4}{4} \cdot \left| z_i \right| + 3 \cdot \frac{z^3}{3} \cdot \left| z_i \right|^2 + \frac{z^2}{2} \cdot \left( c_i^3 - \left| z_i \right|^3 \right) \right]_{z_i}^{z_f} \]

The same has to be done with the second integral as:

\[ z \cdot \left( z - z_i \right) = z \cdot \left| z \right| - z \left| z_i \right| \]

\[ \int \limits_{z_i}^{z_f} z \cdot \left( z - z_i + c_i \right) \cdot dz = \int \limits_{z_i}^{z_f} \left[ z^2 + z \cdot \left( c_i - \left| z_i \right| \right) \right] \cdot dz = \left[ \frac{z^3}{3} + \frac{z^2}{2} \left( c_i - \left| z_i \right| \right) \right]_{z_i}^{z_f} \]

Final statement for calculation of static section area moment for y axis is:
$M_y = \sum_{\beta=3}^{\infty} w_\beta \left[ \frac{z^{5}}{5} \left( -3 \frac{z^4}{4} y_i + 3 \frac{z^3}{3} y_i^2 + 2 \left( c_3 - |z_i| \right) \right) \right]_{z_i}^{z_f} + \sum_{\beta=1}^{\infty} w_\beta \left[ \frac{z^3}{3} + \frac{z^2}{2} (c_2 - |z_i|) \right]_{z_i}^{z_f} \tag{27}$

Or we can write differently as

$z \cdot |z - z_i|^3 = \left( z - z_i \right) \cdot |z - z_i|^3 + z_i \cdot |z - z_i|^3$

And

$z \cdot |z - z_i| = \left( z - z_i \right) \cdot |z - z_i| + z_i \cdot |z - z_i|$

Final statement for direct calculation of static section area moment for $y$ axis is than:

$M_y = \sum_{\beta=3}^{\infty} w_\beta \left[ \frac{(z - z_i)^5}{5} + \frac{(z - z_i)^4}{4} y_i + c_3 \frac{z^2}{2} \right]_{z_i}^{z_f} + \sum_{\beta=1}^{\infty} w_\beta \left[ \frac{(z - z_i)^3}{3} + \frac{(z - z_i)^2}{2} y_i + c_2 \frac{z^2}{2} \right]_{z_i}^{z_f} \tag{28}$

7.3 Integral of compatible cubic-linear PRBF for $y$ axis written for polynomial basis

The equation for static section area moment calculation using PRBFs with polynomial basis can be generally written as:

$M_y = \int_{z_i}^{z_f} \left( \sum_{k=0}^{n} C_k \cdot z^k \right) \cdot dz \tag{29}$

Belonging solution of moment integral can be than written as:

$M_y = \left( \sum_{k=0}^{n} \frac{1}{k+2} \cdot C_k \cdot z^{k+2} \right)_{z_i}^{z_f} \tag{30}$

After including limit values we obtain:

$M_y = \sum_{k=0}^{n} \frac{1}{k+2} \cdot C_k \cdot \left( z_f^{k+2} - z_i^{k+2} \right) \tag{30}$

8 Calculation of the static section area moment for $z$ axis

8.1 General integral

The integral for the calculation of static section area moment for $z$ axis in (3) can be rewritten as:

$M_z = \frac{1}{2} \int_{z_i}^{z_f} y^2 \, dz \tag{31}$

If we insert explicit curve definition $y = f(z)$ we obtain integral:
Analytical solution of basic ship hydrostatics integrals using polynomial radial basis functions

Dario BAN
Josip BAŠIĆ

\[ M_z = \frac{1}{2} \int_{z_i}^{z_2} f^2(z) \cdot dz \] \tag{32}

This integral has form of basic integral number 3, as defined in chapter 2.

Similarly to the calculation of static area moment for \( y \) axis, direct integration of above equation with radial basis functions is very complex even for Polynomial RBFs, and therefore will not be done. In order to solve integral (32) above, Polynomial RBFs will be written in the form of algebraic polynomials, instead. Additionally, it will be shown that another efficient solution of above integral can be obtained by using the solution of interpolation problem for \( y^2 \).

### 8.2 Integral of static area moment for \( z \) axis using compatible cubic-linear PRBF

The integral of static section area moment for \( z \) axis using cubic-linear PRBF can be written as:

\[ M_z = \frac{1}{2} \int_{z_i}^{z_2} \left[ \sum_{\beta=3}^{n} w_i \left( |z - z_i|^3 + c_1 \right) + \sum_{\beta=1}^{n} w_i \left( |z - z_i| + c_2 \right) \right]^2 dz \] \tag{33}

This integral can be divided in three integrals as:

\[
M_z = \frac{1}{2} \int_{z_i}^{z_2} \left[ \sum_{\beta=3}^{n} w_i \left( |z - z_i|^3 + c_1 \right) \right] \cdot dz + \int_{z_i}^{z_2} \left[ \sum_{\beta=3}^{n} w_i \left( |z - z_i|^3 + c_1 \right) \right] \cdot \left[ \sum_{\beta=1}^{n} w_i \left( |z - z_i| + c_2 \right) \right] \cdot dz + \\
\frac{1}{2} \int_{z_i}^{z_2} \left[ \sum_{\beta=1}^{n} w_i \left( |z - z_i| + c_2 \right) \right]^2 \cdot dz
\]

This equation with RBF description contains translated weighted sums with basis functions and therefore has multiple integral terms non suitable for direct calculation. More suitable polynomial form will be used instead, as will be shown below.

### 8.3 Static area moment integral for \( z \) axis using PRBFs in polynomial form

When radial basis function can be developed in polynomial form, static area moment integral for \( z \) axis can be generally written as:

\[ M_z = \frac{1}{2} \int_{z_i}^{z_2} \left( \sum_{k=0}^{n} C_k \cdot z^k \right)^2 \cdot dz \] \tag{34}

By squaring we get:

\[ M_z = \frac{1}{2} \int_{z_i}^{z_2} \sum_{k=0}^{2n} D_k \cdot z^k \cdot dz \] \tag{35}

where \( D_k = f(C_k) \) are polynomial coefficients.

In the case of cubic-linear Polynomial RBFs, belonging solution of the moment integral can be written as:
Analytical solution of basic ship hydrostatics integrals

using polynomial radial basis functions

8.4 Direct integration using $y^2$

Instead of direct solving of basic integral for $M_z$ by developing complex RBF form with norms, that integral can be solved by determining weight coefficients $w_2$ for curve equation $y = f(z)$ squared, i.e. for $y^2$. Therefore, we are introducing the variable $q = y^2$ and obtaining the integral with the same form as the area integral (1), i.e.:

$$M_z = \frac{1}{2} \int_{z_1}^{z_2} \left( C_3 \cdot z^3 + C_2 \cdot z^2 + C_1 \cdot z + C_0 \right) \cdot dz$$

By squaring we get:

$$M_z = \frac{1}{2} \int_{z_1}^{z_2} \left( D_6 \cdot z^6 + D_5 \cdot z^5 + D_4 \cdot z^4 + D_3 \cdot z^3 + D_2 \cdot z^2 + D_1 \cdot z + D_0 \right) \cdot dz$$

The solution can be then written as

$$M_z = \frac{1}{2} \left( \frac{D_6}{7} \cdot z^7 + \frac{D_5}{6} \cdot z^6 + \frac{D_4}{5} \cdot z^5 + \frac{D_3}{4} \cdot z^4 + \frac{D_2}{3} \cdot z^3 + D_1 \cdot z + D_0 \right) \bigg|_{z_1}^{z_2}$$

Or

$$M_z = \frac{1}{2} \sum_{k=0}^{2n} \frac{1}{k+1} D_k z^{k+1} \bigg|_{z_1}^{z_2} = \frac{1}{2} \sum_{k=0}^{2n} \frac{1}{k+1} D_k (z^{k+1} - z_1^{k+1})$$

8.4.1 Interpolation problem for $y^2$

Instead of basic interpolation problem $y = H \cdot w$, we are defining the problem:

$$y^2 = f_2(z) = H_2 \cdot w_2$$

Where $H_2$ is interpolating matrix for input data set, and $w_2$ is weight coefficients vector.

Similar to basic scattered data interpolation problem, the solution of this problem can be obtained by the inversion of interpolating matrix $H_2$, and according to (7) we have:

$$w_2 = H_2^{-1} \cdot y^2$$
Since interpolating matrix does not depend on output data, but input data set, the interpolating matrix for \( y \) and \( y^2 \) are equal:

\[ H_2 = H \]

Since interpolating matrices \( H \) and \( H_2 \) equal, it is not necessary to do another matrix inversion that can be advantageous, i.e. we have:

\[ w_2 = H^{-1} \cdot y^2 \]  \hspace{1cm} (41)

Therefore, after solving the inversion of basic interpolating matrix, it is possible to determine weight coefficients of any output data set, and therefore it is possible to scale output data set. Following definition can be written:

**Definition 1:** The RBF description has basic property of affine transformation by scaling output data set \( Y \).

### 8.5 Solution of interpolating problem for \( y^2 \) using cubic-linear PRBF

The solution of the static area moment integral for \( z \) axis is identical to the solution of the area integral (20) for cubic-linear Polynomial RBFs but with weight coefficients \( w_2 \) used:

\[ M_z = \left[ \sum_{j=3} w_{2j} \left( \frac{|z - z_j|^4}{4} + c_1^3 z \right) + \sum_{j=1} w_{2j} \left( \frac{|z - z_j|^2}{2} + c_2^3 z \right) \right]_{z_1}^{z_2} \]  \hspace{1cm} (42)

Therefore, if direct solution of the basic RBF integral is known, it is easy to determine static moment for \( z \) axis then. This is the easiest way for solving above integral, and it will be applied in further calculations.

### 9 Calculation of waterplane inertia moment for \( x \) axis

#### 9.1 Basic integral

The basic integral for calculation of waterplane inertia moment around \( x \) axis, in (4), can be rewritten as:

\[ I_x = \frac{1}{3} \int_{x_i}^{x_f} y^3 \, dx \]  \hspace{1cm} (43)

By inserting curve functional definition in explicit form \( y = f(x) \) in it, following integral is obtained:

\[ I_x = \frac{1}{3} \int_{x_i}^{x_f} f^3(x) \cdot dx \]  \hspace{1cm} (44)

This integral represents fourth basic integral for ship's hydrostatic particulars calculation to be solved and its solution follows below.
9.2 Integral of inertia moment for x axis using cubic-linear Polynomial RBFs

Belonging inertia moment integral for cubic-linear Polynomial RBFs can be written as:

\[ I_x = \frac{1}{3} \int_{x_1}^{x_2} \left( \sum_{i=1}^{n} w_i \left( x - x_i \right)^3 + c_1 \right) + \sum_{i=1}^{n} w_i \left( x - x_i \right)^2 + c_2 \] \ dx \tag{45} 

Above integral can be separated into four integrals as:

\[ I_x = \frac{1}{3} \int_{x_1}^{x_2} \left( \sum_{i=1}^{n} w_i \left( x - x_i \right)^3 + c_1 \right)^3 \ dx + \frac{1}{3} \int_{x_1}^{x_2} \left( \sum_{i=1}^{n} w_i \left( x - x_i \right)^3 + c_1 \right)^2 \ \left( \sum_{i=1}^{n} w_i \left( x - x_i \right) + c_2 \right) \ dx + \frac{1}{3} \int_{x_1}^{x_2} \left( \sum_{i=1}^{n} w_i \left( x - x_i \right) + c_2 \right)^3 \ dx + \frac{1}{3} \int_{x_1}^{x_2} \left( \sum_{i=1}^{n} w_i \left( x - x_i \right) + c_2 \right)^2 \ \left( \sum_{i=1}^{n} w_i \left( x - x_i \right)^3 + c_1 \right) \ dx \]

This integral contains multiple terms because of basic RBF's definition with translated sums, and therefore it is not suitable for calculation in this form. Therefore, it will be solved for Polynomial RBFs written in basic polynomial form, (9).

9.3 Moment of Inertia Integral for PRBFs written in general polynomial form

The integral of inertia moment around x axis written for PRBF in general polynomial form can be written as:

\[ I_x = \frac{1}{3} \int_{x_1}^{x_2} \left( \sum_{j=0}^{m} C_j \cdot x^j \right)^3 \ dx \tag{46} \]

By cubing it we obtain:

\[ I_x = \frac{1}{3} \int_{x_1}^{x_2} \sum_{j=0}^{3m} D_j \cdot x^j \ dx \]

Where: \( D_j = f(C_j) \)

Belonging integral solution, in the case of cubic-linear PRBF, can be written as:

\[ I_x = \frac{1}{3} \int_{x_1}^{x_2} \left( C_3 \cdot x^3 + C_2 \cdot x^2 + C_1 \cdot x + C_0 \right)^3 \ dx \tag{47} \]

By cubing it, complex solution is obtained, that can be written as:

\[ I_x = \left. \frac{1}{3} \sum_{j=0}^{3n} \frac{1}{j+1} D_j \ x^j \right|_{x_1}^{x_2} = \frac{1}{3} \sum_{j=0}^{3n} \frac{1}{j+1} D_j \left( x_{j+1}^{x_2} - x_{j+1}^{x_1} \right) \tag{48} \]
9.4 Direct integration using $y^3$

Instead of solving basic moment of inertia integral $I_x$ by developing complex RBF form with norms, corresponding integral can be solved using weight coefficients $w_3$ for cubed curve equation, i.e. for $y^3$, as shown before for $y^2$ in (41). Therefore, by substituting $q = y^3$ the same integral form as basic area integral is obtained, i.e. we obtain:

$$ I_x = \frac{1}{3} \int_3^1 q \, dx $$

(49)

And by inserting $f_3(x)$, following integral is obtained:

$$ I_x = \frac{1}{3} \int_3^1 f_3(x) \cdot dx $$

The function $f_3$ has different weight coefficients $w_3$ than function $f$ with coefficients $w$, and it is necessary to define separate interpolation problem for $y^3$, as will be shown below.

9.5 Interpolation problem for $y^3$

Instead of interpolation problem for $y = H \cdot w$ we define problem:

$$ y^3 = f_3(x) = H \cdot w_3 $$

(50)

Where $H$ is interpolating matrix for input data set, and $w_3$ is weight coefficients vector.

But interpolating matrix does not depend on output, but only input data set, as shown before, so we have:

$$ H_3 = H $$

(51)

Similar to basic scattered data interpolation problem, the solution of this problem is obtained by inverting interpolating matrix $H$ which is according to (7) known as:

$$ w_3 = H^{-1} \cdot y^3 $$

(52)

9.6 The solution of interpolation problem for $y^3$ using cubic-linear PRBF

The solution of the inertia moment integral for $x$ axis (4) using cubic-linear Polynomial RBF is identical to the solution of the area integral (20) with weight coefficients $w_3$ used:

$$ I_x = \sum_{\beta=3} w_{3i} \left( \frac{|x - x_i|^4}{4} + c_1 x_i \right) + \sum_{\beta=4} w_{3i} \left( \frac{|x - x_i|^2}{2} + c_2 \cdot x \right) $$

(53)

Therefore, if direct solution of the integral for general radial function is known, corresponding solution of the waterplane moment of inertia for $x$ axis is known, also.
10 Calculation of waterplane inertia moment for y axis

10.1 Basic integral

Basic integral form for the calculation of waterplane moment of inertia for y axis shown in (5) can be rewritten as:

\[ I_y = \int_{x_1}^{x_2} x^2 \cdot y \cdot dx \] (54)

By inserting waterline curve description in explicit form \( y = f(x) \) following integral can be obtained:

\[ I_y = \int_{x_1}^{x_2} x^2 \cdot f(x) \cdot dx \] (55)

This integral represents basic fifth integral of ship's hydrostatics.

Direct integration of above integral can be very complex when general Polynomial RBFs are used. Therefore, the integration of cubic-linear Polynomial RBFs will be shown instead, i.e. the integration of RBFs written in polynomial form.

10.2 Moment of inertia integral for y axis using compatible cubic-linear PRBF

Belonging integral of waterplane moment of inertia for y axis using cubic-linear Polynomial RBF can be written as:

\[ I_y = \int_{x_1}^{x_2} x^2 \cdot \left[ \sum_{i=1}^{3} w_i (|x - x_i|^3 + c_i^3) + \sum_{j=1}^{3} w_j (|x - x_j| + c_j^2) \right] dx \] (56)

I.e., above integral can be separated in two integrals as:

\[ I_y = \int_{x_1}^{x_2} x^2 \cdot \left[ \sum_{i=1}^{3} w_i (|x - x_i|^3 + c_i^3) \right] dx + \int_{x_1}^{x_2} x^2 \cdot \left[ \sum_{j=1}^{3} w_j (|x - x_j| + c_j^2) \right] dx \]

In order to solve first integral in above equation (56), it is necessary to develop the argument of \( L_1 \) norm, i.e. develop \( |x - x_i|^3 \) and multiply it with \( x^2 \) as:

\[ x^2 \cdot |x - x_i|^3 = x^2 \cdot |x|^3 - 3 \cdot x^2 \cdot |x|^2 \cdot |x_i| + 3 \cdot x^2 \cdot |x||x_i|^2 - x^2 \cdot |x_i|^3 \]

After inserting we have

\[ \int_{x_1}^{x_2} x^2 \cdot |x - x_i|^3 + c_i^3 \right) dx = \left[ \frac{x^6}{6} - 3 \cdot \frac{x^5}{5} \cdot |x_i| + 3 \cdot \frac{x^4}{4} \cdot |x_i|^2 + \frac{x^3}{3} \cdot (c_i^3 - |x_i|^3) \right]_{x_1}^{x_2} \]

Similar has to be done with second integral with:
Analytical solution of basic ship hydrostatics using polynomial radial basis functions

Dario BAN
Josip BAŠIĆ

\[ x^2 \cdot |x - x_i| = x^2 \cdot |x - x_i| \]

We get

\[
\int_{x_i}^{x_2} x^2 \cdot (x - x_i + c_2) \, dx = \int_{x_i}^{x_2} \left[ x^3 + x^2 \cdot (c_2 - |x_i|) \right] \, dx = \left[ \frac{x^4}{4} + \frac{x^3}{3} (c_2 - |x_i|) \right]_{x_i}^{x_2}
\]

Final solution for ship's waterplane moment of inertia for y axis is then:

\[
I_y = \sum_{\beta=3}^{n} w_i \left[ \frac{x^6}{6} - 3\frac{x^5}{5} |x_i| + 3\frac{x^4}{4} |x_i|^2 + \frac{x^3}{3} (c_i^3 - |x_i|^3) \right]_{x_i}^{x_2} + \sum_{\beta=1}^{n} w_i \left[ \frac{x^4}{4} + \frac{x^3}{3} (c_2 - |x_i|) \right]_{x_i}^{x_2}
\] (57)

10.3 Moment of inertia Integral for y axis using Polynomial RBF in general polynomial form

For general RBF written in polynomial form, the integral for moment of inertia for y axis can be generally written as:

\[
I_y = \int_{x_i}^{x_2} x^2 \cdot \left( \sum_{j=0}^{n} C_j \cdot x^j \right) \, dx
\] (58)

I.e., we have:

\[
I_y = \int_{x_i}^{x_2} \left( \sum_{j=0}^{n} C_j \cdot x^{j+2} \right) \, dx
\] (59)

In the case of cubic-linear Polynomial RBF, belonging moment of inertia integral solution can be written as:

\[
I_y = \int_{x_i}^{x_2} \left( C_3 \cdot x^5 + C_2 \cdot x^4 + C_1 \cdot x^3 + C_0 x^2 \right) \, dx
\] (60)

The solution of this integral is simply obtained by integrating every polynomial term as:

\[
I_y = \left( \frac{1}{6} C_3 \cdot x^6 + \frac{1}{5} C_2 \cdot x^5 + \frac{1}{4} C_1 \cdot x^4 + \frac{1}{3} C_0 x^3 \right)_{x_i}^{x_2}
\] (61)

Or:

\[
I_y = \left( \sum_{j=0}^{n} \frac{1}{j+3} C_j \cdot x^{j+3} \right)_{x_i}^{x_2}
\]

Finally, when limits are inserted, the solution can be written as:

\[
I_y = \sum_{j=0}^{n} \frac{1}{j+3} C_j \cdot \left( x_2^{j+3} - x_i^{j+3} \right)
\] (62)

This solution of fifth basic integral (5) used for calculation of waterplane inertia moment for y axis is the easiest possible, and can be used in practice.
11 Curve length calculation using Polynomial RBF

Except five basic integrals of ship' hydrostatics, there is a need for determination of wetted area of immersed ship hull on actual waterline, in ship resistance calculation. The integral of curve length for some frame section therefore can be included into basic integrals to be determined, too. Similarly to first five integrals, this integral should be always solvable.

In order to define the length integral for curve description defined with \( f(x) \), it is necessary to differentiate it once, obtaining \( g(x) = f'(x) \). If Polynomial RBF is written in general polynomial form, with polynomial coefficients introduced for \( f(x) \), the function \( g(x) \) can be obtained after differentiation with:

\[
g(x) = \left( \sum_{k=1}^{n} C_k \cdot z^{k-1} \right)^2
\]

where \( C_k \) are polynomial coefficients.

After inserting \( g(x) \) in integral (6) following integral is obtained:

\[
L = \int_{x_1}^{x_2} \sqrt{1 + \left( \sum_{k=1}^{n} C_k \cdot z^{k-1} \right)^2} \, dx
\]

Written by coefficients it becomes:

\[
L = \int_{x_1}^{x_2} \sqrt{1 + c_0 + c_1 x + \ldots + c_{2\beta-2} x^{2\beta-2}} \, dx
\]

where \( c_k \) are polynomial coefficients after derivation and squaring of polynomial.

However, as described before, only Polynomial RBFs with integer exponent \( \beta \) equal three is acceptable as smooth description solution, and therefore that case will be investigated here. After squaring Polynomial RBF with \( \beta = 3 \), belonging polynomial function \( g(x) \) with degree four is obtained.

The indefinite integral for Polynomial RBF with exponent \( \beta = 3 \) is:

\[
L = \int x \sqrt{1 + c_0 + c_1 x + c_2 x^2 + c_3 x^3 + c_4 x^4} \, dx
\]

The solution contains elliptic integrals of the first kind \( F(x|m) \), elliptic integrals of the second kind \( E(x|m) \) and elliptic integrals of the third kind \( \Pi(x|m) \).

For special case where \( g(x) = (x - a)^2 \) the solution is:

\[
L = \frac{1}{3} \left\{ \sqrt{(x-a)^4 + 1 \cdot (x-a)} + 2 \cdot \frac{1}{4} \cdot F \left[ \sinh^{-1} \left( \sqrt{-1 \cdot (x-a)} \right) \right] \right\}
\]

Where \( F(x|m) \) is elliptic integral of the first kind.
Above length integrals are not always solvable and their existing general solutions are complex. Therefore, it is more efficient to calculate curve length by segments using curve description with Polynomial RBFs, for total input data set $X_G$ used for generalization, with:

$$L = \sum_{j=1}^{N} \sqrt{\left(x_j - x_{j-1}\right)^2 + \left(y_j - y_{j-1}\right)^2}$$ \hspace{1cm} (67)

where curve segments represent $l_2$ norm of input and output point pairs.

It can be concluded, that there is no efficient polynomial method available for direct solution of basic curve length integral using Polynomial RBFs with exponent $\beta \geq 3$. Therefore, the combination of analytical and numerical calculation method should be used for curve length determination, instead of solely analytical method.

12 The example of the frame section area particulars calculation

The solutions of above integrals are tested for actual ship's Test frame No. 1 shown on Figure 3, aft frame of car-truck carrier with flat of the side, camber and discontinuities, [9]; and theoretical semicircle test frame section, Test frame No. 2, Figure 5.

![Figure 5: Theoretical Test frame No. 2 in the form of semicircle](image)

The integration results will be tested for test frame sections Bonjean curves as:

$$\{A, y_B, z_B\} = f(z)$$

The comparison of calculated results using composition of cubic-linear Polynomial RBFs and corresponding test frames are shown in Table 1, below.

<table>
<thead>
<tr>
<th></th>
<th>Test frame No. 1 until camber</th>
<th>Test frame No. 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>PRBF</td>
<td>Trim &amp; Stability Booklet</td>
<td>PRBF</td>
</tr>
<tr>
<td>$A \ (m^2)$</td>
<td>311.3901</td>
<td>9.8178</td>
</tr>
<tr>
<td>$z_B \ (m)$</td>
<td>311.3901</td>
<td>9.8175</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Analytical values</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1.4389</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1.4389</td>
</tr>
</tbody>
</table>
Theoretically, vertical centroid of the semicircle area $z_B$ can be obtained by:

$$z_B = r - \frac{4r}{3\pi}$$

Vertical position of the centre of semicircle area for $r = 2.5$ (m) is than $z_B = 1.4389$ (m). It can be seen from Table 1 that the same vertical centre of semicircle area is obtained by the calculations with Polynomial RBFs, while area value differs in forth decimal, thus satisfying required calculating precision of 1 (mm) in shipbuilding.

Also, it is visible from Polynomial RBF integration results of area particulars for Test frame No. 1 shown on Figure 6, that their distribution is smooth for whole draft, as required by ship computational geometry. Table 1, above, shows that result for actual ship test frame are very accurate and correspond to the values from Trim and Stability Booklet for Test frame No. 1, [9].

13 Conclusion

Global curve description using composition of cubic-linear polynomial radial basis functions enables direct solution of five basic integrals of ship's hydrostatics for whole drafts range. Besides integrability, high precision of the geometric and hydrostatic particulars is shown, with the elimination of the effect of grouping of error in a point, connected with global geometry description. Belonging integral for curve length calculation is not always solvable, but functional geometry description using cubic-linear Polynomial radial basis functions enables its calculation using summation of segments used for its generalization.

Except above mentioned, the direct solution of RBF integrals eliminates the need for numerical integration procedures that dominated ship's computational geometry calculations.
for a long time, and dictated the distribution of ship frame sections and waterlines in wireframe modelling of ship geometry. Direct RBF integration enables arbitrary distribution of ship frame sections using meshless RBF principle, as well as high precision of calculations, even for the existence of the discontinuities in ship geometry.

It can be concluded that solution of arbitrary ship geometry using composition of cubic-linear Polynomial RBFs represents required analytical solution of all ship's computational geometry problems in the case of global, non-two manifold 2D description, with remark of using the combination of analytical and numerical solution for curve length calculation.

Corresponding analytical solution for 3D problem of ship geometry description is not found yet, and that will be the subject of further research of authors of this paper.

REFERENCES

NEW ALGORITHM FOR OPTIMAL DIELECTRIC MATERIAL SELECTION IN MARINE ENVIRONMENT

UDC 629.5.03:629.5.04:537.226
Preliminary communication

Summary

The materials’ selection demands knowledge from different disciplines, depending on application. Very important parameters that influence dielectric material’s properties in, for example, marine applications are operating frequency, expected temperature in practice and moisture. The proposed optimal solution for the dielectrics choice is based on theory of sets. Instead of using function to find best value for the dielectric constant, the parameters’ dependences are used to produce sets of possible values, which are used to find the optimal material for the desired application. The intersection of the three sets of possible solution is the optimal solution if the material with such value exists. If not, maximum acceptable deviation is used to find the acceptable material.

Key words: relative dielectric constant; frequency dependence; temperature dependence; moisture dependence; material’s resonant frequency

1. Introduction

Dielectric materials are widely used for many purposes. Classic usage is in covers, masks, materials which enable ergonomy, efficiency and compactness of final products, but main usage is to ensure correct and safe ship power system operation trough insulator properties (shielding of cables, capacitors, etc.). Cutting-edge of their usage is in the optic communications for fibre optics, and bleeding-edge is in the LCD technology (liquid crystals exhibit dielectric properties), but it is not primary goal of the paper. One of the important characteristics of the dielectric materials is the relative dielectric constant (permittivity).

Dielectric materials in the marine environment, as an example, are under negative influence of many parameters. These parameters are present in the shore applications as well. Due to adverse weather and atmospheric conditions, change of properties of dielectric materials plays an even more important role in marine applications.

In this paper, an algorithm for optimal dielectric material selection is presented. It is based on above mentioned parameters. In the second section, a literature background is presented. The third section covers mathematical background. The fourth section describes the proposed algorithm and illustration of an example of the material. Simulation results are presented in the 3D graph. Conclusions are presented in the final section.
2. Literature background

Researches on influence of various parameters to the materials’ characteristics are scope of many papers, but none specifically deals with marine applications. Some of them are briefly mentioned. Literature will be grouped by parameters of their studies. Moisture was studied in [1, 3, 4, 6, 8, 11, 17]. Various types of frequencies dependences (operating, which is the most relevant for operating characteristics, natural, resonant) are the scope in e.g. [5, 9, 13]. A combination of frequency and temperature dependences is researched in [7, 10, 16]. Temperature dependence for the dielectric constant is the topic of [2, 12, 15].

An attempt to model moisture influence by math is presented in [3]. The paper deals with the implementation of dielectric constant change to measure moisture (in moisture-meters).

Influence of moisture on electrical properties and reliability of materials is researched in [1]. It is pointed out that porous dielectrics with low dielectric constant absorb more moisture than dense ones. Furthermore, chemically-adsorbed moisture degrades the electrical and reliability performance of the dielectrics with low dielectric constant. It is also reported that higher temperature anneals at 400 °C can decompose physically-adsorbed water and restore reliability performance. Mechanism for degradation of performance is the electromigration of the copper interconnections.

The aim of [8] was to determine the mathematical model for soil moisture (but not for materials for the off-shore applications) and dielectric constant. The experimental results are compared to different mathematical models and one of polynomial model was shown as the best match. Similar is covered by [4], but as the response to different frequencies. Although investigation of soil types is not the equivalent to investigation of dielectric material in engineering, the value of research is in experimental data, which are connected to exact mathematical expressions, and shows the framework for research in engineering materials. Dielectric constant is used as an input to moisture calculation in [11], which is the opposite of the research in this paper, when the aim is to find out the influence of moisture to dielectric constant. Research in electromagnetic for the moisture-dielectric constant dependences is presented in [6].

Thin films, which are popular in nowadays technology, were investigated for GHz range in [5]. The problem observed in references is that articles deal with experimental properties of the specific material, and the conclusion is not usually generalized except in some specific group. For example, operating frequency dependence based on a type of modified Debye equation named by Havriliak and Negami is considered for elastomers in [13]. The experimental data is fitted to the theoretical curve.

Since it was observed that both operating frequency and temperature dependence play an important role in the final result, some authors manage to cope with both parameters. For example, in [7], composite material Epoxy/BaTiO$_3$ is researched and Poly Vinylidene Fluoride (PVDF) in [16]. An interesting experimental data are presented in [10], where printed circuit board (PCB) substrate is considered. Investigated operating frequency range is between 1 kHz and 1 MHz. It was shown that dielectric constant falls when temperature is increased. However, frequency dependence in the considered range is not so clear.

Temperature dependence of dielectric permittivity is considered on very popular materials in [15]. Ferroelectrics” dielectric constant is considered in [2] and ice in [12].

3. Mathematical background

Operating frequency dependence is expressed depending on type of polarization. The polarization type existing in broad frequency range is the electronic polarization. It can be
used as a framework to define the resonant frequency of the material. The resonant frequency is an input to Debye equations. The model to calculate resonant frequency can be explained in a simple manner. In the absence of an electric field, the centre of mass of the nucleus and the electrons’ orbital motions coincide. There are no dipoles and dipole moment is zero. Electrons become displaced if the electric field producing dipole moment is applied. The restoring force, $F_r$, is proportional to the displacement $x$ with a constant $\beta$[9]:

$$F_r = -\beta \cdot x$$ (1)

In the equilibrium:

$$Z \cdot e \cdot E = \beta \cdot x$$ (2)

Magnitude of the induced electronic dipole moment is:

$$p_e = (Z \cdot e) \cdot x = \left(\frac{Z^2 \cdot e^2}{\beta}\right) \cdot E$$ (3)

The equation of motion is:

$$-\beta \cdot x = Z \cdot m_e \cdot \frac{d^2 x}{dt^2}$$ (4)

Displacement can be expressed with:

$$x(t) = x_0 \cdot \cos (\omega_r \cdot t)$$ (5)

where the oscillation of the centre of mass of the electron cloud is actually electronic polarization resonance frequency:

$$\omega_r = \sqrt{\frac{\beta}{Z \cdot m_e}}$$ (6)

Depending on material, model of one or two relaxations can be used to describe the dependence of complex dielectric constant on frequency. Debye and Cole-Cole models are widely used to mathematically describe the dependence. For one relaxation, the dependence is described with [4]:

$$\varepsilon_r(\omega) = \varepsilon_\infty + \frac{\varepsilon_{DC} - \varepsilon_\infty}{1 + \left(\frac{i\omega}{\omega_r}\right)^{1-\alpha}}$$ (7)

where $\varepsilon(\omega)$ is complex relative dielectric constant, $\varepsilon_{DC}$ the value of the constant when DC field is applied, $\varepsilon_\infty$ is the value of the constant at high operating frequencies (\omega) in range where orientational polarization is negligible and $\alpha$ is exponent that describes the spread of the relaxation peak. For two relaxations, the dependence is expressed with:

$$\varepsilon_r(\omega) = \varepsilon_\infty + \frac{\varepsilon_{DC2} - \varepsilon_{DC1}}{1 + \left(\frac{i\omega}{\omega_{r1}}\right)^{1-\alpha_1}} + \frac{\varepsilon_{DC1} - \varepsilon_\infty}{1 + \left(\frac{i\omega}{\omega_{r2}}\right)^{1-\alpha_2}}$$ (8)

where $\varepsilon_{DC1}$ and $\varepsilon_{DC2}$ are values of $\varepsilon_{DC}$ at the first and the second relaxation peak respectively and $\alpha_1$ and $\alpha_2$ coefficients that describe spread of the first and the second relaxation peak.
Real and imaginary part of complex relative dielectric constant is expressed according to Debye [3, 9]:

\[ \varepsilon'_r = \varepsilon_\infty + \frac{\varepsilon_{DC} - \varepsilon_\infty}{1 + \left(\frac{\omega}{\omega_r}\right)^2} \]

\[ \varepsilon''_r = \frac{(\varepsilon_{DC} - \varepsilon_\infty) \cdot \frac{\omega}{\omega_r}}{1 + \left(\frac{\omega}{\omega_r}\right)^2} \]

Figure 1 shows an example of the resonant frequency’s contribution to relative dielectric constant. It shows the behaviour of the same group of materials with a different resonant frequency. The rule of the operating frequency dependence is the same, but the values are different. Every curve represents another material. Therefore, one can develop an algorithm for materials’ selection based on resonant frequency and required relative dielectric constant.

**Dependance of \( \varepsilon_r \) in function of resonant frequency**

**Fig. 1** Resonant frequency vs. relative dielectric constant graph

The maximum value (15.66) of the relative dielectric constant is the value at DC or low-frequency electric field. The minimum value (3.33) is at the frequency where orientational polarization is negligible. Starting material for this simulation is 1-pentanol. Instead of \( \omega_r \), relaxation time, \( \tau \), can be used.
Since, modern polymeric materials are more widely used every day, original Debye equations have been modified to cope with broad relaxation peaks that have been observed [9]. The modified expression for complex dielectric constant is:

$$\varepsilon_r = \varepsilon_\infty + \frac{\varepsilon_{DC} - \varepsilon_\infty}{1 + (j\omega\tau)\alpha\beta}$$  \hspace{1cm} (11)

where $\alpha$ and $\beta$ are axis constants (less than 1). This model enables inclusion of the temperature dependences through relaxation time, which is temperature-dependent.

In [14, 15], frame of classical dielectric model is used to describe dielectric function:

$$\varepsilon_r(\omega) = \varepsilon_\infty + \sum_j \frac{S_j \Omega_j^2}{\Omega_j^2 - \omega^2 - i\gamma_j \omega}$$  \hspace{1cm} (12)

where $\varepsilon_\infty$, $\Omega_j$, $S_j$ and $\gamma_j$ are respectively the high-frequency value of the dielectric constant, and the transverse optical wave number, the dielectric strength and the damping of the $j$th phonon. This description does not include lattice anharmonicities and phonon interactions. In order to include temperature, (6) is modified to:

$$\varepsilon_r(\omega,T) = \varepsilon_\infty + \sum_j \frac{S_j(T)\Omega_j^2(T)}{\Omega_j^2(T) - \omega^2 - i\gamma_j(T)\omega}$$  \hspace{1cm} (13)

Temperature dependent parameters in (13) are:

$$\Omega_j(T) = \Omega_j(T_0) + a_j[T - T_0]$$  \hspace{1cm} (14)

$$S_j(T) = S_j(T_0) + b_j[T - T_0]$$  \hspace{1cm} (15)

$$\frac{\gamma_j(T)}{\Omega_j(T)} = \frac{\gamma_j(T_0)}{\Omega_j(T_0)} + c_j[T - T_0]$$  \hspace{1cm} (16)

$$\varepsilon_\infty(T) = \varepsilon_\infty(T_0) + e[T - T_0]$$  \hspace{1cm} (17)

where $e$, $a_j$, $b_j$, and $c_j$ are constant coefficients, and $T_0$ a reference temperature.

Moisture is modelled by a few functions, depending on application. In [8], the relationship between dielectric constant and volumetric moisture is obtained in dependence of soil class as polynomial of the first or the third order:

$$\theta = a \pm b \cdot \varepsilon_r \left( + c \cdot \varepsilon_r^2 \pm d \cdot \varepsilon_r^3 \right)$$  \hspace{1cm} (18)

where $a$, $b$, $c$ and $d$ depend on soil type.

In [6], the relationship between the moisture content and leaf parameters is given for X-band applications. For X-band, real and imaginary parts of the dielectric constant are given as:

$$\text{real}(\varepsilon_r) = a \cdot \exp(b \cdot Mc) - c$$  \hspace{1cm} (19)

$$\text{imag}(\varepsilon_r) = d \cdot \exp(e \cdot Mc) - f$$  \hspace{1cm} (20)

where $a = 3.95$, $b = 2.79$, $c = 2.25$, $d = 2.69$, $e = 2.15$ and $f = 2.68$.

As it can be seen, the dependences mentioned can be combined by substituting (14) – (17) into (13) and then the result into (18). However, this approach has many limitations, such as the range of materials to which it is applicable, limited frequency range, etc. Therefore, no one suggested the usage of such unified equation. Instead of it, the approach in this paper is to
try to use the known experimental/theoretical data to find optimal material for the desired application, in our case usage in ocean atmosphere. In order to describe data, a lookup table is used. The material is represented by the relative dielectric constant.

4. Proposed algorithm with an example

Relative dielectric constant is a characteristic of the dielectric’s properties of the material. It depends on temperature and moisture, which degrade the material’s properties. It is obvious that the optimal solution must be found by defining the function, which represents temperature, frequency and moisture dependences, but it is not obvious and often impossible to explicitly define such dependences. Since interdependences are too complex to be covered by one multidimensional function, a different strategy is tried in this paper – the theory of sets.

Instead of calculating the optimal number for relative dielectric constant, we will find a range of possible values for it based on temperature dependence, frequency dependence and moisture dependence. Then, the optimal result can be found in the intersection of these three sets of possible values. The algorithm proposed can be modified for computer and for manual graphical solution. Figure 2 illustrates the basic idea of the proposed. The simulated material is used for PCB, which consists of resin, glass fibre and copper foil. In electronics it is very important to know dielectric properties of PCB [10].

![Fig. 2 Illustration of the operational principle for the developed algorithm](image)
In order to retrieve temperature in Kelvins from Fig. 2, one should use:

\[ T [\text{K}] = 300 + (n_0 - 1) \cdot 100 \]  \hspace{1cm} (21)

where \( n_0 \) is the number at the axis in the Fig. 2.

The numbers in the frequency axis represent the number of elements in frequency vector programmed in Matlab/Octave simulating environment. Frequency, temperature and moisture vectors are taken in order to execute a fast algorithm. In theory, all values of any parameter could be taken into account, but range and step of such operation are limited with memory and processor capacities. In our simulation, for Fig. 2, the vectors are:

- \( f = [50, 100, 500, 1000, 5000, 10000, 50000, 100000, 500000, 1000000] \) [Hz]
- \( T = [300, 400, 500, 600] \) [K]
- \( Moisture = 1:5:50 \) [%]

Vector Moisture is described in the Matlab/Octave expression and it is designation for 1, 1+5, 1+5+5, 1+5+5+5, etc.

Since related groups of materials exhibit similar dependences, it is easy to simulate dielectric constant dependence on frequency and temperature with specific moisture. After all moistures are taken into account, the 3D graph is obtained. The next step is to take account the range of possible values of \( \varepsilon_r \). The range of possible values for \( \varepsilon_r \) and consequently corresponding dielectric material is determined. In the example from Fig. 2, the optimal material should have the relative dielectric constant between 27 and 42 (if there is such a material in the domain set of materials). It should be noted that simulation of dependences can be performed only for the group of similar materials. If someone would like to know which group is better, the procedure should be repeated for all the groups of interest.

The proposed algorithm consists of several steps (see Fig. 3) and two pre-steps. The first pre-step is to form a lookup table, which consists of materials and values of the relative dielectric constant. Of course, the materials which cannot be used in such an application due to important reasons (such as galvanic currents, corrosion mechanisms, etc.) should not be entered into the lookup table. The second pre-step is to delete such materials from the lookup table. The second pre-step is to delete such materials from the lookup table.

Step 1: Input of necessary data. The necessary data are: operating frequency, expected temperature and moisture. If the actual computer realization does not include data such as constants, such data should be entered as well.

Step 2: The optimal solution is found by the expression:

\[ \varepsilon_{\text{optimal}} = \text{arg}\left\{ f_1(T), f_2(M), f_3(OF) \right\} \]  \hspace{1cm} (22)

where \( T \) stands for temperature, \( M \) for moisture, \( OF \) for operating frequency and \( f_i \) stands for dielectric constant dependence on temperature, \( f_2 \) for dielectric constant dependence on moisture and \( f_3 \) dependence on frequency.

Equation (22) is the variation for the typical optimal function definition. The difference from most of the references is in the function of optimization. Namely, the most common are maximum or minimum, such as minimization of oil consumption or profit maximization. In this case, intersection of the sets is used.

Step 3: Lookup table is used to find the optimal material. The value of dielectric constant is connected to the material.

Step 4: If the material with the desired dielectric constant exists in the lookup table, then the algorithm finishes. If it is not the case, one can choose to find optimal solution based on
two more important factors or by entering the maximum tolerable deviation from the desired values.

Step 5 (additional): If two factors are chosen, then the new intersection is calculated by these two parameters by:

$$
\varepsilon_{optimal} = \arg\cap \{f_x(x), f_y(y)\}
$$

where \( x \) is the first important factor, \( y \) the second, and \( f_x, f_y \) corresponding dependence functions.

![Proposed algorithm for optimal selection of dielectric materials](image)

**Fig. 3** Proposed algorithm for optimal selection of dielectric materials

Step 6 (additional): From lookup table, the optimal material is chosen. If there is no optimal material in the table, the maximum deviation from desired parameters’ values can be entered as the input to the next step.

Step 7 (additional): Intersection is calculated. If the optimal material exists, then the algorithm finishes successfully. However, if there is no match, the only conclusion is that there is no such material in the lookup table.
5. Conclusions

In this paper a new algorithm which can be used to find optimal material is presented. For marine purposes, dominant factors for materials’ selection are identified and analysed, but not specific factors, which depend on actual location of the material aboard ship (e.g. deck or engine room). However, the proposed algorithm can be used for other properties, such as deck salinity, oil and fuel impact, dust, fumes, cross-resistance to punctures, etc. There are just domain sets. Even more, the proposed algorithm can be extended to include all specific parameters by including more than three sets and finding the intersection between more sets. The final goal of the paper is to develop automated method for the material selection based on necessary parameters in specific applications. Further work should include such specific factors as: deck salinity, impact of oils, fuels, dust and fumes to cross-resistance to punctures, surfaces, etc. In some cases, such as in the case of an insulator or mount switch, the contact must be characterized by appropriate hardness and elasticity.

An approach to marine application is explained in the paper. Identified dominant factors are operating frequency, moisture and temperature. Theory of sets is used to find the optimal solution and lookup table to find the optimal material.

The possible shortcomings of the proposed are limitations in computer capacity and different dependences for the same parameters depending on material’s group.

The advantage is in the framework to find the optimal material, which is flexible to different inputs, such as ranges for frequency, temperature and/or moisture. This could help shipyards to make ships with higher additional value and quality.

The proposed algorithm can be used in different applications, because the input of ranges is dictated by the application and explicit expressions dictate different behaviours of different groups of materials.

For example, marine engineering is an interesting area of application, because the knowledge of several disciplines has to be considered in the materials’ selection, such as meteorology, law, electrical engineering, mechanical engineering, chemistry and physics. The materials in the marine environment are under impacts from atmospheric conditions, type of vessel, place of installation, special maritime regulations that have to be fulfilled, etc.

Finally, an example is simulated. The simulated case deals with the determination of the material for PCB based on relative dielectric constant.

REFERENCES


ANALYSIS OF THE ENERGY EFFICIENCY DESIGN INDEX WITH A PROPOSAL FOR IMPROVEMENT

UDC 629.5.01:629.03:629.5.014:629.5.011
Preliminary communication

Summary

This paper presents the actual method for the calculation of Energy Efficiency Design Index and analyses the influence of particular variables on the resulted EEDI. Perceived inconsistencies of the actual method of calculation of the attained EEDI are presented with explanation of the influence that the actual calculation has on the design of new ships. Objections are clearly demonstrated on the example of conceptual design of Handy Bulk Carrier. Alternative proposals for improvement of calculation of attained EEDI are introduced. Suggested alterations eliminate observed deficiencies of the existing calculation.

Key words: energy efficiency design index; ship design

1. Introduction

In 2011 International Maritime Organization adopted amendments to MARPOL Annex VI, according to which Energy Efficiency Design Index (EEDI) became mandatory for all new ships contracted on or after 1 January 2013 [1]. Amendments set measures to reduce emissions of greenhouse gases - reference lines for the calculation of the required EEDI for various types of ships and formula for calculation of the attained EEDI. Also, required EEDI reduction factors (improvements of design efficiency) were set for the period up to 1 January 2025.

2. Calculation of Attained and Required Energy Efficiency Design Index

Attained EEDI must be less than or equal to the value required for the particular period [1].

\[
\text{Attained EEDI} \leq \text{Required EEDI} = (1 - \frac{X}{100}) \times \text{Reference line value}
\]

where:

\[X\] – reduction factor for the required EEDI compared to EEDI Reference line

The reference line for required EEDI can be calculated as:

\[
\text{Reference line} = a \times b^c
\]
where:

- $b(t)$ – deadweight
- $a(\cdot)$, $c(\cdot)$ – non-dimensional parameters for the different ship types

Formula for calculation of attained EEDI is as follows [2,3]:

$$
\frac{P_{ME} \cdot C_{FME} \cdot SFC_{ME} + P_{AE} \cdot C_{FAE} \cdot SFC_{AE}}{f_i \cdot DWT \cdot V_{ref}}
$$

where:

- $P_{ME}$ (kW) – 75% of the selected maximum continuous rating (SMCR)
- $C_{FME}$ (-) – non-dimensional conversion factor between main engine fuel consumption and CO$_2$ emission
- $SFC_{ME}$ (g/kWh) – main engine specific fuel oil consumption
- $P_{AE}$ (kW) – required auxiliary engine power to supply normal maximum sea loading, calculated as:
  - $P_{AE} = 0.05 \times SMCR$ for SMCR < 10,000 kW
  - $P_{AE} = 0.025 \times SMCR + 250$ for SMCR $\geq$ 10,000 kW
- $C_{FAE}$ (-) – non-dimensional conversion factor between auxiliary engine fuel consumption and CO$_2$ emission
- $SFC_{AE}$ (g/kWh) – auxiliary engine specific fuel oil consumption
- $f_i = 1 + (0.08 \times LWT / DWT)$ – for ships built under Common Structural Rules
- $LWT$ – lightweight of the ship
- $DWT$ (t) – deadweight on summer load draught
- $V_{ref}$ (kn) – ship speed on deep water corresponding to $DWT$ and $P_{ME}$, no sea margin

Basic ally, attained EEDI represents the quantity of emitted CO$_2$ per deadweight tonne and knot.

3. Influence of Particular Variables on the Attained EEDI

It can be demonstrated that the formula for calculation of attained EEDI favours designs with low $P_{ME}$, i.e. with low SMCR. If we consider deadweight and $f_i$ as fixed, EEDI is a simple function linearly proportional to the main engine power $P_{ME}$, and inversely proportional to the ship’s speed $V_{ref}$. Knowing that power grows approximately with the cubic exponent of ship’s speed, it is easy to conclude that the numerator of expression grows much faster than the denominator. This method of EEDI calculation has forced designers to set continuous service rating (CSR) on the level of 90% SMCR, or very close to it. As a consequence, there appears to be a problem of 'shortage' of the main engine power which is evident especially when sailing in heavy seas.
4. Negative Consequences of the Existing Method of Calculation of the Attained EEDI

In the rough sea, a propeller curve in the power-speed diagram shifts to the left, where the main engine can deliver less power. In this situation safety of the ship is reduced and both crew and cargo are endangered. All of this has already been observed, and the major main engine manufacturers revised the recommendations for the propeller light running margin, so now they recommend to increase it [4]. Unfortunately, this recommendation leads to the increase of the attained EEDI. As a conclusion it can be said that the present method of calculation of EEDI has the following negative consequences:

1. Lower safety level of ship
2. Endangered crew and cargo in heavy seas

and, what is especially interesting
3. Lower EEDI does not necessarily mean lower fuel consumption

5. Case Study: Concept Design of Handy Bulk Carrier

To elaborate on the above statements, let us consider the following case. Conceptual design of two Handy Bulk Carriers has been developed following design procedure elaborated and published in [5,6,7,8]. Developed designs have basic characteristics [9]:

\[
\begin{align*}
D_W &= 35,000 \text{ t} \\
L_{pp} &= 176.0 \text{ m} \\
B &= 31.5 \text{ m} \\
d_s &= 10.2 \text{ m} \\
CSR &= 4,860 \text{ kW at 85.9 rpm} \\
\end{align*}
\]

5.1 Optional Main Engines

Main engine and SMCR are selected in two options:

A) MAN B&W 5S50ME-B9.5 TII

\[
SMCR = 5,400 \text{ kW at 89 rpm (CSR = 90\% SMCR), } P_{ME} = 4,050 \text{ kW (75\% SMCR)}
\]

B) MAN B&W 6S50ME-B9.5 TII

\[
SMCR = 7,477 \text{ kW at 99.2 rpm (CSR = 65\% SMCR), } P_{ME} = 5,608 \text{ kW (75\% SMCR)}
\]

Power-speed diagrams of the alternative main engines are shown in the following figures.
Fig. 1 Power-Speed Diagram of the 5S50ME-B9.5 TII Engine
As it can be seen, in both cases CSR is set at the same power and speed.

5.2 Calculation of the Attained EEDI

The calculation of the specific fuel oil consumption has shown the best results for the following cases:

A) For 5S50ME-B9.5, high load tuning, ISO conditions:
   - at SMCR 162.0 g/kWh
   - at CSR 160.0 g/kWh
   - for EEDI calculation 169.0 g/kWh
B) For 6S50ME-B9.5, low load, variable turbine area tuning, ISO conditions:
   at SMCR 163.3 g/kWh
   at CSR 156.3 g/kWh
   for EEDI calculation 169.3 g/kWh

   Speed-power estimation is shown in the following figure.

   ![Speed-Power Estimation](image)

   **Fig. 3 Speed-Power Estimation**

   It can be observed that the speed in the case with 5S50ME-B9.5 (75% SMCR = 4,050 kW) is abt. 13.84 kn and in the case with 6S50ME-B9.5 (75% SMCR = 5,608 kW) is abt. 15.25 kn.

   EEDI calculations are shown in the following figures.
Fig. 4 EEDI for 5S50ME-B9.5 TII Engine

Fig. 5 EEDI for 6S50ME-B9.5 TII Engine
Required EEDI for the phase 1 is 5.886. In the case of 5S50ME-B9.5 attained EEDI is 4.767 and it is much better than required one, even close to the limit set for the end of the monitoring period, i.e. for the phase 3. In the case of 6S50ME-B9.5 attained EEDI is 6.000 (worse for abt. 26% than for 5-cylinder engine), even not satisfying the phase 1 requirement.

5.3 Fuel Oil Consumption

Calculation of the main engine daily fuel oil consumption for the standard conditions (CSR, \(d_c\), no sea margin) is as follows:

A) For 5S50ME-B9.5:
\[
dfoc = 4,860 \text{ kW} \times 160.0 \text{ g/kWh} \times 24 \text{ hours} \times 10^{-6} = 18.66 \text{ t/day}
\]

B) For 6S50ME-B9.5:
\[
dfoc = 4,860 \text{ kW} \times 156.3 \text{ g/kWh} \times 24 \text{ hours} \times 10^{-6} = 18.23 \text{ t/day}
\]

It can be seen that version with 5-cylinder main engine has daily fuel oil consumption higher for abt. 0.43 t (abt. 2.4%) than the version with 6-cylinder engine. This situation indicates that the actual calculation of the attained EEDI is imperfect and that it would be advisable to improve it.

5.4 Safety Level of Ship

Analysis of safety level for both optional designs is shown in the following figures.

![Fig. 6 Available Main Engine Power in Heavy Seas (for 5S50ME-B9.5 TII)](image-url)
As it can be seen, when sea margin is 30%, 5S50ME-B9.5 can deliver only abt. 4,360 kW, while 6S50ME-B9.5 can deliver abt. 6,040 kW. When sea margin is 50%, 5S50ME-B9.5 can deliver only abt. 3,270 kW, while 6S50ME-B9.5 can deliver abt. 4,540 kW.

6. Proposal for Improvement of Calculation of the Attained EEDI

Hereinafter are presented two proposals for correction of the calculation of the attained EEDI. Both proposals are made in a way to minimize changes of the existing calculation and to achieve the desired effect to balance calculated EEDI with the main engine fuel consumption. Also, intention of the proposed corrections is to encourage the development of designs of a higher level of safety of the ship, crew and cargo.

Proposal 1: to define $P_{ME}$ as 83.33% of CSR, instead of 75% of SMCR, i.e.:

$$P_{ME} = \frac{0.75}{0.9} \cdot CSR = 0.8333 \cdot CSR$$

Why 83.33% of CSR? As previously explained, actual EEDI calculation favours selection of MCR as low as possible. In the extreme case when $CSR/SMCR = 0.9$, 75% of SMCR corresponds to 83.33% of CSR.
In the shown optional designs, this proposal will result with the same result for the alternative A and significantly better result for the alternative B (main engine specific fuel oil consumption at 4,050 kW + 6% margin is about 167.5 g/kWh, $P_{AE} = 0.05 \times SMCR = 373.85$ kW, remained the same)

\[
P_{MEA} = P_{MEB} = 83.33\% \times CSR = 0.8333 \times 4,860 = 4,050 \text{ kW}
\]

attained $EEDI_A = 4.767$

\[
\text{attained } EEDI_B = \frac{4050 \cdot 3.206 \cdot 167.5 + 373.85 \cdot 3.206 \cdot 185}{1.0196 \cdot 35000 \cdot 13.84} = 4.852
\]

It can be seen that proposed calculation of EEDI gives very close results for both options – attained EEDI for option B is slightly worse mainly because of higher auxiliary engine power.

**Proposal 2**: to define $P_{ME}$ as needed power to reach referent speed $V_{ref}$. Conditions for the definition of the power-speed characteristic remain unchanged: summer load line $d_s$, deep water, no sea margin. Referent speed $V_{ref}$ must be defined in relation to the type and size of the vessel.

Just for illustration purposes, let us set $V_{ref}$ for bulk carriers of 10,000 dwt and above as follows:

\[
V_{ref} = 6.75 + 1.5625 \cdot \log DWT
\]

The proposed formula is created in such a way that small bulk carriers of 10,000 dwt have referent speed of 13 knots, handy size bulkers of about 14 knots, and the largest VLOC of about 15.5 knots, which can be considered as appropriate speed for corresponding bulk carrier’s sizes. In the following table referent speeds for typical deadweight are shown.

**Table 1** Proposed referent speeds $V_{ref}$

<table>
<thead>
<tr>
<th>$DWT$ (t)</th>
<th>$V_{ref}$ (kn)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10,000</td>
<td>13.00</td>
</tr>
<tr>
<td>30,000</td>
<td>13.75</td>
</tr>
<tr>
<td>50,000</td>
<td>14.09</td>
</tr>
<tr>
<td>200,000</td>
<td>15.03</td>
</tr>
<tr>
<td>400,000</td>
<td>15.50</td>
</tr>
</tbody>
</table>

For the optional designs referent speed can be calculated as follows:

\[
V_{ref} = 6.75 + 1.5625 \log (35000) = 13.85 \text{ (kn)}
\]

which is very close to the speed estimated for the main engine power of 4,050 kW (13.84 kn), with marginal influence on the EEDI calculation.
7. Conclusion

The presented improvement of calculation of the attained EEDI eliminates detected inconsistencies of the actual method. Higher safety level of ship, cargo and crew can be achieved without significant deterioration of the attained EEDI. Also, proposed calculation enables realization of ship designs with lower fuel oil consumption keeping the attained EEDI on the same level as the actual one. Of course, presented corrections should be considered only as a starting point for the development of a new calculation of attained EEDI for all types of ships covered by the regulations.

REFERENCES

SWOT ANALYSIS OF DEFICIENCIES ON SHIP COMPONENTS IDENTIFIED BY PORT STATE CONTROL INSPECTIONS WITH THE AIM TO IMPROVE THE SAFETY OF MARITIME NAVIGATION

UDC 629.5.072:629.5.078.4:629.5.011
Review paper

Summary

This paper analyses components of the ship system with regard to the recommendations that have impact on the safety of maritime navigation according to the evaluation of port control inspections. What the research particularly examined are recommendations (deficiencies), i.e. cases when ships are not allowed to proceed to sea and continue their navigation. The results of the comparative and SWOT analyses are presented, as well as the functional dependence (tendency) of not complying with specific ship components concerning the age of the ship. Both the results of the research and the performed analyses show that the majority of recommendations concern the emergency systems, i.e. the safety of navigation and rescue appliances, while the least number of recommendations has been recorded in the training and drills of the crew for emergency situations. What has also been established is that by a better on-board training of the crew and better education of seafarers it is possible to reduce the number of recommendations (deficiencies). The age of the ship, which has also been found to contribute to the increasing number of recommendations (deficiencies), constitutes the objective reason for this due to the current unsatisfactory financial situation in the market.

Key words: port control inspections, recommendations, deficiencies, tendencies, SWOT analysis

1. Introduction

Port State Control (PSC) is the inspection of foreign ships in national ports, carried out by authorised port State control inspectors to verify that the competences of the master and the crew, the condition of the ship and its equipment comply with the requirements of international conventions and regulations [1]. It is only the authorised Port State Control inspectors who carry out these inspections in Croatia. However, it has to be said that the inspection that takes place cannot be considered to be the port State control [2] when the ship comes into the port of the state in which it was registered.
Although the responsibility for the implementation of international conventions and regulations has been ceded to the “flags” of ships, i.e. authorities that provide the ship’s registration, port State authorities have the right and duty to inspect foreign ships in national ports [3]. It is common that regionally connected states associate and sign the memorandum of understanding as in this way the inspection of one of the state signatories of the memorandum, is recognised in all other states signatories of the same memorandum. A Memorandum of Understanding (MoU) is a document that defines a relationship between signatories, being less formal than a contract. It is thus avoided that the inspection of the port State control takes place in every port the ship calls at. There are ten such associations in the world, Croatia being the signatory of the Paris Memorandum of Understanding (Paris MoU). The Paris MoU has been signed by 27 states, namely European Union coastal states, joined by Canada and Russia (i.e. its ports in the European part of Russia). Besides the Paris MoU, the best known PSC associations are the Tokyo MoU, the Indian Ocean MoU, the Mediterranean MoU, the US Coast Guard and the Caribbean MoU [4].

As Croatia is the signatory of the Paris Memorandum and makes part of the Paris MoU organisation, this paper will deal with the principles of work of port States controls in the countries signatories of this Memorandum only, presenting the results of the research regarding port State controls in the period from the year 2011 to the year 2013, as well as the results of the research concerning the same period dealing with the grounds for the detention of ships, when, according to the evaluation of inspection, the identified recommendations significantly decreased the ability of the ship for the continuation of navigation and execution of a specific task. In the case of detention, the ship is not allowed to proceed to sea and continue navigation until deficiencies have been rectified, such cases being specifically analysed in this paper.

Using the tabular display, data on the number of inspections and the number of detentions of ships in the period from the year 2011 to the year 2013 are presented, while it is by diagrams (graphs) that tendencies of a specific component (which caused the detention of the ship) regarding the age of the ship are shown. The SWOT analysis is presented at the end of the paper.

2. Port Authority Inspections of Ships

Fundamental principles governing the inspection of a ship by port authorities are the same worldwide or very similar. However, there are differences in the statistical processing and the interpretation of deficiencies that have been identified during the inspection [5]. Accurate records are kept on inspections performed by a port State control, the data of which are then recorded on the central computer database partly available online [1], namely on the official Paris MoU website, where statistically processed data for each month and each year respectively can be found. The findings recorded upon the inspection of a ship by a port State control can be the following:

a) Upon the performed inspection no recommendations have been recorded;

b) Upon the performed inspection deficiencies have been identified, however, according to the evaluation of the inspection, they do not have impact on the ability of a ship to execute a specific task (in this case the ship is allowed further exploitation under the condition that identified deficiencies be rectified in the time given) and

c) The inspection has been performed identifying deficiencies, which, according to the evaluation of the inspection, do have impact on the ability of the ship to proceed to sea and
execute a specific task (in which case the ship is forbidden to continue navigation until the
deficiencies have been rectified).

What was analysed were only the cases found under c), Table 1 presenting the number
of inspections and the number of ship detentions in the period from the year 2011 until the
year 2013[6].

<table>
<thead>
<tr>
<th>Year</th>
<th>2011</th>
<th>2012</th>
<th>2013</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of inspections</td>
<td>19,058</td>
<td>18,308</td>
<td>17,687</td>
</tr>
<tr>
<td>Number of individual ships inspected</td>
<td>15,268</td>
<td>14,646</td>
<td>14,108</td>
</tr>
<tr>
<td>Number of detentions</td>
<td>688</td>
<td>669</td>
<td>668</td>
</tr>
<tr>
<td>Number of deficiencies</td>
<td>50,738</td>
<td>49,261</td>
<td>49,074</td>
</tr>
<tr>
<td>Number of detention related deficiencies</td>
<td>2,414</td>
<td>2,691</td>
<td>3,057</td>
</tr>
</tbody>
</table>

(Source material: www.parismou.org)

The Paris MoU organisation has divided detention related deficiencies into 18
categories, the list of which is presented in Table 2 [7].
Table 2  Deficiencies categorised according to the Paris MoU organisation

<table>
<thead>
<tr>
<th>Number of category</th>
<th>Category of deficiencies</th>
</tr>
</thead>
<tbody>
<tr>
<td>01</td>
<td>Certificates &amp; Documentation</td>
</tr>
<tr>
<td>02</td>
<td>Structural condition of the hull</td>
</tr>
<tr>
<td>03</td>
<td>Water / weathertight condition</td>
</tr>
<tr>
<td>04</td>
<td>Emergency systems</td>
</tr>
<tr>
<td>05</td>
<td>Radio communication appliances</td>
</tr>
<tr>
<td>06</td>
<td>Cargo operations including equipment</td>
</tr>
<tr>
<td>07</td>
<td>Fire safety</td>
</tr>
<tr>
<td>08</td>
<td>Different alarms</td>
</tr>
<tr>
<td>09</td>
<td>Working and living conditions</td>
</tr>
<tr>
<td>10</td>
<td>Safety of navigation appliances</td>
</tr>
<tr>
<td>11</td>
<td>Life saving appliances</td>
</tr>
<tr>
<td>12</td>
<td>Dangerous goods appliances</td>
</tr>
<tr>
<td>13</td>
<td>Propulsion and auxiliary machinery</td>
</tr>
<tr>
<td>14</td>
<td>Sea pollution prevention appliances</td>
</tr>
<tr>
<td>15</td>
<td>ISM Code – International Safety Management Code for the safe management and operation of ships and for pollution prevention</td>
</tr>
<tr>
<td>16</td>
<td>ISPS Code - The International Ship and Port Facility Security Code</td>
</tr>
<tr>
<td>17</td>
<td>Other</td>
</tr>
</tbody>
</table>

(Source material: www.parismou.org)

The above-mentioned deficiencies from Table 2 focus on 11 components of the ship [8], i.e.:

1) Group of technical components:
   a) Ship hull and watertight condition;
   b) Propulsion and auxiliary machinery, alarms;
   c) Emergency systems;
   d) Structural fire safety;
   e) Safety of navigation and life saving appliances and
   f) Sea pollution prevention appliances.

2) Group of administrative components:
   a) Master and crew certificates, as well as
   b) Ship certificates and mandatory manuals.

3) Group of other components:
   a) Training of the crew for emergency situations;
   b) ISM Code and
c) Working and living conditions on board.

Proceeding from the above-mentioned classification and on the basis of our analysis of the report on detention of ships by inspections of port State controls that are members of the Paris MoU organization, diagrams have been made presenting the functional dependence of the number of detentions (in percentages) with regard to the age of the ships for the period from 2011 until 2013.

3. Analysis of Functional Dependence

The above-mentioned functional dependence is in fact the tendency of a specific ship component and can be defined as the ratio of the number of detentions performed on the grounds of deficiencies identified on that component and the total number of inspected ships in the first 36 years of the age of the ship. Diagrams (graphs) obtained in this way (Figures 1 – 3) show the tendency for each component, focusing on a specific technical, administrative or other component. These diagrams have been made on the basis of the carried out research on the detentions of ships in the period from 2011 to 2013.

**Fig. 1** Diagrams of tendencies for technical ship components
(Source material: copyright work based on the carried out research)
Fig. 2 Diagrams of tendencies for administrative ship components
(Source material: copyright work based on the carried out research)

Fig. 3 Diagrams of tendencies for other ship components
(Source material: copyright work based on the carried out research)

Functional dependence (tendencies) of ship components presented in Figures 1, 2 and 3 can be approximated by quadratic functions, the results of which are shown in Table 3.

On the basis of quadratic approximations shown in Table 3, Figure 4 shows tendencies of detentions of ships due to deficiencies on a specific component concerning the age of the ship.
### Table 3 Functional dependence (tendencies) of ship components

<table>
<thead>
<tr>
<th>Component</th>
<th>Approximation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1A - Ship hull and watertight condition</td>
<td>( y = 0.0052t^2 + 0.0132t + 0.0816 )</td>
</tr>
<tr>
<td>1B - Propulsion and auxiliary machinery, alarms</td>
<td>( y = -0.0030t^2 + 0.2033t - 0.3531 )</td>
</tr>
<tr>
<td>1C – Emergency systems</td>
<td>( y = -0.0006t^2 + 0.1202t + 0.1944 )</td>
</tr>
<tr>
<td>1D – Structural fire safety</td>
<td>( y = 0.0007t^2 + 0.1835t + 0.3848 )</td>
</tr>
<tr>
<td>1E - Safety of navigation and life saving appliances</td>
<td>( y = 0.0074t^2 + 0.0158t + 1.7771 )</td>
</tr>
<tr>
<td>1F - Sea pollution prevention appliances</td>
<td>( y = -0.0010t^2 + 0.1239t - 0.0901 )</td>
</tr>
<tr>
<td>2A - Master and crew certificates</td>
<td>( y = 0.0043t^2 - 0.0943t + 0.8949 )</td>
</tr>
<tr>
<td>2B - Ship certificates and mandatory manuals</td>
<td>( y = 0.0062t^2 - 0.0818t + 1.5450 )</td>
</tr>
<tr>
<td>3A – Training of the crew for emergency situations</td>
<td>( y = 0.0019t^2 - 0.0000t + 0.1724 )</td>
</tr>
<tr>
<td>3B - ISM Code</td>
<td>( y = 0.0007t^2 + 0.2929t + 0.8714 )</td>
</tr>
<tr>
<td>3C - Working and living conditions on board</td>
<td>( y = 0.0005t^2 + 0.0676t + 0.2430 )</td>
</tr>
</tbody>
</table>

(Source material: copyright work based on diagrams in Figures 1, 2 and 3)

**Fig. 4** Diagram showing the percentage of detained ships with regard to the age of the ship according to components

(Source material: copyright work based on diagrams in Figures 1, 2 and 3)
What follows are the results of this research and the performed comparative analysis.

4. Results of Comparative Analysis

On the basis of the diagram in Figure 4, it is possible to conclude that the biggest number of recommendations made according to the evaluation of inspection, namely recommendations which affect the ability of the ship to continue navigation and perform a specific task are those which concern emergency systems, where the functional dependence increases in the greatest degree with the increase of the age of the ship (having almost a linear dependence, Function 1C). The second place is taken by recommendations concerning the safety of navigation and life saving appliances (dependence being exponential, Function 1E), while the smallest increase regarding the age of the ship has been recorded with the training of the crew for emergency situations (Function 3A).

Because of the comparative analysis of trends, the coordinate system shown in Figure 4 is divided in four quadrants determined by directed lines A and B. The directed line A presents the calculated mean value of all recommendations and amounts to 2.78%, while the directed line B presents half the period of the identified age of the ship (18 years).

Figure 5 presents a diagram with six characteristic cases concerning the relation between the age of the ship and the percentage of detentions for a specific component. The tendency of each component from the diagram in Figure 4 can be assigned to one of the six characteristic cases. Characteristic cases are depicted by a quadrant in which they start, by their tendency and the quadrant in which they end.

Characteristic cases are as follows:

**Characteristic case 1:** The component starts in quadrant II and ends in quadrant III, possessing a high, above-average number of deficiencies during the whole of the ship's life cycle.

![Fig. 5 Diagrams of characteristic cases](Source material: copyright work)
**Characteristic case 2:** The component starts in quadrant II, goes through quadrant III and ends in quadrant IV. In the beginning of exploitation there is a high number of deficiencies, this number decreasing slowly over time.

**Characteristic case 3:** The component starts in quadrant II, goes through quadrant I and ends in quadrant IV. In the beginning of exploitation there is a high number of deficiencies, this number decreasing fast during the exploitation.

**Characteristic case 4:** The component starts in quadrant I, goes through quadrant II and ends in quadrant III. In the beginning of exploitation this component has a small number of deficiencies, however, the number of deficiencies increases fast during the exploitation.

**Characteristic case 5:** The component starts in quadrant I and goes through quadrant IV, ending in quadrant III. In the beginning of exploitation, this component has a small number of deficiencies, this number increasing during the exploitation.

**Characteristic case 6:** The component starts in quadrant I and ends in quadrant IV. During the whole of the ship's life cycle, this component has a small, below-average number of deficiencies.

On the basis of the comparative analysis (diagrams in Figures 4 and 5), it can be concluded that in the analysed example this concerns cases from 4 to 6, as shown in Table 4. Not one component in the analysed example has been found to have acquired the characteristic cases from 1 to 3, the common characteristic of which is a high number of deficiencies in the beginning of ship’s exploitation.

### Table 4 Results of the analysis of characteristic cases

<table>
<thead>
<tr>
<th>Component</th>
<th>Grade</th>
</tr>
</thead>
<tbody>
<tr>
<td>Characteristic case 1</td>
<td>Nil</td>
</tr>
<tr>
<td>Characteristic case 2</td>
<td>Nil</td>
</tr>
<tr>
<td>Characteristic case 3</td>
<td>Nil</td>
</tr>
<tr>
<td>Characteristic case 4</td>
<td>1D, 1E, 3B</td>
</tr>
<tr>
<td>Characteristic case 5</td>
<td>1A, 1B, 1C, 1F, 2A, 2B, 3C</td>
</tr>
<tr>
<td>Characteristic case 6</td>
<td>3A</td>
</tr>
</tbody>
</table>

(Source material: copyright work)

It is beyond dispute that it can be affirmed and concluded that the greatest number of deficiencies occur in components 1D, 1E and 3B, namely, the component has a small number of deficiencies in the beginning of exploitation, however, this number of deficiencies rapidly increases with the age of the ship.
5. Results of the SWOT Analysis

The SWOT analysis is concerned with the analysis of internal processes and external environment with the aim of identifying its internal strengths in the process in order to take advantage of its external opportunities and avoid its external threats, while addressing its inner weaknesses. This technique is attributed to Albert Humphrey, who led a research project at Stanford University in the 1960s and 1970s of the last century. The SWOT analysis is used to evaluate the Strengths, Weaknesses, Opportunities, and Threats involved in a project, having received its name after the first letters of the analysed factors [9].

Proceeding from the research results and the performed comparative analysis with the aim to determine strengths, weaknesses, opportunities and threats, a SWOT analysis has been made [10] [11], the results of which are shown in Table 5.

<table>
<thead>
<tr>
<th>STRENGTHS</th>
<th>WEAKNESSES</th>
</tr>
</thead>
<tbody>
<tr>
<td>On-board training of the crew</td>
<td>Insufficient number of educated seafarers on board</td>
</tr>
<tr>
<td>Ship maintenance procedures</td>
<td>Poor motivation of the crew due to bad financial conditions</td>
</tr>
<tr>
<td>Dissemination of information on new regulations</td>
<td>Careless service provided for the ship by agents</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>OPPORTUNITIES</th>
<th>THREATS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Education of the crew</td>
<td>Sudden breakdown of appliances, the ones on which direct impact cannot be made</td>
</tr>
<tr>
<td>Ship maintenance with services that are expert in specific areas</td>
<td>Insufficient inflow of educated seafarers</td>
</tr>
<tr>
<td></td>
<td>Age of the ship</td>
</tr>
<tr>
<td></td>
<td>Poor ship maintenance due to ever more difficult financial conditions of shipping companies</td>
</tr>
</tbody>
</table>

(Source material: [11] and copyright work)

Table 6 shows the matrix of reciprocal impacts of SWOT factors upon the tendency of a specific component. The impact of each component according to the analysis of characteristic cases (Table 4) was marked with grades from 1 to 6. Grade 1 means that a small number of deficiencies were recorded on the component, according to the diagram in Figure 4, the component consequently belonging to Case 6. In the same way grades were given for components that make part of Case 5 (grade 2), Case 4 (grade 3), Case 3 (grade 4), Case 2 (grade 5) and Case 1 (grade 6). Not one component in this research has been found to have acquired the characteristic cases from 1 to 3.

The impact of a specific SWOT factor on the occurrence of deficiencies in a specific component was marked with grades from 1 to 5. Grade 1 means that the SWOT factor has a
small impact on the occurrence of deficiencies in a specific component; while grade 5 means that the SWOT factor has a great impact on the occurrence of deficiencies in a specific component.

Table 6 The matrix of reciprocal impacts of SWOT factors upon the tendency of a specific component

<table>
<thead>
<tr>
<th>SWOT factor</th>
<th>Component / Impact</th>
<th>Σ</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1D / 3</td>
<td>1E / 3</td>
</tr>
<tr>
<td>On-board training of the crew</td>
<td>3 5 5 2 5 5 4 5 10</td>
<td>106</td>
</tr>
<tr>
<td>Ship maintenance procedures</td>
<td>2 5 5 3 5 4 4 3 3</td>
<td>95</td>
</tr>
<tr>
<td>Dissemination of information on new regulations</td>
<td>5 5 5 3 2 2 3 3 2 3</td>
<td>86</td>
</tr>
<tr>
<td>Insufficient number of educated seafarers on board</td>
<td>5 5 5 4 4 4 4 3 3</td>
<td>100</td>
</tr>
<tr>
<td>Poor motivation of the crew due to bad financial conditions</td>
<td>3 4 3 3 3 3 3 4 3</td>
<td>71</td>
</tr>
<tr>
<td>Careless service provided for the ship by agents</td>
<td>5 3 3 5 4 3 4 5 3 1</td>
<td>92</td>
</tr>
<tr>
<td>Education of the crew</td>
<td>4 3 5 3 4 5 5 4 3 3</td>
<td>95</td>
</tr>
<tr>
<td>Ship maintenance with services that are expert in specific areas</td>
<td>3 5 3 2 5 2 5 1 1 1</td>
<td>68</td>
</tr>
<tr>
<td>Sudden breakdown of appliances, the ones on which direct impact cannot be made</td>
<td>1 5 3 5 5 5 5 1 1 3 3</td>
<td>80</td>
</tr>
<tr>
<td>Insufficient inflow of educated seafarers</td>
<td>3 5 3 4 4 4 4 3 2</td>
<td>88</td>
</tr>
<tr>
<td>Age of the ship</td>
<td>5 5 4 5 5 5 1 5 1</td>
<td>101</td>
</tr>
<tr>
<td>Poor ship maintenance due to difficult financial conditions of shipping companies</td>
<td>5 4 4 5 3 3 1 1 5 3</td>
<td>88</td>
</tr>
</tbody>
</table>

(Source material: [12] and [13] and copyright work)

According to Table 6 it is clear that factors that have the greatest impact are the on-board training of the crew, the insufficient number of educated seafarers on board and the age of the ship. The assessment of the impact of each SWOT factor can be ascribed to the subjective judgment of the authors, this subjective judgement being the only limitation of this method.
6. Conclusion

This paper presents the results of the research and the carried out analysis of recommendations that according to the evaluation of the inspection have impact on the ability of a ship to proceed to sea and execute a specific task (in which case the ship is forbidden to continue navigation until the deficiencies are rectified) in compliance with the criteria of the Paris Memorandum (Paris MoU). Eleven components of the ship system were singled out, for each of them a comparative and SWOT analysis made. The function of tendency of deficiency incidence regarding the age of the ship was calculated for each component, while all the components were divided into three groups with regard to the detention related impact they possess.

What was also made was the SWOT analysis of internal and external factors that can have impact on the incidence of deficiencies on components during the port State control inspection, while three factors were singled out, namely those have the greatest impact on the incidence of deficiencies, on-board training of the crew, insufficient number of educated seafarers on board and the age of the ship. Two of these factors concern the crew of the ship while one regards the characteristics of the ship.

On the basis of the presented results and the performed analysis, it can be concluded that by a better on-board training of the crew and a better education of seafarers it is possible to decrease the number of deficiencies, the most significant being the structural fire safety and safety of navigation and life saving appliances. The age of the ship is linked with the financial situation in the market, so that until financial conditions of shipping companies running a business are improved, this factor will present a significant influence. The introduction of new ships will certainly help decrease deficiencies that are identified by port control during the inspection.

The presented research results and analyses can be of help to shipping companies in undertaking measures with the help of which the safety of ships will be increased, that is, the requirements of fulfilling international convention that Croatia accepted met, while we hold it necessary to direct further research to the area of the design and building of ships, as well as to the contemporary forms of education and training of seafarers.

ACKNOWLEDGMENTS

We express our gratitude to Mr. Mladen Mandić from the Ministry of Maritime Affairs, Transport and Infrastructure for his great help in our collecting data on inspections of ships by port State controls.

REFERENCES

[1] www.parismou.org


CONSIDERING ANTI-PIRACY SHIP SECURITY: CITADEL DESIGN AND USE

UDC 629.5.067;629.5.047:629.5.043
Professional paper

Summary

As piracy continues to pose a threat to the shipping industry, a number of measures for protecting ships, cargo and crew will need to be implemented. Along with other steps, such as deploying military rescue teams, securing the crew within a ship’s citadel has proved to be a highly effective form of self-protection against hostage-taking by pirates. From a design standpoint, aspects that should be considered include the location and dimensions of the citadel, the maximum time crew can stay within it and the specific requirements for various elements of the ship or citadel equipment. Exploiting data on pirate attacks recently occurring in High Risk Areas, this article analyses the design and use of the citadel as a self-protection measure against piracy. As a conclusion, various requirements for these structures are recommended.

Keywords: piracy; hostage-taking; self-protection; citadel

1. Introduction.

The fact that piracy is threatening the shipping industry makes the news. However, crew safety has always been a major concern in navigation and piracy should not be considered a new phenomenon. It is an activity whose intensity responds to a typical pattern.

Many factors facilitate piracy: the shipping business is thriving, making attacks profitable. Pirates also enjoy easy access to technology and there is a lack of effective protective measures.

If the combined effect of these factors remains below a certain level, anti-piracy measures are not stepped up. It is felt that the consequences of piracy on maritime trade do not justify further action. On the other hand, when these factors reach a certain level and pirate attacks increase, they are then considered a significant threat to shipping and to the safety of seafarers. Protective measures should then be increased in proportion to existing pirate activity. Moreover, this effort must be kept up over time to reach the level of security and confidence necessary for the shipping business.

Stealing a ship’s cargo has always been the primary objective of piracy [1], [2]. With an increase in the volume of goods transported by sea over recent decades [3], the instability of riparian countries along maritime routes of communication and the easy access pirates have to technology, this activity has spiralled.
From 2006, some coastal states, especially in the Gulf of Aden and, to a lesser extent, in the Gulf of Guinea, do not monitor maritime zones effectively. As a result, pirates operate in a different way; in turn the world alters how it perceives this phenomenon. Cargo theft is no longer the sole purpose of pirates; a rescue operation is sought instead [3], [2]. Hostages are taken to increase the amount demanded for ransom, as well as the chances of the operation being a success. Protecting the crew now becomes a priority, to prevent pirates from seizing control of the ship as well as to protect the lives of seafarers.

Leaving aside political and economic measures to eradicate poverty and stabilize the countries in which pirate organisations flourish, the international community, along with shipping companies, has implemented a set of measures to avoid attacks or minimize their effects. These include deploying naval forces, finding alternative routes, organising convoys or establishing specific traffic control devices [4]. Self-protection measures must also be carried out on ships. Some are nonlethal, such as carrying out evasive manoeuvres or using citadels as well as fire hoses, sliding paint, acoustic devices, fences and other objects on the flanks of a vessel. Finally the creation of decisions support systems (like SARGOS system), involve the development of an overall protection method, automatic threat detection and identification, risk assessment and management of an appropriate response [5]. Other measures are lethal, with private, armed teams coming on board.

In the area around the Gulf of Aden, a combination of all these measures, with naval forces present and private armed teams coming on board, has proved to be very effective. The number of successful pirate attacks has gone down, but, these measures are not universally embraced. For one thing, they are costly. Ship owners may also be reluctant to have weapons on board; indeed, doing so is prohibited by law in some countries [6], [3], [4]. The number of attacks was expected to be significantly reduced and statistics show this is already the case in the Gulf of Aden. As a result, the most costly or controversial measures have not been deemed as necessary. However, it seems desirable to maintain medium and long -term non-lethal self-protection measures, which are less expensive and free from controversy. Another point is that piracy has indeed accelerated in other regions.

In any case, both in high-risk areas (HRA) and in those in which it has been possible to reduce this risk, adequate self-protection measures should be kept up. Among nonlethal measures, the citadel is the last defence. When other strategies have failed, citadels make it possible to stop pirates from gaining access to the control bridge, propulsion and crew. This article discusses the use, features and equipment that a citadel must have for the dual purpose of protecting the crew and blocking the pirates’ access to the vital systems of the ship.

2. Evolution of high risk areas (HRA).

At the beginning of the XXI century, piracy had very low activity levels (Fig. 1), and was therefore met with the indifference by the international community. Only the region of Southeast Asia experienced a high number of attacks, the main objective of these was not hostage – taking.
However, getting a ransom in exchange for the release of a ship, cargo and crew became the main objective of Somali pirates between 2006 and 2011 [6]. Piracy off the coast of Somalia evolved from a disorganized activity into a perfectly planned and developed criminal enterprise. Somali pirates were able to improve their skills, expanding their area of operation to areas well off the coast.

The number of attacks and kidnappings increased significantly, causing notable losses to shipping [8], [3] and raising serious concern among the international community.

Since 2010, however, piracy off Somalia has decreased (Fig. 1) because supranational protective measures were implemented. Proved to be highly effective, these measures included implementing Best Management Practices on ships (BMP) [9] and using an increasingly strong multinational naval presence and a growing number of armed, private security teams on board (PCASP) [2], [10], [6].

At the same time, an alarming trend in piracy is the way in which HRA has grown in the Gulf of Guinea. In 2003, the number of piracy acts far exceeded the figures for Somalia. While a slight decrease occurred in 2007, since then, there has been a return to previous levels (Fig. 1). Moreover, the pirates in that area have changed their mode of action. Before 2007, attacks mainly took place within territorial waters, where maritime domain could be exercised to some extent. Since that year, attacks have occurred more frequently and in more remote areas off the coast [5].

These pirates have adopted the performance principles of their Somali counterparts, such as kidnapping the crew to get ransom. On the other hand, some of the countries in this area forbid private armed personnel to be shipped out. Within the waters of the coastal countries, they require private security company workers or armed personnel to hold their nationality [11]. This requirement leads to inefficiency and corruption and makes it extremely difficult to apply effective protective measures against piracy [6]. Therefore, the HRA are considered to be in even greater danger [7].

In response to piracy off the coast of Somalia, the International Maritime Organisation (IMO) produced the circular MSC.1339 2011 [12] to stress the importance of implementing and updating the Best Management Practice Guidance BMP MSC.324 1989 [13]. INTERTANKO (independent tanker owners and operators of oil and chemical tankers), OCIMF (Oil Companies International Marine Forum), and other industries were represented by the International Chamber of Shipping (ICS), which drafted the document BMP4 (Best Management Practices) [9]. This is a practice guide relevant to ship owners, operators, shipmasters and vessel crews.

The document BMP4 [9] provides the three fundamental pillars in the self-defence of a ship. The first two are administrative measures: previously registering the path - MSCHOA and vessel position - UKMTO. The third pillar is known as ship protection measures – SPM-in which the document states [9], p. 23: “The Ship Protection Measures described in BMP are the most basic that are likely to be effective. Owners may wish to consider making alterations to the vessel beyond the scope of this booklet, and/or provide additional equipment, and/or manpower as a means of further reducing the risk of piracy attack. If pirates are unable to board a ship they cannot hijack it.”

This document also mentions the convenience of having a safe meeting place and a citadel as a last layer of security to protect the crew [9], p. 37: “A Safe Muster Point is a designated area chosen to provide maximum physical protection to the crew, preferably low down within the vessel”. Similarly, the need to accommodate the entire crew is noted [9], p. 38: “A Citadel is a designated pre-planned area purpose built into the ship where, in the event of imminent boarding by pirates, all crew will seek protection. A Citadel is designed and constructed to resist a determined pirate trying to gain entry for a fixed period of time.”

The aim of the citadel is to prevent the crew from falling into the hands of aggressors. Two goals are sought: preserving the vessel’s integrity and stopping the pirates from operating and steering it themselves [9].

Some consider self-protection measures like the citadel as only a short-term solution [8]. However, people in industry have a different perception. In this sense, the appendix “Guidance relating to the construction and use of citadels in waters affected by Somalia Piracy” [14], p.1 states that “…although the initial use of the citadel was limited to the Gulf of Aden, which could be a quick response of the naval forces, now its use has spread to other areas outside the Indian Ocean”.

Now that a period of time has passed since the BMP was implemented [9], it would be interesting to analyse how effectively citadels have been designed and used. Table 1, 2, 3 and 4 have been compiled from the 1,690 reports collected from pirate attacks [15], [16], [17], [18], [19]. Only reported incidents have been covered and therefore they only represent a partial number of the total [20].

Using data from the last five years, Table 1 shows that, when ships were attacked by pirates and the crew took refuge in the citadel, in just 5.5% of cases did the pirates take control of the ship. The reason a citadel went unused in the first place may be that only some of the crew could reach it or the citadel was not properly designed in the first place. In both cases the result was the same. The crew could not stay safely in the citadel.
Table 1: Citadel use during acts of piracy and robbery. Data collected by the ICC-IMB, indicating those cases where it failed to protect the crew. Source: author’s own information and IMB in [15], [16], [17], [18], [19].

<table>
<thead>
<tr>
<th>CASES IN WHICH CITADEL USED</th>
<th>2010</th>
<th>2011</th>
<th>2012</th>
<th>2013</th>
<th>2014</th>
<th>TOTAL/MEDIA</th>
</tr>
</thead>
<tbody>
<tr>
<td>CITADEL USED IN TOTAL</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>NUMBER OF CASES FOR YEAR (%)</td>
<td>4.3%</td>
<td>12.5%</td>
<td>8.4%</td>
<td>6.1%</td>
<td>5.7%</td>
<td>7.6%</td>
</tr>
<tr>
<td>CITADEL FAILURE RESULTING IN CREW BEING KIDNAPPED OR HARMED</td>
<td>1</td>
<td>2</td>
<td>1</td>
<td>2</td>
<td>1</td>
<td>7</td>
</tr>
<tr>
<td>CITADEL FAILURE IN TOTAL NUMBER OF CASES CITADEL USED (%)</td>
<td>5.2%</td>
<td>3.7%</td>
<td>4%</td>
<td>12.5%</td>
<td>7.1%</td>
<td>5.5%</td>
</tr>
<tr>
<td>TOTAL NUMBER OF ACTS OF PIRACY</td>
<td>445</td>
<td>439</td>
<td>297</td>
<td>264</td>
<td>245</td>
<td>1690</td>
</tr>
</tbody>
</table>

If only some of the crew is confined, pirates can take hostages and the effectiveness of the citadel is impairs (Table 1). Once the crew is outside, it is captured by the raiders. It is therefore easy to force the others to leave the place (Cases include: M.v.Gulf Coast - 2010, M.v. UAL transporter - 2012, M.v. Walvis - 2013). From the seven cases in which an undesirable outcome took place in the citadel and, as a result, crewmembers were kidnapped or harmed, only in three cases (42%) did all the crew reach the place. Also, during the period being studied, in 45% of the cases (Table 2), partial confinement of the crew was used along with other self-protection measures. For example, armed teams went on board to repel the attack.

Table 2: Comparing the degree in which crew members could protect themselves: by confining the entire crew in the citadel versus having only some of the crew confined and using other measures adopted by the members who remained outside. Source: The author’s own and IMB in [15], [16], [17], [18], [19].

<table>
<thead>
<tr>
<th>ATTACKS MADE AND CITADEL USED</th>
<th>2010</th>
<th>2011</th>
<th>2012</th>
<th>2013</th>
<th>2014</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>ONLY SOME OF THE STAFF INSIDE; THOSE OUTSIDE IMPLEMENTED OTHER SELF-PROTECTION MEASURES</td>
<td>3</td>
<td>20</td>
<td>15</td>
<td>10</td>
<td>10</td>
<td>58</td>
</tr>
<tr>
<td>PERCENTAGE OF TOTAL</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>45%</td>
</tr>
<tr>
<td>ENTIRE CREW HAD TO TAKE REFUGE</td>
<td>16</td>
<td>34</td>
<td>10</td>
<td>6</td>
<td>4</td>
<td>70</td>
</tr>
<tr>
<td>PERCENTAGE OF TOTAL</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>55%</td>
</tr>
</tbody>
</table>

There is no doubt that, as time passes, pirates can change their tactics to adapt to citadels in an effort to reach the crew and take them hostage. They could use firearms or more powerful explosives and cutting tools found in the vessel itself or they may intentionally start a flood and fire. The citadels would become more vulnerable [9]. If the citadel is properly designed, and the crew scrupulously follows known and tested protocols, its integrity would remain intact until the rescue team arrived. In any case, the data in Tables 1 and 2 show that,
when coupled with other protective measures, using the citadel is an effective step that contributes to the safety of the ship and crew.

Several factors determine if a citadel should be installed. These include the extent to which a ship is vulnerable, the likelihood that pirates will attack and the existence of other protective measures. Therefore, the following point should be taken into account:

- Freeboard height and ship speed. Less than eight meters of freeboard and a low speed increase the likelihood of a ship being attacked by pirates. There are no cases of piracy on ships whose speed exceed 18 knots [9], [20].

- Cruising areas. The use of a citadel is more indicated for vessels that regularly operate from areas with high risk of pirate attack.

- Other measures of protection. Using the citadel has to be coupled with other protective measures to delay the approach long enough for the crew to take refuge. Also necessary are protective devices, like the MSPA (Maritime Security Patrol Areas) with intervention teams that can act in a reasonable time. (Table 3)

**Table 3** Use of the citadel within HRA areas, indicating in brackets the number of times armed forces rescuers on helicopter or ship intervened to release the crew. Source: Author’s own and IMB in [15], [16], [17], [18], [19].

<table>
<thead>
<tr>
<th>CITADEL USED DURING ATTACK</th>
<th>2010</th>
<th>2011</th>
<th>2012</th>
<th>2013</th>
<th>2014</th>
<th>total</th>
<th>%</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOMALIA</td>
<td>5 (2)</td>
<td>39 (13)</td>
<td>10 (3)</td>
<td>2 (2)</td>
<td>4</td>
<td>60</td>
<td>46.9</td>
</tr>
<tr>
<td>GULF OF ADEN/RED SEA</td>
<td>2 (1)</td>
<td>15 (3)</td>
<td>4 (0)</td>
<td>5 (4)</td>
<td>4</td>
<td>30</td>
<td>23.4</td>
</tr>
<tr>
<td>OTHER AREAS OF AFRICA</td>
<td>12 (4)</td>
<td>-</td>
<td>11 (4)</td>
<td>9 (2)</td>
<td>5</td>
<td>37</td>
<td>28.9</td>
</tr>
<tr>
<td>OTHERS</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>1</td>
<td>1</td>
<td>0.8</td>
</tr>
</tbody>
</table>

4. Citadel design.

When designing a citadel, factors have to be taken into account. These include its location or dimensions and the maximum time it is necessary for the crew to be confined. One should also consider other specific requirements related to different elements of the ship, like doors, hatches, bulkheads and decks. A further consideration is related to the materials and equipment needed to service the citadel, including the remote control system platform, power generation, ventilation, communications equipment, water and food.

4.1 Location

Considering the different types of ships, each one with its corresponding distribution of spaces, it is difficult to identify a location that is valid for all types of vessel. However, it is possible to identify some general criteria that should be taken into account as far as possible:

a) The citadel should be difficult for pirates to locate. Access to the citadel should be camouflaged and close to the safe assembly point (muster safe point). It seems reasonable to have a dual path to reach and leave the citadel, that is, access to two different passageways.

b) The citadel should be on a middle deck, if possible within the hull. Superstructure areas should be avoided. Because of the lower sheet thickness used there, it can be easily pierced by small calibre ammunition.
c) The citadel should not be in direct contact with the ship's side or outside decks. Nor should it be vulnerable to an attack with grenade type weapons of greater calibre.

d) As far as possible, one should avoid having any type of openings to the outside or to other easily accessible compartments that are difficult to protect.

e) The citadel should be located close to a supply area and exhaust ducts with independent, forced ventilation. This area has to be difficult to sabotage.

4.2. Length of time confined in the citadel

With the design of the citadel, it should be possible for the crew to be confined for at least the time it takes the rescue team to intervene. Since 2010, in cases in which the entire crew was enclosed, the average time in hiding has been 12.5 hours. The maximum registered time was 19 hours in 2010 (Table 4). If citadel use extends to other areas where the MSPA runs less efficiently [14], p. 3, the design may have to be altered so that the crew can stay for up to 48 hours.

Table 4 Average time period spent inside the citadel in cases where the entire crew was secured. Source: Author’s own and IMB in [15], [16], [17], [18], [19].

<table>
<thead>
<tr>
<th>CASES CITADEL USED WITH ENTIRE CREW SECURED</th>
<th>2010</th>
<th>2011</th>
<th>2012</th>
<th>2013</th>
<th>2014</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AVERAGE TIME SPENT THERE (HOURS)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2010</td>
<td>16</td>
<td>34</td>
<td>10</td>
<td>6</td>
<td>4</td>
<td>70</td>
</tr>
<tr>
<td>2011</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2012</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2013</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2014</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total</td>
<td>19</td>
<td>14</td>
<td>5</td>
<td>SHORT</td>
<td>SHORT</td>
<td>12.5</td>
</tr>
</tbody>
</table>

4.3. Dimensions

The citadel capacity must be capable of accommodating the entire crew [11]. When it comes to building community, safe rooms on onshore facilities, the USA - Federal Emergency Management Agency [21] determines that the area ranges from 6 to 12 m² per person, depending on how long they are to stay there. On the other hand, warships are clearly limited in terms of space. The area for common use is reduced to 5 and 7 m² [22]; ILO Convention 92 [23] concerning accommodation recommends a lever value, 2.78 m² (Table 5).

Therefore, considering the limited space on board and the length of stay required (48 hours), it seems reasonable for the citadel to have an area of about 3 m² per person.

Table 5. Estimating the citadel area required for confined spaces for people in case of inclement weather outside [21] (FEMAP), necessary living space aboard warships [22] (NATO) and minimum space for cabins [23] (ILO). Source: Author’s own

<table>
<thead>
<tr>
<th>Category</th>
<th>FEMAP, [21]</th>
<th>NATO, [22]</th>
<th>ILO Convention 92, [23]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time inside (days)</td>
<td>1-7</td>
<td>&gt; 7</td>
<td>Undefined</td>
</tr>
<tr>
<td></td>
<td>5-7</td>
<td>2.25</td>
<td>2.78</td>
</tr>
<tr>
<td>Area per person (m²)</td>
<td>6</td>
<td>12</td>
<td>5-7</td>
</tr>
</tbody>
</table>
4.4 Specific requirements for the ship

4.4.1 Doors and hatches

Installing metal safety doors and hatches in certain areas will make it more difficult for pirates to gain access to essential equipment and systems. Thus they cannot gain control of the ship in terms of the bridge, motors, generators and steering gear. Nor can they reach existing cutting tools on board and use them in their assault of the citadel [9].

Lashings should be independent, possibly with a common drive system to make manoeuvring easier. The same is true of fittings and hinges along the inside surface. Here the aim is to stop the pirates from taking them apart on the outside. Moreover, the elements are better integrated and camouflaged along the bulkhead [14].

4.4.2 Bulkheads and decks

Bulkheads and decks should be made of welded steel to achieve a degree of gas tightness, and then they should undergo pneumatic testing.

4.4.3 Ballistic protection

The ballistic protection of certain elements on the ship is considered crucial in citadel design; it directly affects the safety of the area against armed attack. Thus, installing suitable steel doors, bulkheads and decks or reinforcing existing ones will provide appropriate ballistic protection (Table 6).

<table>
<thead>
<tr>
<th>Ballistic Resistance</th>
<th>Blast Resistance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bulkhead and decks</td>
<td></td>
</tr>
<tr>
<td>Opaque zone</td>
<td>Glazing</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Ballistic Resistance</th>
<th>Blast Resistance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Doors and frames</td>
<td></td>
</tr>
<tr>
<td>Opaque zone</td>
<td>Glazing</td>
</tr>
<tr>
<td>Impacts Calibre 7.62 x 51 a 10 m.</td>
<td>BR6 or BR7</td>
</tr>
</tbody>
</table>

* Equivalent to an AK 47 rifle type
4.4.4 Fire extinguisher Systems

Given that pirates could start a fire to make, the crew evacuate the citadel, local systems should have active and passive fire prevention. An example of this is A60 type fire retardant insulation in [28], Cap.II.2, part A, rule 3, in bulkheads and decks, as well as along the inside of doors. The fixed fire-fighting system has to be water mist. When faced with a fire recently started inside the premises, it should be enough to have one or two fire extinguishers, preferably water ones.

On the other hand, two other ideas would be useful for the remote control system of the platform. It should receive information from the fire detection system [14]. Moreover, there should be remote control for local extinguishing systems, such as in the engine room or cabins (Fig. 3).

4.5 Equipment

4.5.1 Platform remote control system

Pirate will have difficulty gaining control of the ship if equipment and systems can be operated remotely from the citadel. Another useful device would be a remote locking system for doors and hatches providing access to this equipment. The citadel would control this equipment as long as possible. From the platform, this would include essential ship systems, such as steering, propulsion and power generation, were controlled from the citadel. Crew would also be able to switch off bridge consoles and camera controls. Figure 2 shows the arrangement for the control of these systems. Similarly it would be desirable to receive information from and control of the fire detection system, as well as the ventilation.

4.5.2 Autonomous power generation system

Ventilation and fire control are crucial to the crew’s survival in the citadel. Communication equipment must also function. The corresponding electrical balance will help calculate the power, which determines the characteristics of an enclosed autonomous generator system. For reliability and adaptability to the environment, the most suitable system is a generator with electric starter and battery power. Should the generator fail, this battery would feed emergency lighting. The exhaust would be located outside and the main way of cooling the engine would be with air.

4.5.3 Ventilation

On occasion, pirates have used various methods to force the crew to leave the site. One of them is to start a fire outside so that smoke enters the premises through the vents [14]. Therefore, vents are the main point of weakness within the citadel.

CONSIDERING ANTI-PIRACY SHIP SECURITY: CITADEL DESIGN AND USE

**Fig. 2** Control of propulsion and steering of the ship from the citadel, with the possibility of disabling the machine control console and navigation console so that crew can take control of the propulsion and power generation plant, as well as steering, from the secure space. Source: Author’s own

To prevent smoke from making its way in, the room’s ventilation should be autonomous, working independently from the ship’s ventilation systems and general electricity generation. It is advisable to have two selectable points of air admission that are sufficiently spaced. Both the exterior intake and exhaust grille openings should be easily accessible, and not easily recognizable.

**Fig. 3** Functional scheme of autonomous ventilation for the citadel, according to strict design specifications: Double capacity of selectable intake and discharge by a centrifugal axial-type fan, with in-house heating, air filtration and distribution. There is an axial extractor fan with discharge to the outside and possibility of air recirculation with a minimum replacement rate over a limited time. Source: Author’s own
The system should be capable of renewing the room’s air every six min., re-circulating and cooling it for short periods of time. This factor may be the way to counter any attempt made by the assailants to introduce smoke through the main air intake. The air is re-circulated, so that a minimum of air is drawn from the outside and comes from a different weather inlet. The flowchart in Fig. 3 charts this process; its technical specifications are contained in Table 7, while the ventilation control is represented in the flowchart for Fig. 4.

**Table 7** Calculation of ventilation needed in siege conditions. Source [29]

<table>
<thead>
<tr>
<th>Volume</th>
<th>10 renovations/hour</th>
</tr>
</thead>
<tbody>
<tr>
<td>Discharge</td>
<td>mechanical</td>
</tr>
<tr>
<td>Extraction</td>
<td>mechanical</td>
</tr>
<tr>
<td>Design temperature</td>
<td>18 – 26.7 ºC</td>
</tr>
<tr>
<td>Relative humidity</td>
<td>55%</td>
</tr>
<tr>
<td>Minimum renewal for recirculation</td>
<td>8.5 – 17 m³/hour-crewmember</td>
</tr>
</tbody>
</table>

**Fig. 4 Ship Control Platform.** The ventilation and firefighting systems for the citadel operate by means of an autonomous generator. At the same time, information from the fire detection system is received at the citadel console and the fixed firefighting systems can be activated. Source: Author’s own
4.5.4 Communications Equipment

It is vital to have a link with the military units that will be involved in the rescue of the ship. These units need to be informed of the crew’s situation in general and confirm that all of its members have been secured. In the absence of this information, the rescue team will refuse to intervene.

The ICS [14] recommends an autonomous communication system, with a discrete wire antenna, to stop the pirates from doing damage. This equipment should run autonomously for at least three days, relying on the autonomous power system from the citadel. (Fig. 5).

4.5.5 Closed circuit television (CCTV)

A television circuit should be installed on board. Ideally it would be possible to monitor the cameras from the citadel. This system would therefore allow the confined crew to know the movements of the assailants and to inform military units about the situation while the intruders are on board.

4.5.6 General equipment: Food and water, first aid, housing and documentation

Food and the water supplies should reflect the number of crew members and be sufficient for the maximum stay of two days mentioned in paragraph 4.3.

It is also necessary to calculate the space needed for collecting the waste that is generated, as well as for chemical toilets. Again, crew size has to be considered, with one toilet for every ten members (Table 8).

A reinforced standard first aid kit should be enough to meet requirements and provide initial treatment of gunshot wounds or burns. It is important that the kit also includes specific medicines usually required by some crew, such as insulin or allergy relief.

As for documentation, the citadel should include a nominal list of the crew to verify those who are present. There should also be an updated list of contacts to be maintained with control agencies. Other items are recommended to make confinement as comfortable as possible without unduly taking up space. These include sleeping bags, mats or mattresses and torches.
**Fig. 5** Communications and navigation systems built into the console of the citadel, receiving GPS and CCTV signals and using a wire communications antenna. Source: Author’s own

**Table 8** Number of crew per toilet [22], [23] recommended number of toilets. Source: Author’s own

<table>
<thead>
<tr>
<th>Category</th>
<th><strong>NATO [22]</strong></th>
<th><strong>ILO [22]</strong></th>
<th>Recommendation for citadel</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Toilets</td>
<td>Urinals</td>
<td>Toilets</td>
</tr>
<tr>
<td>Officers</td>
<td>8-15 persons per toilet</td>
<td>15-30 persons per urinal</td>
<td>8</td>
</tr>
<tr>
<td>Intermediate staff</td>
<td>10-15</td>
<td>20-30</td>
<td></td>
</tr>
<tr>
<td>Sailors</td>
<td>13-15</td>
<td>20-30</td>
<td></td>
</tr>
</tbody>
</table>

CONSIDERING ANTI-PIRACY SHIP SECURITY: CITADEL DESIGN AND USE

Table 9 Recommendations for the location, length of time in confinement, dimensions, specific elements and equipment in the citadel. Source: Author’s own

<table>
<thead>
<tr>
<th>LOCATION AND DESIGN OF THE CITADEL - RECOMMENDATIONS</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>LOCATION</strong></td>
</tr>
<tr>
<td>Deck</td>
</tr>
<tr>
<td>Separation at the sides</td>
</tr>
<tr>
<td>Intermediate</td>
</tr>
<tr>
<td>&gt; 2 metres</td>
</tr>
<tr>
<td><strong>LENGTH OF TIME IN CONFINEMENT</strong></td>
</tr>
<tr>
<td>48 hours</td>
</tr>
<tr>
<td><strong>SURFACE</strong></td>
</tr>
<tr>
<td>3 m² per member of the crew</td>
</tr>
</tbody>
</table>

**SPECIFIC ELEMENTS**

<table>
<thead>
<tr>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Firefighting system</td>
</tr>
<tr>
<td>- Water mist</td>
</tr>
<tr>
<td>- 1 / 2 water extinguishers</td>
</tr>
</tbody>
</table>

**EQUIPMENT**

| Remote control system of the platform                                                                        |
| - Steering                                                                                                  |
| - Propulsion                                                                                                 |
| - Power generation                                                                                           |
| - Navigation                                                                                                 |
| - CCTV                                                                                                       |
| Power generator                                                                                                |
| - Battery start-up                                                                                            |
| - Air -cooled                                                                                                 |
| Communications equipment                                                                                      |
| - Discrete wire antenna                                                                                       |
| Ventilation system                                                                                            |
| - Inlet and exhaust                                                                                            |
| - Two separate points of admission                                                                           |
| - Discrete and inaccessible exterior grilles                                                                 |
| -> 8.5 m³ per hour and member of the crew                                                                    |
| - Renewal every 6 min.                                                                                         |
| - Recycling and renovation for short periods of time                                                          |
| Food, water and first aid kit                                                                                 |
| - For 3 days                                                                                                  |
| Chemical toilets                                                                                              |
| - 1 toilet per every 10 members of the crew                                                                   |

5. Conclusions.

With citadels, the main objective is to prevent pirates from taking hostages. They have proved to be an effective self-protection measure, contributing to the safety of the ship and crew. This is especially the case when they are linked to other protection measures, such as deploying military forces as rescue teams. In 2010-2014 the failure rate of the citadel was 5.5 %, or 7 out of 128 cases.

As for the citadel’s design, various factors come into play, including location, size, maximum confinement time, the specific requirements for different elements of the ship and the equipment and materials needed to service the citadel (Table 9).

Ballistic protection and the existence of certain independent and autonomous systems are critical aspects of the citadel’s design. They directly affect the room’s integrity and the crew’s resistance to an armed attack. Among these systems, power generation equipment stands out. This equipment makes it possible to feed, among other items, the communication equipment needed to link with the military authorities responsible for the rescue operation. It also runs the ventilation. This crucial system must have two selectable points of admission with discrete and inaccessible grids, with the option of re-circulating the air for short periods of time.
REFERENCES


CONSIDERING ANTI-PIRACY SHIP SECURITY: CITADEL DESIGN AND USE


Submitted: 18.03.2015.
Accepted: 07.08.2015.

Luis Carral, lcarral@udc.es, University of A Coruña
Carlos Fernández-Garrido, cfgarrido@fn.mde.es, Escuela “Antonio de Escaño” – Spanish Navy
José J. de Troya, troya@udc.es, University of A Coruña
José Ángel Fraguela, jafraguela@udc.es, University of A Coruña
AN ENHANCED EQUATION FOR VIBRATION PREDICTION OF NEW TYPES OF SHIPS

UDC 629.5.01: 629.5.035: 629.5.015.5
Original scientific paper

Summary

A simplified approach developed to evaluate the vibration levels of complex structures such as passenger and similar ships with large shell and deck openings and extended superstructures is here presented. The final objective is to give an useful tool to ship designers, to establish since the first design stage the dynamic response of the ship with sufficient precision.

This approach is based on the assumption that the ship hull can be represented as a non uniform section beam. The propeller excitations in terms of pressure pulses and shaft line moments and forces are introduced. To take into account this exciting source in the early design stage a statistical formula for dynamic excitation of propeller was developed. Furthermore the superimposition of local effects has been performed with the use of an analytical formula. The local effect due to the different space topologies such as cabins, public spaces, technical and machinery areas has been taken into account. The transversal beams, longitudinal girders, stiffeners and pillars as supported structural elements are considered in the vibration local response.

The reliability of the results obtained using the formula has been improved with more precise results obtained by FEM analysis. The calculated vibration response has been verified and compared to vibration measurements performed on board of ships.

Key words: ship vibrations; non-uniform section beam; correction factor; local magnification factor; vibration measurements; error index

1. Introduction

At the early beginning Kumai developed simple analytical formulas for the calculation of natural frequencies of merchant ships [1]. Modern Pax and Ro-Pax vessels have large shell and deck openings and extended superstructures unlike older merchant ships. The FE method was developed for the design of ships and at the same time classification societies developed FE analysis procedures for static and dynamic investigation [2-9]. Thus these formulas nowadays are not useful for passenger and Ro-Ro passenger ships. In addition the owners and the Classification societies reduced vibration limits defined into the Comfort Class requirements [10-15]. At the early design stage there is not enough time and technical data to develop a FEM
Model. On the other side eventual errors in dynamic design introduce high costs on the total construction [16]. Due to this it has been decided to develop a new formula for ship global and local vibration response prediction.

The objective of this investigation is to present and prepare the data for the concept design phase where the most significant decisions are made regarding the ship dynamic design. It is important to take the most important decisions in early design stages because the earlier a problem is addressed, the easier it is to implement changes, and less costly it is to resolve. On the contrary, if most of the changes happen in late design stages, the cost of making changes will dramatically increase since design freedom is highly limited in these stages [16].

![Figure 1-1 Cost Influence](image)

Furthermore this problem is complex and time consuming, this is in contrast with low available time for the basic–concept structural design. This means that huge amount of data required in the 3D FEM models cannot be generated and are not available. Concept design is the phase in structural design when geometry typology and shapes are subject to modification and structural options are investigated in accordance to layout and design [17,18].

The vibration problem is significant for multi-deck ships, passenger cruise ships, RO-RO pax ships, mega yachts with the extended, multi-deck superstructures. Definition of the adequate scantling is also very important for the dynamic design that is correlated to structural weight, achievable clearances regarding heights of the webs; transversal beams longitudinal girders distribution and pillars layout.

A simplified approach to dynamic prediction design is the basic objective of the method here presented, to give acceptable results during the basic design phase. The simplified approach has been developed for evaluating the vibration levels of complex structures such as passenger and similar ships with large shell and deck openings, and large primary structure. The final objective is to give an useful tool to ship designers, to establish in the first design stage the dynamic ship response with sufficient precision.

2. **Generalized formulation for a non-uniform beam**

It is known that the ship section vary with length. Different authors solved the problem for cargo ships with direct partial differential equation solution [19-21]. The idea to use in any way the uniform section beam theory and expand it for a non-uniform hull ship section has been here developed. The ship was divided in ten sections along her length in accordance with recommended practice from Naval Literature [1]. A regression of the dimensionless area, inertia
An Enhanced Equation for Vibration Prediction of New Types of Ships

Valter CERGOL, Peter VIDMAR

data, mass and added mass of various ships has been carried out. The cross-section area and inertia data used for the regression refer to transversal sections of ships whose sectional properties had been calculated. Mass distributions are taken from the author’s database.

The solution of a two stepped beam can be viewed as expansion of the uniform beam as separate beams with continuity conditions applied at the joints. The basic idea developed by Koplov [22,23] for two-stepped beams has been expanded to ten-stepped beams, that has been used for a global ship dynamic response. The solution is found in terms of dynamic force and coupled excitation. The calculated results have been compared to author’s experimental data obtained during sea trials.

In general for the \( i \)th section of a non-uniform beam of \( n \) sections the solution can be obtained from the simpler case of a beam with two different sections that can be written in the following form:

\[
V_i(x_i) = c_{1i}\sin\beta_i x_i + c_{2i}\cos\beta_i x_i + c_{3i}\sinh\beta_i x_i + c_{4i}\cosh\beta_i x_i
\]

\( \forall \ i = 1, \ldots, n \) (1)

where \( V_i(x_i) \) is the general mode shape solution, \( c_{1i}, c_{2i}, c_{3i}, c_{4i} \) are constants determined by suitable boundary conditions and where:

\[
\beta_i^4 = \frac{\omega^2\rho_i A_i}{E_i I_i(1 + i\eta)}
\]

The boundary condition, considering the dynamic excitation \( F \) pulsing-oscillating at frequency \( \omega \) and applied at the free end of the beam belonging to the 1st section, is:

\[
\frac{\partial^2 v_1(0)}{\partial x^2_1} = 0
\]

\( E_i l_1 \frac{\partial^3 v_1(0)}{\partial x^3_1} = -F \sin \omega t \) (4)

The continuity conditions between two adjacent sections of the beam \( i \) and \( i+1 \) are:

\[
v_i(L_i) = v_{i+1}(0)
\]

\[
\frac{dv_i(L_i)}{dx_i} = \frac{dv_{i+1}(0)}{dx_{i+1}}
\]

\[
E_i l_i \frac{d^2 v_i(L_i)}{dx^2_i} = E_{i+1} l_{i+1} \frac{d^2 v_{i+1}(0)}{dx^2_{i+1}}
\]

\[
E_i l_i \frac{d^3 v_i(L_i)}{dx^3_i} = E_{i+1} l_{i+1} \frac{d^3 v_{i+1}(0)}{dx^3_{i+1}}
\]

The free boundary condition at the free beam end belonging to the last \( n \)th section is:

\[
\frac{\partial^2 v_n(L_n)}{\partial x^2_n} = 0
\]

93
An Enhanced Equation for Vibration Prediction of New Types of Ships

\[ E_n I_n \frac{\partial^3 v_n(L_n)}{\partial x_n^3} = 0 \]  \hspace{1cm} (10)

With reference to the previous relationships it has been decided to search the solution for a linear algebraic system of equations. This leads to a solvable linear algebraic system composed of:

- \( n \times 4 \) unknowns \( c_{ki} ; k=1,2,3,4; i=1,…,n \)
- \( (n-1) \times 4 \) continuity equations
- \( 2+2=4 \) free-free boundary conditions

Further steps in this approach are based on the fundamental procedure just explained. Based on these formulae the ship structure sections were discretised in 10 parts, each with its sectional properties. This way the ship’s global dynamic frequency response is obtained.

Similarly, the procedure for the generalized equation for moment has been developed. The free boundary condition at the free beam belonging to the \( i \)th section is:

\[ \frac{\partial^3 v_i(0)}{\partial x_i^3} = 0 \]  \hspace{1cm} (11)

\[ E_1 I_1 \frac{\partial^2 v_i(0)}{\partial x_i^2} = M \sin \omega t \]  \hspace{1cm} (12)

The continuity conditions between two adjacent sections of the beam \( i \) and \( i+1 \) are:

\[ v_i(L_i) = v_{i+1}(0) \]  \hspace{1cm} (13)

\[ \frac{\partial v_i(L_i)}{\partial x_i} = \frac{\partial v_{i+1}(0)}{\partial x_{i+1}} \]  \hspace{1cm} (14)

\[ E_i I_i \frac{\partial^3 v_i(L_i)}{\partial x_i^3} = E_{i+1} I_{i+1} \frac{\partial^3 v_{i+1}(0)}{\partial x_{i+1}^3} \]  \hspace{1cm} (15)

\[ E_i I_i \frac{\partial^2 v_i(L_i)}{\partial x_i^2} = E_{i+1} I_{i+1} \frac{\partial^2 v_{i+1}(0)}{\partial x_{i+1}^2} \]  \hspace{1cm} (16)

The free boundary condition at the free beam end belonging to the \( n \)th section \( i \).

\[ \frac{\partial^2 v_n(L_n)}{\partial x_n^2} = \frac{\partial^3 v_n(L_n)}{\partial x_n^3} = 0 \]  \hspace{1cm} (17)

3. Cross Section Area Inertia and Mass Distribution

The new developed formula is based on available data of ships investigated by the author. For Ro/Ro Pax, Pax passenger ships the significant section and mass distribution data are mathematically calculated. The cross-section area, inertia and mass distribution are used in the next chapters as parameterized input for the new formula. More precisely \( I(x) \), \( \rho(x) \) and \( A(x) \) are represented by fourth-order polynomial regression for geometrical properties and fifth-order for cross section properties. In general the section area, inertia and mass can be written in the following polynomial expressions [24]:

**Section Area**

\[ A(x) = \sum_{k=0}^{4} a_k x^k \]  \hspace{1cm} (18)
Section Inertia

\[ I(x) = \sum_{k=0}^{4} i_k x^k \]  

(19)

Section Mass

\[ \rho_m(x) = \sum_{k=0}^{5} r_k x^k \]  

(20)

3.1 Midship Section Characteristics for Analysed Ship

During the last twenty years the author performed dynamic analysis of several passenger and RO-RO passenger ships. The hull section characteristics have been calculated based on midship section drawings where scantlings are represented. These data are used for the mathematical processing. Table 3-1 presents the calculated midship section data for each ship analysed. The drawings are shown in Figure 3-1.

**Table 3-1 Midship general data**

<table>
<thead>
<tr>
<th>Significant Ships</th>
<th>SECTION</th>
<th>L [m]</th>
<th>B [m]</th>
<th>T [m]</th>
<th>(\Lambda) [t]</th>
<th>(L) [m³]</th>
<th>(A) [m²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship 1</td>
<td>Midship</td>
<td>254.8</td>
<td>30.8</td>
<td>7.3</td>
<td>29741</td>
<td>421.57</td>
<td>4.37</td>
</tr>
<tr>
<td>Ship 2</td>
<td>Midship</td>
<td>185.1</td>
<td>24.7</td>
<td>6.7</td>
<td>16480</td>
<td>219.41</td>
<td>3.61</td>
</tr>
<tr>
<td>Ship 3</td>
<td>Midship</td>
<td>280.8</td>
<td>32.2</td>
<td>7.8</td>
<td>43423</td>
<td>486.20</td>
<td>5.71</td>
</tr>
<tr>
<td>Ship 4</td>
<td>Midship</td>
<td>203.0</td>
<td>30.4</td>
<td>7.8</td>
<td>27650</td>
<td>472.85</td>
<td>4.67</td>
</tr>
</tbody>
</table>

**Figure 3-1** In order from left to right: Ship 1, Ship 2, Ship 3 and Ship 4 midship section

4. Calculation of Hull Cross Sectional data and Weight Distribution

It has been decided, in agreement to Naval Architecture good practice, to divide the ship hull length in ten sections. Not only in midship, but also aft and fore section, areas, inertias and mass distribution have been evaluated. The fifth order polynomial data regression is used in full load conditions based on the weight distribution data of three ships at different sections. The result can be seen on Figure 4-1.
An Enhanced Equation for Vibration Prediction of New Types of Ships

Weight distribution regression

\[ W[\] = \frac{W_L}{\Delta/L} \left( \frac{t}{m} \cdot \frac{1}{t} \cdot \frac{m}{t} \right) \]  
\[ (21) \]

where \( W_L \) is the total weight per unit length.

The polynomial regression is:

\[ y = 30.07 x^5 - 86.96 x^4 + 91.77 x^3 - 45.53 x^2 + 10.54 x + 0.33 \]  
\[ (22) \]

It has been decided to parameterize the cross section area and the moment of inertia to midship section value. The area has been represented as non-dimensional value through the following formula:

\[ A' = \frac{A(x)}{A_M} \]  
\[ (23) \]

with \( A_M \) is the cross section area of the midship section.

The dimensionless moment of inertia \( I_z' \) has been calculated with the following formula.

\[ I_z' = \frac{I_z(x)}{I_{zM}} \]  
\[ (24) \]

where \( I_{zM} \) is the moment of inertia of the midship section.

In Figure 4-2 and in Figure 4-3 the polynomial regression of the obtained data for five ships at different sections, about cross section Area \( A' \) and about moment of inertia \( I_z' \).

In particular the polynomial regression formula are:

\[ A_i' = 2.68 x^4 - 4.86 x^3 + 1.69 x^2 + 0.45 x + 0.78 \]  
\[ (25) \]

\[ I_{zi}' = -8.31 x^4 + 13.96 x^3 - 8.77 x^2 + 2.30 x + 0.79 \]  
\[ (26) \]
4.1 Added Mass and Vertical Motion

For a circular cylinder in two-dimensional flow, inviscid fluid theory shows, that the added mass equals the mass of the cylinder if the cylinder has the density of the fluid. For a sphere it is one half the mass of the sphere if the sphere has the density of the fluid [25,26].

A vibrating ship acts as if the added mass, varying along its length, were attached. The magnitude of this added mass is a function of the ship underwater shape and of the mode of the vibration. The added mass per unit length is then given by:

$$A.M. = 0.5\rho \pi b_L^2 C_V$$

where:
- $\rho$ = density of the fluid
- $C_V$ = added weight vertical coefficient
- $J$ = coefficient for 3-dimensional effects
- $b_L$ = half-beam at the waterline for the section being considered

For the ships under investigation the amount of added mass has been calculated and parametrized. The regression analysis used in this case is the third order polynomial.

Similarly to what has been carried out for the cross section characteristics and weight distribution on board, also the added mass has been calculated on the ship length. The following formula has been developed for the added mass parameterized to displacement and length:

$$A.M. \text{[dimensional]} = A.M. \left( \frac{L}{\Delta \cdot 1000} \right) \left[ \frac{kg}{m \cdot t \cdot m} \right]$$

Figure 4-4 shows the added mass for eight ships as a function of the ratio $x/L$ for the combined and local effect mode chosen.

The polynomial regression is:

$$y = 0.41 x^3 - 1.85 x^2 + 1.06 x + 1.25$$

5. Dynamic Excitation on ship

In order to perform the dynamic vibration response calculation, the propeller has been used as the dynamic excitation as it is the main source of vibrations induced to the ship. The excitations induced by the propeller are mainly divided those into the line shafting and those
into the hull [27-30]. The referenced values here used have been calculated by author or obtained from model base in tests.

Diesel engine as primary source and/or Diesel generators is considered as the second source of excitations, which might be at the origin of vibrations appearing on board ship. In this paper this source will not be numerically investigated, but the excitation data are normally supplied by engine or manufacturer [31].

5.1 Propeller excitation calculation based on Regression

The non-uniform beam used to mathematically represent the ship has been excited with a dynamic force due to the propeller pressure fluctuation and shaft-line induced dynamic bending moment.

For defining the propeller excitation there are several significant parameters that can be considered. The power $P$ [kW], the cruise speed of the ship $V$ [kn] and displacement $\Delta$ [tonnes] have been selected. For the analysed ships the equivalent propeller excitation has been calculated based on data available to the author.

<table>
<thead>
<tr>
<th>Ships</th>
<th>$\Delta$ [t]</th>
<th>Power [kW]</th>
<th>Speed [kn]</th>
<th>LOA [m]</th>
<th>Fn [/]</th>
<th>P/\Delta</th>
<th>F [kN]</th>
<th>M [kNm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship 1</td>
<td>29741</td>
<td>21120</td>
<td>20</td>
<td>265.4</td>
<td>0.20</td>
<td>0.71</td>
<td>13.8</td>
<td>50.0</td>
</tr>
<tr>
<td>Ship 2</td>
<td>16480</td>
<td>17300</td>
<td>21</td>
<td>192.8</td>
<td>0.25</td>
<td>1.05</td>
<td>14.0</td>
<td>48.0</td>
</tr>
<tr>
<td>Ship 3</td>
<td>43423</td>
<td>67370</td>
<td>24</td>
<td>292.6</td>
<td>0.23</td>
<td>1.55</td>
<td>23.8</td>
<td>55.0</td>
</tr>
<tr>
<td>Ship 4</td>
<td>27650</td>
<td>67200</td>
<td>28</td>
<td>211.5</td>
<td>0.31</td>
<td>2.43</td>
<td>41.0</td>
<td>60.0</td>
</tr>
</tbody>
</table>

A regression analysis of the propeller excitation has been performed for the total vertical integrated force and shaft-line dynamic bending moment:

$$ F = 2148 + 17795 \frac{P}{\Delta} - 17755 Fn \quad [N] $$

$$ M = 34940 - 1382 \frac{P}{\Delta} + 81175 Fn \quad [Nm] $$

where $P$ is the power, $\Delta$ the displacement, $Fn$ the Froude number.

This way the equivalent vertical total integrated force and shaft-line bending moment have been obtained.

6. New Formula mathematical development

The new formula, based on the previously obtained data, is developed in several steps, where the total vertical vibrations response will be obtained as the product of the global, local deck’s amplification and deck’s position. The ship length $L_R$ has been divided in ten sections, $i=1,...,10$. In the $i$-th ship section this form is valid:

$$ (V_{T,i},v_{z,j},v_{y,k}) = V(x)_i \cdot v_{z,j} \cdot v_{y,k} \quad i = 1,...,10 \quad k = 1,..,n $$

where $n$ is the last deck and:

- $v_{y,k}$ is the $k$-th deck’s local amplification factor at $i$-section
An Enhanced Equation for Vibration Prediction of New Types of Ships

Valter CERGOL, Peter VIDMAR

- $v_{z_{k,i}}$ is the variation of the vibration’s with deck height
- $V(x)_i$ is the global vibration level of hull girder [mm/s]
- $(V_{x,y,x})_{k,i}$ is the total vertical vibration level [mm/s]

Here below the expression of each term of the obtained formula:

- $v_{y_{k,i}}$, the deck’s local amplification factor
  
  $$v_{y_{k,i}} = \max \delta_{s_k}(y) \frac{\omega^2}{g} A_R + 1 \quad k = 1, \ldots, n \quad \forall \ i$$

  where $y \in [0; \ l/2]$, and $l$ is the beam span;

- $v_{z_{k,i}}$, the variation of the vibration’s with deck height.
  
  $$v_{z_{k,i}} = a_i z_k^6 + b_i z_k^5 + c_i z_k^4 + d_i z_k^3 + e_i z_k^2 + f_i z_k + g_i$$

  $\forall \ i \quad k = 1, \ldots, n$

  $$z_k = \frac{z_{dk}}{z_b}$$

  where $z_{dk}$ is the k-th deck height and $z_b$ is the bottom height;

- $V(x) = V_G(x) \cdot R(x)$ is the global vibration level of hull girder [mm/s]
  
  $$V_G(x)_i = \max_j \left[ a_{1+4i} \sin \left( \beta_{i+1} j \frac{L_R}{m \cdot n} \right) + a_{2+4i} \cos \left( \beta_{i+1} j \frac{L_R}{m \cdot n} \right) + 
  + a_{3+4i} \sinh \left( \beta_{i+1} j \frac{L_R}{m \cdot n} \right) + a_{4+4i} \cosh \left( \beta_{i+1} j \frac{L_R}{m \cdot n} \right) \right]$$

  $\forall i = 1, \ldots, m$

  The ship length $L_R$ has been divided in ten sections, $i=1,\ldots,10$. Furthermore it has been decided to split each of the $L_{10}$ section in ten parts, $j=1,\ldots,10$. $R(x)_i$ is a correction factor, parameterised to Rule length:

  $$R(x)_i = -31.56 \ x_i^4 + 61.79 \ x_i^3 - 36.57 \ x_i^2 + 4.95 \ x_i + 1.66$$

  where $x$ is the dimensionless length

  The vector $a$, in the expression of $V_G(x)_i$, is the result of the product between matrix $G$ and the dynamic excitation vector $f$:

  $$a = G^{-1} f$$

  where $G$ is the generalized matrix and $f$ the excitation vector. In particular:
The hull girder global vibration has to be calculated by applying a corrector factor \( R(x) \) to the obtained analytical response levels. The corrector factor was derived from the ratio between analytical and results obtained in complete FEM models (Figure 6-1).

6.1 Global Vibration Level Calibration

The calculation of the global vibration level of hull girder \( V_G \) is now described. In particular the vibration displacement has been converted by multiplying by \( 2\pi f \) to velocity. The hull girder global vibration has to be calculated by applying a corrector factor \( R(x) \) to the obtained analytical response levels. The corrector factor \( R(x) \) was derived from the ratio between analytical and results obtained in complete FEM models (Figure 6-1).
Due to the fact that the ships have been analytically discretized in ten parts, there are ten correction coefficients, one for each ship section. In Figure 6-2, Figure 6-3, Figure 6-4 and Figure 6-5 we show the FEM and corresponding analytical results.

In Figure 6-6 the ratio between the analytical and the FEM results, i.e. the coefficient $R(x)$, is presented for each of the ten sections:
An Enhanced Equation for Vibration Prediction of New Types of Ships

\[ R(x)_i \] is the polynomial regression of the correction factor:

\[
R(x)_i = -31.56x^4 + 61.79x^3 - 36.57x^2 + 4.95x + 1.66
\]  
(39)

6.2 Local Amplification Dynamic Effects of Deck Structures

It is physically reasonable that local deck effects are very significant in case the natural frequencies of deck structures are close to propellers or engines excitation frequencies. The analytical formula for calculating local amplification effects of the deck's structures will be described in this paragraph. The local amplification factors \( V(y) \) have been calculated based on:
- natural frequencies of deck supporting structural members;
- static deflection of deck’s depending on deck weights in accordance with general arrangement;
- ratio between the natural and exciting frequency;

The following data are required for calculate local amplification \( V(y) \) factor calculation:
- deck structural arrangement considering spans, deck plating, deck longitudinal, deck girders and deck transverses;
- deck general arrangement in order to establish the local deck load and consequently local static deflection.

With this information in hand it is possible to calculate deck deflections and natural frequencies. Due to the fact that the ship has been discretized in ten parts, the magnification factor has to be calculated for all ship decks and for all ten ship’s discretized sections. Deck structure natural frequencies have also been calculated to consider the coupled influence of deck transverses and deck longitudinal stiffeners.

The adopted beam natural frequency formula is:

\[
\omega_n = c_G \cdot 5.12 \cdot 10^2 \cdot \sqrt[3]{I A_B} \frac{1}{l^2}
\]  
(40)

where
- \( c_G \) constant that depend on type of boundary condition;
- \( I \) beam inertia \([m^4]\);
- \( A_B \) beam area \([m^3]\);
- \( l \) beam span \([m]\).
The ratio between natural frequency of the deck transverse and natural frequency of deck stiffeners is equal to $\beta$. Considering the transversal beam and stiffeners as two springs supporting the same mass, the total coupled natural frequency is then calculated by the use of the formula:

$$f = \sqrt{f_{\text{transversal beam}}^2 + f_{\text{stiffener}}^2} = f_{\text{stiffener}} \sqrt{1 + \beta^2}$$  \hspace{1cm} (41)

The static deflection $\delta_s$ has been calculated with the Grashoff method.

$$\delta_s = \frac{5q_lL^4}{384EI}$$  \hspace{1cm} (42)

where $q_l$ is the distribution of the load $q$ into two loads, one for each length of the panel.

Once the natural frequency and static deflection have been calculated it is possible to obtain the local magnification factor as $V(y)$:

$$v_{y,k,i} = \max \delta_{s_k}(y) \frac{\omega^2}{g} A_R + 1 \quad k = 1, \ldots, n \quad \forall \ i$$  \hspace{1cm} (43)

where:
- $\delta_{s_k}(y)$: static deflection of deck area [mm];
- $g$: gravity constant [mm/s$^2$];
- $\omega$: angular velocity in terms of blade pulsing frequency;
- $A_R$: resonance factor for a one-mass system defined below

$$A_R = \frac{1}{\sqrt{(1 - r^2)^2 + \left(2r \frac{\zeta}{\omega_r}ight)^2}}$$  \hspace{1cm} (44)

$r = \text{ratio between the excitation frequency and the natural frequency};$

$\zeta = C/C_r = \text{critical damping ratio}.$

6.3 Vibration variation with Deck Height

The vibration variation with deck height will be now investigated. As can be seen from the FEM results [32,33], deck vibration levels usually globally decrease from bottom, where the main dynamic sources occur, up to the upper decks. Due to the large super extension and similarity the vertical distribution of vibrations for this type of ship is very similar. Due to this property the regression has been performed on the most significant one.

Figure 6-7 - Hull girder side shell and superstructure longitudinal bulkhead vibration velocities

For the analyzed sections, the polynomial regression has been used:
\[ v_{z_{k,i}} = a_i z_k^6 + b_i z_k^5 + c_i z_k^4 + d_i z_k^3 + e_i z_k^2 + f_i z_k + g_i \]
\[ \forall i \; k = 1, \ldots, n \]

with "i" indicating the section in which the level of vibration is being considered.

In order to calculate the regression formula for each of the ten discretized section, global dynamic velocity response has been obtained from global FEM model [34] considering different ship sections and different inter deck areas, deck heights. The values reported in Table 6-1 have been taken in way of side shell area.

<table>
<thead>
<tr>
<th>X/L</th>
<th>0.0-0.1</th>
<th>0.1-0.2</th>
<th>0.2-0.3</th>
<th>0.3-0.4</th>
<th>0.4-0.5</th>
<th>0.5-0.6</th>
<th>0.6-0.7</th>
<th>0.7-0.8</th>
<th>0.8-0.9</th>
<th>0.9-1.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bottom</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>Deck 1</td>
<td>0.48</td>
<td>0.38</td>
<td>0.37</td>
<td>0.16</td>
<td>0.22</td>
<td>0.41</td>
<td>0.25</td>
<td>0.40</td>
<td>0.75</td>
<td>0.70</td>
</tr>
<tr>
<td>Deck 2</td>
<td>0.47</td>
<td>0.39</td>
<td>0.36</td>
<td>0.09</td>
<td>0.22</td>
<td>0.43</td>
<td>0.24</td>
<td>0.25</td>
<td>0.75</td>
<td>0.70</td>
</tr>
<tr>
<td>Deck 3</td>
<td>0.47</td>
<td>0.39</td>
<td>0.35</td>
<td>0.18</td>
<td>0.22</td>
<td>0.38</td>
<td>0.23</td>
<td>0.22</td>
<td>0.69</td>
<td>0.65</td>
</tr>
<tr>
<td>Deck 4</td>
<td>0.47</td>
<td>0.40</td>
<td>0.35</td>
<td>0.08</td>
<td>0.22</td>
<td>0.32</td>
<td>0.22</td>
<td>0.21</td>
<td>0.69</td>
<td>0.65</td>
</tr>
<tr>
<td>Deck 5</td>
<td>0.58</td>
<td>0.41</td>
<td>0.33</td>
<td>0.08</td>
<td>0.21</td>
<td>0.32</td>
<td>0.21</td>
<td>0.21</td>
<td>0.81</td>
<td>0.75</td>
</tr>
<tr>
<td>Deck 6</td>
<td>0.63</td>
<td>0.42</td>
<td>0.33</td>
<td>0.08</td>
<td>0.21</td>
<td>0.32</td>
<td>0.20</td>
<td>0.20</td>
<td>0.81</td>
<td>0.75</td>
</tr>
<tr>
<td>Deck 7</td>
<td>0.59</td>
<td>0.42</td>
<td>0.33</td>
<td>0.08</td>
<td>0.17</td>
<td>0.22</td>
<td>0.18</td>
<td>0.17</td>
<td>0.88</td>
<td>0.00</td>
</tr>
<tr>
<td>Deck 8</td>
<td>0.50</td>
<td>0.52</td>
<td>0.35</td>
<td>0.13</td>
<td>0.21</td>
<td>0.22</td>
<td>0.21</td>
<td>0.21</td>
<td>0.94</td>
<td>0.00</td>
</tr>
<tr>
<td>Deck 9</td>
<td>0.49</td>
<td>0.47</td>
<td>0.33</td>
<td>0.08</td>
<td>0.17</td>
<td>0.17</td>
<td>0.17</td>
<td>0.17</td>
<td>0.94</td>
<td>0.00</td>
</tr>
<tr>
<td>Deck 10</td>
<td>0.49</td>
<td>0.47</td>
<td>0.35</td>
<td>0.00</td>
<td>0.21</td>
<td>0.21</td>
<td>0.21</td>
<td>0.21</td>
<td>0.94</td>
<td>0.00</td>
</tr>
<tr>
<td>Deck 11</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>0.69</td>
<td>0.00</td>
</tr>
</tbody>
</table>

In order to obtain dimensionless values, valid for each ship, the vibration levels have been parameterized as a ratio between the vibration value, at the considered intermediate deck and bottom calculated vibration levels. A typical representation is show for one ship section of Ship 4, in this case for the section 0.3-0.4 (Figure 6-8).

![Figure 6-8](image-url)
Below are reported all the calculated $v(z)$ factors for each of ten ship’s discretized sections:

Section **0-0.1L** : $(v_{z_k})_1 = -20.72z_k^5 + 64.53z_k^4 - 75.01z_k^3 + 38.88z_k^2 - 8.17z_k + 0.95$,  
$k = 1, \ldots, n$

Section **0.1-0.2L** : $(v_{z_k})_2 = -61.59z_k^5 + 157.03z_k^4 - 146.93z_k^3 + 61.39z_k^2 - 10.80z_k + +0.95$,  
$k = 1, \ldots, n$

Section **0.2-0.3L** : $(v_{z_k})_3 = -55.14z_k^5 + 144.89z_k^4 - 139.79z_k^3 + 60.08z_k^2 - 10.94z_k + +0.95$,  
$k = 1, \ldots, n$

Section **0.3-0.4L** : $(v_{z_k})_4 = -150.13z_k^5 + 349.07z_k^4 - 296.17z_k^3 + 110.93z_k^2 - 17.42z_k + +0.96$,  
$k = 1, \ldots, n$

Section **0.4-0.5L** : $(v_{z_k})_5 = -69.20z_k^5 + 182.36z_k^4 - 176.08z_k^3 + 138.92z_k^2 - 18.06z_k + +0.97$,  
$k = 1, \ldots, n$

Section **0.5-0.6L** : $(v_{z_k})_6 = -45.20z_k^5 + 117.56z_k^4 - 116.99z_k^3 + 51.34z_k^2 - 9.62z_k + +0.95$,  
$k = 1, \ldots, n$

Section **0.6-0.7L** : $(v_{z_k})_7 = -65.87z_k^5 + 173.62z_k^4 - 167.77z_k^3 + 71.99z_k^2 - 13.04z_k + +0.94$,  
$k = 1, \ldots, n$

Section **0.7-0.8L** : $(v_{z_k})_8 = -48.88z_k^5 + 131.60z_k^4 - 131.01z_k^3 + 58.86z_k^2 - 11.56z_k + +0.97$,  
$k = 1, \ldots, n$

Section **0.8-0.9L** : $(v_{z_k})_9 = -18.60z_k^5 + 44.61z_k^4 - 41.09z_k^3 + 18.74z_k^2 - 3.94z_k + 0.98$,  
$k = 1, \ldots, n$

Section **0.9-1.0L** : $(v_{z_k})_{10} = -421.18z_k^5 + 649.72z_k^4 - 358.24z_k^3 + 85.93z_k^2 - 8.81z_k + +1.00$,  
$k = 1, \ldots, n$

The total investigated vibrations level is obtained if the previous factors are multiplied by global hull girder response $V_G(x)$ - corrector factor $R(x)$ and by local amplification factors $v(y)$.

### 7. Validation of the new developed Formula

The new developed formula has been applied for vibration level predictions on 4 ships. The calculated vibration velocities have been compared with measurements on board.

**7.1 Global Vibration Level VG(X)** with correction $R(x)$ and vibration decreasing $V(z)$

In Table 7-1 there are the main characteristics of the analyzed ships and in the Figure 7-1, Figure 7-2, Figure 7-3 and Figure 7-4 the dynamic vibration velocity (“constant” line) is computed and then the value is multiplied by the correction factor $R(x)$ (“points”).
Table 7-1 Ship 1, vibration level with and without $v_{y_{kl}}$

<table>
<thead>
<tr>
<th>Main Characteristics</th>
<th>Ship 1</th>
<th>Ship 2</th>
<th>Ship 3</th>
<th>Ship 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_R$ [m]</td>
<td>254.8</td>
<td>185.1</td>
<td>280.8</td>
<td>203.0</td>
</tr>
<tr>
<td>$f$ [Hz]</td>
<td>9.5</td>
<td>9.2</td>
<td>6.9</td>
<td>10.0</td>
</tr>
<tr>
<td>$F$ [N]</td>
<td>13800</td>
<td>14000</td>
<td>23769</td>
<td>41000</td>
</tr>
<tr>
<td>$M$ [Nm]</td>
<td>50000</td>
<td>48000</td>
<td>55000</td>
<td>60000</td>
</tr>
<tr>
<td>$A_{midship}$ [m$^2$]</td>
<td>4.4</td>
<td>3.6</td>
<td>5.7</td>
<td>4.7</td>
</tr>
<tr>
<td>$I_{midship}$ [m$^4$]</td>
<td>421.6</td>
<td>219.4</td>
<td>486.2</td>
<td>472.9</td>
</tr>
</tbody>
</table>

**Ship 1**

![Figure 7-1 Ship 1, global vibration level and absolute value with correction](image)

Table 7-2 Ship 1, vibration level with and without $v_{y_{kl}}$

<table>
<thead>
<tr>
<th>$V(x)<em>i v</em>{y_{kl}} v_{z_{kl}}$ [mm/s]</th>
<th>$V(x)<em>i v</em>{z_{kl}}$ [mm/s]</th>
<th>Deck 12</th>
<th>Deck 9</th>
<th>Deck 12</th>
<th>Deck 9</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4-0.5</td>
<td></td>
<td>0.111</td>
<td>0.018</td>
<td>0.139</td>
<td>0.047</td>
</tr>
<tr>
<td>0.7-0.8</td>
<td></td>
<td>0.111</td>
<td>0.018</td>
<td>0.139</td>
<td>0.047</td>
</tr>
</tbody>
</table>

**Ship 2**

![Figure 7-2 Ship 2, Global vibration level and absolute value with correction](image)
An Enhanced Equation for Vibration Prediction of New Types of Ships

Valter CERGOL, Peter VIDMAR

Table 7-3 Ship 2, vibration level with and without $v_{y_{kl}}$

<table>
<thead>
<tr>
<th>$V(x)<em>i v</em>{y_{kl}} v_{z_{kl}} \omega$ [mm/s]</th>
<th>$V(x)<em>i v</em>{z_{kl}} \omega$ [mm/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deck 6</td>
<td>Deck 5</td>
</tr>
<tr>
<td>0.0-0.1</td>
<td>0.779</td>
</tr>
<tr>
<td>0.1-0.2</td>
<td>0.195</td>
</tr>
</tbody>
</table>

**Ship 3**

![Figure 7-3 Ship 3, Global vibration level and absolute value with correction](image)

Table 7-4 Ship 3, vibration level with and without $v_{y_{kl}}$

<table>
<thead>
<tr>
<th>$V(x)<em>i v</em>{y_{kl}} v_{z_{kl}} \omega$ [mm/s]</th>
<th>$V(x)<em>i v</em>{z_{kl}} \omega$ [mm/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deck 9</td>
<td>Deck 5</td>
</tr>
<tr>
<td>0.4-0.5</td>
<td>0.135</td>
</tr>
<tr>
<td>0.8-0.9</td>
<td>0.018</td>
</tr>
</tbody>
</table>

**Ship 4**

![Figure 7-4 Ship 4, Global vibration level and absolute value with correction](image)

Table 7-5 Ship 4, vibration level with and without $v_{y_{kl}}$

<table>
<thead>
<tr>
<th>$V(x)<em>i v</em>{y_{kl}} v_{z_{kl}} \omega$ [mm/s]</th>
<th>$V(x)<em>i v</em>{z_{kl}} \omega$ [mm/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deck 4</td>
<td>Deck 5</td>
</tr>
<tr>
<td>0.0-0.1</td>
<td>0.904</td>
</tr>
<tr>
<td>0.137</td>
<td>0.528</td>
</tr>
<tr>
<td>0.152</td>
<td></td>
</tr>
</tbody>
</table>
8. Experimental Measurements on board

8.1 Description of measurements procedures

Into the Classification Societies Comfort Class Rules are in general defined the instructions for measurement procedures, condition and limits [4].

a) Measuring equipment
Measurement and calibration equipment are to meet the requirements of ISO 6954 and ISO 8041.

Vibrations calibrations (Figure 8-1) are to be verified at least every year, while the measuring equipment, part of measuring chain, is all together verified at least every two years.

Figure 8-1: FFT Analyser, accelerometers and calibrator in order

This verification is to be done by a national standard laboratory on a competent laboratory.

The instrumentation includes at least one transducer accelerometer (Figure 8-2) with amplifier and the Fast Fourier Transform analyser (Figure 8-2).

The Classification Societies suggests that the instrumentations have to be calibrated in situ before and after the tests.

b) Sea trial conditions
During the sea trials the propeller output has to correspond to the operation conditions of the technical ships specification and not less than 85% of maximum continuous rating.

The test conditions should correspond to the loading conditions defined for sea trials. Vibration measurements have to be performed in sea and weather conditions with sea state 3 or less on the WMO sea state code. The tests have to be performed in deep water, with water depth greater than five times the mean draft. Ship course has to be kept constant, with rudder angle less than two degrees portside or starboard for the duration of the measurements.
c) **Measuring positions**

Measurements are to be taken in vertical direction. In cabins, offices and other small rooms as library, meeting room the measurements are to be taken on the floor in the centre of the room. For longer rooms several measuring points may be required in general based on author experience each measuring points covering 15-20 m².

Vibrations are to be measured in all accommodation spaces, navigation, wheelhouse room and crew spaces.

In order to define the location and number of measuring points in general the length of the ships is divided in two parts:

- from the aft part of the ship to the front bulkhead of the casing:
  - minimum of 20% of cabins;
  - all public spaces and open daily.

  For long public areas (lounges, restaurants, etc.) measurements are to be carried out in different locations. At first tentative each measuring points covering less than 80 m². In case the measured levels are at limit or above the covering measuring point area should be reduced to 15-20 m², based on author experience.

  It is reasonable to divide the ship in length in two parts, because in the aft part from transom to the fore engine room bulkhead, are located all the significant sources of vibration:
  - propellers;
  - main propulsion engines;
  - diesel generators.

- from the front bulkhead of the casing to the fore end of the ship:
  - minimum of 10% of cabins;
  - all public spaces and open deck.

  For large public rooms measurements are to be carried out with each measuring point covering less than 150m².

d) **Technical organization**

The measurements are undertaken or by the Classification Societies or by an approved organization that satisfied all the criteria listed out below:

- Have instrumentation whose calibration, both before and after the measurements, can be traced back to National Standards and, hence, back to International Standards.
- Have analysis procedures capable of data reduction to the requirements and standards set out in these Rules.
- Be able to provide a written report in English with contents.

8.2 Vibration levels measurements on board during sea trials

In order to validate the developed simplified approach and the newly developed formula, several measurements on real ships have been performed in agreement with the procedures described above. Here only the most representative and more significant examples of two measurements for each ship are presented.
**Measured Values - Ship 1**

*Figure 8-2* Measurement points on deck 12 (left) and on deck 9 (right)

*Figure 8-3* – Ship 1 deck 12 - measure 8 (left) and deck 9 - measure 39 (right)
An Enhanced Equation for Vibration Prediction of New Types of Ships

Valter CERGOL, Peter VIDMAR

**Measured Values - Ship 2**

*Figure 8-4* Measurement points on deck 6 (left) and 5 (right)

*Figure 8-5* Ship 2 deck 5 - measure 021 (left) and deck 6 - measure 018 (right)
Measured Values - Ship 3

Figure 8-6 Measurement points on deck 9 (left) and on deck 5 (right)

Figure 8-7 Ship 3 deck 9 - measure 241 (left) and deck 5 - measure 143 (right)
An Enhanced Equation
for Vibration Prediction of New Types of Ships
Valter CERGOL, Peter VIDMAR

Measured Values - Ship 4

Figure 8-8 Measurement points on deck 4 and 5

Figure 8-9 Ship 4 deck 4 - measure 20 (left) and deck 5 - measure 22 (right)

9. Comparative Analysis and Error Calculation

In order to validate the developed mathematical-physical model approach [35,36] in this section the comparison between the calculated vibration levels and the ones measured on board has been performed.

In Table 9-1 the calculated and measured data are shown and compared. The difference has been calculated, but in order to obtain a quantity able to indicate the reliability of the analytical predictions, the absolute error has been compared with the ISO6954-1984 lower bound limit, vibration level 4mm/s, see [37,38,39]. This ratio will be called Error Index 4(EI4)
\[
\text{Error index 4} = \frac{\text{calculated} - \text{measured}}{4}
\]

The original particular choice of the Error index 4 is based on author experience gained during presence at several sea trials measurements. Investigating more deeply the real comfort on board, the main feature of ISO 6954-1984 is that to represent the human perception of comfort expressed as maximum 0-peak value for each frequency. The author selection of 4 mm/s is based on statement from this norm “adverse comment not probable”, which define the area below the lower limit curve in Figure 9-1.

![Figure 9-1: ISO 6954-1984 vibration limits](image)

This boundary curve is very effective and of general acceptance for real defining and human perception of the basic level of comfort. This is the reason for author decision to this New original definition.

**Ship 1**

**Table 9-1** Ship 1, measured and calculated vibration level

<table>
<thead>
<tr>
<th></th>
<th>x/L</th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.7-0.8</td>
<td>0.4-0.5</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>calculated [mm/s]</td>
<td>measured [mm/s]</td>
<td>error index 4</td>
<td>calculated [mm/s]</td>
<td>measured [mm/s]</td>
<td>error index 4</td>
</tr>
<tr>
<td>deck12</td>
<td>0.111</td>
<td>0.200</td>
<td>-0.022</td>
<td>0.139</td>
<td>0.410</td>
<td>-0.068</td>
</tr>
<tr>
<td>deck9</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Ship 2
Table 9-2 Ship 2, measured and calculated vibration level

<table>
<thead>
<tr>
<th></th>
<th>x/L</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.0 - 0.1</td>
<td>0.1 - 0.2</td>
<td></td>
</tr>
<tr>
<td></td>
<td>calculated [mm/s]</td>
<td>measured [mm/s]</td>
<td>error index 4</td>
</tr>
<tr>
<td>deck6</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>deck5</td>
<td>0.779</td>
<td>0.382</td>
<td>0.099</td>
</tr>
<tr>
<td></td>
<td>0.144</td>
<td>0.441</td>
<td>-0.074</td>
</tr>
</tbody>
</table>

Ship 3
Table 9-3 Ship 3, measured and calculated vibration level

<table>
<thead>
<tr>
<th></th>
<th>x/L</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.4 - 0.5</td>
<td>0.8 - 0.9</td>
<td></td>
</tr>
<tr>
<td></td>
<td>calculated [mm/s]</td>
<td>measured [mm/s]</td>
<td>error index 4</td>
</tr>
<tr>
<td>deck9</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>deck5</td>
<td>0.135</td>
<td>0.058</td>
<td>0.019</td>
</tr>
<tr>
<td></td>
<td>0.143</td>
<td>0.135</td>
<td>0.002</td>
</tr>
</tbody>
</table>

Ship 4
Table 9-4 Ship 4, measured and calculated vibration level excitation

<table>
<thead>
<tr>
<th></th>
<th>x/L</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.0 - 0.1</td>
<td></td>
</tr>
<tr>
<td></td>
<td>calculated [mm/s]</td>
<td>measured [mm/s]</td>
</tr>
<tr>
<td>deck5</td>
<td>0.528</td>
<td>0.430</td>
</tr>
<tr>
<td>deck4</td>
<td>0.904</td>
<td>0.980</td>
</tr>
</tbody>
</table>

10. Discussion
The results obtained for all the studied ships were compared to experimental sea trial measurements. In Table 10-1 and in Figure 10-1 we show the Error Index 4 for each measurement point. For major reasons of space in this paper there are only the two most significant points for each ship. In the whole author’s data base there are a large number of measurement points. The maximum positive Error Index 4, when the calculated value overestimates the measured value, is lower than 10%. The maximum negative Error Index 4, when calculated value underestimates the measured value, is lower than 7.5%.
11. Conclusions

The developed approach is based on the assumption that the ship hull can be represented as a non-uniform section beam. The basic idea for the development of the new formula is that the local vibrations on ship structures are superimposed to global hull vibrations. It has been also considered that the vibration varies with deck position in height.

Furthermore, the results obtained from FEM analyses have been used for the calculation of the coefficients necessary to improve the accuracy of results obtained with the “new developed formula”.

The decrease of vibrations with height -z coordinate- was also obtained with the use of a statistical method and FEM results of an entire ship model.

The presented mathematical calculations bring to a fully consistent formula which was the primary scope of this paper. The results obtained with the use of the new formula were then compared to data from measurements on real passenger and RO-RO passenger ships. The comparison between calculated and measured vibrations values were performed for the most significant areas of different ships. The obtained results show that the maximum calculated Error Index is 10%.

REFERENCES

An Enhanced Equation for Vibration Prediction of New Types of Ships

Valter CERGOL, Peter VIDMAR


Submitted: 12.02.2015 Valter CERGOL, cergol@cergolengineering.com
Accepted: 29.04.2015 Peter VIDMAR
Faculty of Maritime Studies and Transport, University of Ljubljana, Pot pomorščakov 4, 6320 Portorož, Slovenia
<table>
<thead>
<tr>
<th>Title</th>
<th>Authors</th>
<th>Pages</th>
</tr>
</thead>
<tbody>
<tr>
<td>HYDRODYNAMIC PERFORMANCES OF SMALL SIZE SWATH CRAFT</td>
<td>Ermina Begovic, Carlo Bertorello, Simone Mancini</td>
<td>1-22</td>
</tr>
<tr>
<td>A COMPUTATIONAL HYDRODYNAMIC ANALYSIS OF DUISBURG TEST CASE WITH FREE SURFACE AND PROPELLER</td>
<td>Omer Kemal Kinaci, Metin Kemal Gokce</td>
<td>23-38</td>
</tr>
<tr>
<td>A REAL TIME DECISION SUPPORT SYSTEM FOR THE ADJUSTMENT OF SAILBOAT RIGGING</td>
<td>Inmaculada Ortigosa, Julio García-Espinosa, Marcel la Castells</td>
<td>39-56</td>
</tr>
<tr>
<td>STEADY STATE PERFORMANCES ANALYSIS OF MODERN MARINE TWO-STROKE LOW SPEED DIESEL ENGINE USING MLP NEURAL NETWORK MODEL</td>
<td>Ozren Bukovac, Vladimir Medica, Vedran Mrzljak</td>
<td>57-70</td>
</tr>
<tr>
<td>OPTIMIZATION PROCEDURE FOR PRELIMINARY DESIGN STAGE OF CAIRO-DAMIETTA SELF-PROPELLED GRAIN BULK SHIPS</td>
<td>M.M. Moustafa</td>
<td>71-85</td>
</tr>
<tr>
<td>TURKISH SHIPBUILDING INDUSTRY – CHALLENGES AND POTENTIAL</td>
<td>Eda Turan, Hülya Cengiz</td>
<td>87-101</td>
</tr>
</tbody>
</table>
HYDRODYNAMIC PERFORMANCES OF SMALL SIZE SWATH CRAFT

UDC 629.5.52:629.5.022.25:629.5.015.2
Original scientific paper

Summary

The good seakeeping characteristics of SWATH hull form are very interesting for small working craft and pleasure boats. Intrinsic limitations as the low values of weight per inch of immersion and transversal and longitudinal instability, can be acceptable and successfully managed when the mission profile does not ask for significant load variation and shift. The exploitation of SWATH concept is limited by the craft size, but if main dimensions allow enough static stability, this configuration appears very promising. SWATH behaviour in rough sea at zero and low speed have led to consider this hull form within the small craft design research program in progress at University of Naples Federico II.

The design of small size SWATH working/pleasure craft has to begin from the consideration of strut waterplane areas that are the key factor to get acceptable static and dynamic stability. Displacement has to be reduced as most as possible to increase static stability, as shown by last design trends. The results of CFD analysis concerning SWATH resistance and propulsion, aspects are presented. A numerical evaluation of the hull-propeller interactions is performed, through simulations of self-propulsion tests with a simplified method (Actuator Disk model) to discretize the propeller effect. The effective wake coefficient, the thrust deduction fraction and hull efficiency are provided. To validate CFD resistance results a comparison with experimental tests performed by Authors is reported.

The presented work highlights different hydrodynamic aspects, comments advantages and critical issues of SWATH concept and reports detailed CFD modelling procedure with the aim to provide a reference for SWATH small craft design.

Key words: SWATH hull form design; SWATH dynamic instability; SWATH CFD resistance assessment; numerical self-propulsion test; propulsion coefficients;

1. Introduction

The growing activities for offshore wind farms maintenance and survey, as well as the interest about small suppliers for coastal oil platforms, focused small craft designer attention about the lowest possible wave responses at zero or very low speed. The ability to be moored or to stay close to a fixed body with the smallest possible vertical motions is very important for the successful design of small craft operating at open sea. Reduced vertical motions are considered a design criteria and represent the main limit for the ship operability.
Although SWATH craft presents very good seakeeping characteristics, when main dimensions are small, the low values of unit displacement and of transversal and longitudinal stability limit a sound application of such hull form.

The idea of SWATH derives from semi-submersible offshore rigs, which are designed to provide a working platform with minimized motions in open sea. The buoyancy of a SWATH ship is mainly provided by its submerged torpedo-like bodies, which are connected by single or twin struts to the upper platform. The waterplane is minimised to reduce the vertical loads induced by waves. The vertical motions are quite smaller in comparison to catamarans and monohulls of similar displacement, although SWATH behaviour has to be carefully studied to avoid resonance in working conditions.

Beside these very interesting characteristics, SWATHs pose some problems. They are very sensitive to weight distribution and dynamically instable when relative speed increases. As regards motion resistance their performances are in the same range of conventional hull forms, except at very low relative speeds when larger wet surface worsen their performances. In following sea, the pitch response is of particular concern because very large encounter periods, close to the pitch resonance, are likely to occur over a wide range of wave lengths.

SWATH concept has been considered for more than forty years and several modifications of the original basic concept have been performed. At present, the four struts configuration appears preferable versus the twin one and it is used for the small size craft considered in this paper. The reason lies in better longitudinal stability due to the optimal longitudinal distribution of strut waterplane areas.

Strut waterplanes can have different shapes, in any case very slender. A very effective and commonly chosen shape has parabolic sides, symmetrical in respect to both strut centre line and mid length. Torpedos are very slender bodies and can be simplified to get simple and cheap construction. Generally, they present an hemispheric bow, a cylindrical part and a conic tail, as shown in Figure 1. There is plenty of research about such bodies totally immersed in fluid. Streamlined longitudinal profiles are generally superior in terms of resistance; some designs have considered these more complicated shapes when dealing with resistance optimisation for higher speed. Basic circular sections have been modified in some cases when an optional catamaran cruising condition is considered. Anyway, the above general considerations have to be revised to fit the SWATH vessel peculiar characteristics and mission profile.

Fig. 1 SWATH hull components
Since the milestone work of Lee and Curphey [1] it has been pointed out that:

- SWATH ships are dynamically unstable and there is a necessity of fins to maintain heave and pitch stability at high speeds;
- peak motions in heave, roll and pitch are excited in longer waves;
- wave exciting heave force and pitch moment acting on SWATH are smaller than those acting on a monohull of comparable displacement.

Salvesen et al [2] developed a numerical method for wave resistance optimization of torpedo hulls valid for any cross section shape, not only bodies of revolution. Furthermore, authors considered struts shape as a constrain and wave minimization is obtained by reshaping the torpedo hulls. Papanikolau et al [3] applied optimization procedure based on Salvesen [2] method for SWATH in the range of Froude numbers from 0.3 to 0.7. According to the service speed, the displacement of torpedo tends to be concentrated around the midship section (for low Fr) or towards the ends (for high Fr). For this optimized SWATH, Schellin et al [4] reported seakeeping predictions compared against experimental data indicating good overall performances but also necessity of stabilizing fins. Beena et al [5] presented seakeeping analysis of different torpedo forms, and discussion on the appropriate criteria for pitch, roll and vertical accelerations. Authors suggested that higher amplitudes may be imposed in the range of frequencies typical for SWATH motions, identifying a limiting value of 8 degree for roll and 0.4g for vertical acceleration and 3 degree for pitch RMS. Yoshida [6] introduced “resonance free SWATH” obtained by diminution of strut length to approximately one half of torpedo length. Brizzolara et al [7, 8] presented an example of the so called “2nd generation SWATH”. Authors reported that for \( Fr > 0.5 \), the position of contracted section remains almost invariably at midship, independently from the prismatic coefficient \( C_p \), the slenderness coefficient \( L/V^{1/3} \) or the ratio \( L/D \). As maximum weight and dimensions were the assigned design constraints, the strut length decreased drastically, and to obtain the sufficient initial transversal and longitudinal metacentric height, four struts were chosen and positioned at the extreme bow and stern of the torpedo hull. Qian et al [9] investigated a SWATH vehicle with inclined struts by numerical and experimental methods reporting that the inclined struts performed better seakeeping performances in comparison with a vertical-strut SWATH. Begovic and Bertorello [10] presented SWATH yacht design reporting extensive resistance and seakeeping experimental results performed at University of Naples Federico II.

The present paper reports a numerical approach of hydrodynamic assessment of SWATH concept. Numerical simulations in CD Adapco StarCCM+ are performed for four model speeds. The paper reports complete procedure and comparison with experimental data obtained by Authors and presented in previous work [9]. For the considered speeds, the simulations of self-propulsion tests have been performed and the results are elaborated in terms of thrust deduction, wake and hull efficiency coefficients.

Dynamic instability of SWATH craft at higher speeds, resulting in excessive dynamic trim, is counteracted in service by active stabilizing fins. In the CFD simulations, three different setups have been considered: hull restrained for trim only, restrained for sinkage and trim, free for sinkage and trim. Experimental tests have been performed with trim restrain acting as soon as the minimum trim angle has been recorded. This in order to appreciate the maximum possible free ride speed and then to consider the action of trim correctors. Experimental setup is very close to restrained model numerical simulation set up.

The present work highlights different hydrodynamic aspects, commenting advantages and critical issues of SWATH concept, reports detailed CFD procedure for resistance and propulsion test simulation, and can be used as a reference for SWATH small craft design.
2. DESIGN of SWATH small size pleasure/working craft

Several developments of the SWATH concept have been tried since the first applications in the late sixties. Although the expected seakeeping performances are related to smallest possible waterplanes, it must be considered that SWATH vessels, as any floating body follow Euler’s law for ship stability. This means that both waterplane area, through second order moment, and hull volume, that is to say ship weight, affect metacentric radius. This aspect has to be considered at preliminary design stage taking into account both factors as well as a realistic CG vertical position.

Generally, static stability is not problematic for larger SWATH vessels but it becomes a design criteria for smaller craft and represents the lowest limit to main dimensions for a practical application of the concept. The use of composites in some of the most recent small size constructions could be surprising at first sight, but witnesses the importance of the reduction of structural weight and displaced volume related to stability issues. Within design development, the general used procedure to check the (transversal) stability of a given hull form according to defined criteria, could lead to significant loss of design time. For SWATHs it is preferable to identify the maximum allowable heeling angles (both longitudinal and transversal) for a given heeling moment coherent with mission profile and ship characteristics.

The scale model considered in this paper is relative to a SWATH 32 m $L_{OA}$ motor yacht. For this ship, a value of 2 degrees maximum longitudinal or transversal inclination due to 6 crew and 12 passengers crowding on one side has been assumed. This value is quite lower than the maximum of 10 degrees allowed by Classification Societies (RINA, Rules for Pleasure Yachts 2013, Ch.6, Par. 2/2.1/2.1.1/g).

Then the waterplane area second order moment can be calculated, and stability evaluated for Displacement and VCG based on reliable data. The second order moment is divided into two or four contributes according to the chosen number of struts and the single strut waterplane can be drawn. This last has parabolic sides and cord/camber ratio not lower than six, if service speed leads to a significant wave resistance component. In case of very slow vessels, larger and squatter waterplane can be accepted. With this procedure, the strut waterplane area is set to the minimum possible value and SWATH concept used at its best.

From these considerations it is evident that it is very difficult to use SWATH concept for small craft. In practice, there is no manned SWATH vessel below 12 m $L_{OA}$.

The present design trend features four struts and twin torpedoes configuration. Above the struts a catamaran demi-hull with trapezoidal transversal sections assures buoyancy and stability when longitudinal and/or transversal heeling angles reach dangerous values. This configuration reported in the following Figure 2 appears the most promising to exploit SWATH peculiar characteristics in terms of reduced wake and motions with adequate safety margins provided by the upper catamaran hulls. Strut height is a compromise between the highest encounter wave height and maximum allowable draft. A V shaped cross deck bottom can add useful damping in case of impact with the sea surface. This configuration can be defined through few significant parameters as reported in the following Table 1. SWATH reported in Fig. 2 is relative to a pleasure/small working craft 1/32 scale model developed at the University of Naples Federico II.
The parameters with intrinsic limitations are:
- strut camber, to allow access to the engine inside torpedo;
- torpedo diameter, to provide enough room for engine housing;
- minimum waterplane area, to allow positive metacentric height at rest.

These considerations are valid in the most common case of diesel propulsion with each engine housed into one torpedo and the obvious need to access it for control and maintenance without dry dock the ship. In case of different system layouts or if dry dock is considered for any type of engine maintenance (as in case of full electric propulsion) the previous considerations can be forgotten and the strut camber can be optimized and limited only by stability.
2.1 Dynamic Stability

When dynamic forces and moments became of the same order of hydrostatic ones SWATH configuration became intrinsically unstable. Longitudinal instability is much more evident as longitudinal position of centre of hydrodynamic forces shifts according to speed variation and the longitudinal equilibrium with the weight forces applied in the LCG is modified resulting in large trim changes. In case of very small strut waterplane areas, a stable trim at rest does not assure stability at speed as in standard displacement vessels and becomes necessary to consider the involved factors to identify adequate countermeasures.

Although torpedoes are generally axial symmetrical bodies, the pressure field around them is not. They are not deeply immersed and the pressure field around them is influenced by the free surface. This phenomenon is well known as surface suction in submarine hydrodynamics, when such vessels travel close to the free surface, Renilson [11, 12, 13], Bhattacharyya [14].

At speed, the forward strut wave system further modifies the free surface and the pressure field around the torpedo. In calm water, the surface suction is due to the Venturi effect, which results in higher flow velocity between torpedo and free surface and therefore lower pressure. This causes a pitch moment, variable with torpedo’s forward speed and with immersion.

Struts resistance has different contribution in respect to the torpedoes one as speed increases. At lowest speed there is almost no wave system around the strut in an almost total viscous and viscous pressure resistance regime; the wetted surface is the key parameter and the struts have quite less influence in respect to torpedoes. Increasing relative speed the struts, resistance is increased significantly by wave component. This factor enhances the shifting upward of resistance point of application and can explain the observed stern down trim at speed.

Nevertheless, SWATH longitudinal instability cannot be so easily simplified. Even if a small pitch angle is given to torpedoes, they act as foil and tend to increase the angle as speed increases. The strut unitary displacement is too low to counteract such effect and the hull can change trim stern or bow down according to the initial trim perturbation. The trim change can be stopped only by larger volume reaching the water with large resistance increase or by sudden intervention of trim correctors.

After these considerations, it is evident that a full exploitation of the SWATH concept is possible only through an effective stabilizing device which action must be more powerful as speed increases. Transversal dynamic stability is generally larger and adequate at any speed.

3. NUMERICAL ASSESSMENT OF SWATH HYDRODYNAMIC PROPERTIES

Numerical simulations in CD Adapco Star-CCM+ v9.06 RANS software have been performed for four model speeds: 0.976, 1.301, 1.952, 2.603 m/s corresponding to \( Fr_T = 0.316, 0.422, 0.633 \) and 0.844. For the considered model-ship scale \( \lambda = 32 \), these velocities correspond to full scale speed of 10.73, 17.89, 21.47 and 28.62 kn, i.e. speed range reasonable for small size craft service. The SWATH intrinsic instability, required a non conventional approach for the numerical simulation of resistance test using the Chimera grid, also called overset grid technique. In particular this approach permit to not loose numerical accuracy in inclined positions. The unstructured mesh with the fixed and moving regions, shown in Fig. 3, is used for the computations. The grid sensitivity analysis has been performed for resistance test at model speed of 0.976 m/s for three different grids, reported in details in Section 3.4. The chosen mesh, i.e. number of cells and base sizes for both tank (fixed) and overset regions are given in Table 2.
Table 2 Mesh properties summary

<table>
<thead>
<tr>
<th>Type of mesh</th>
<th>Trimmed/ polyhedral</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of cells – tank</td>
<td>358468</td>
</tr>
<tr>
<td>Base size - tank (m)</td>
<td>2.3</td>
</tr>
<tr>
<td>No. of cells – overset</td>
<td>1210190</td>
</tr>
<tr>
<td>Base size – overset (m)</td>
<td>0.65</td>
</tr>
</tbody>
</table>

Fig. 3a Numerical set up- Mesh used and different regions visualization: fixed - blue and overset - gray

Fig. 3b Numerical set up- Mesh used and different regions visualization: fixed - blue and overset - gray
3.1 Numerical set up

To solve the time-marching equations, an implicit solver is used to find the field of all hydrodynamic unknown quantities, in conjunction with an iterative solver to solve each time step. The software uses a Semi Implicit Method for Pressure Linked Equations to conjugate pressure field and velocity field, and an Algebraic Multi-Grid solver to accelerate the convergence of the solution. The free surface is modelled with the two phase volume of fluid technique (VoF). A segregated flow solver approach is used for all simulations.

Coordinate system origin has been imposed in centre of buoyancy of torpedo hull.

The Reynolds stress problem is solved by means of K-Omega SST turbulence model and the All Wall \( y^+ \) is the wall treatment utilized for all simulations. It is a hybrid approach that attempts to emulate the high \( y^+ \) wall treatment for coarse meshes (for \( y^+ > 30 \)), and the low \( y^+ \) wall treatment for fine meshes (for \( y^+ \approx 1 \)). It is also formulated with the desirable characteristic of producing reasonable answers for meshes of intermediate resolution (for \( y^+ \) in the buffer layer), CD Adapco Star-CCM+ v9.06 User’s Guide [20]. This approach is considered a reasonable compromise among the acceptable quality of the boundary layer description and the calculation time. The \( y^+ \) variation on the strut and torpedo is given in Fig. 4a-d for the four velocities examined. It can be seen that \( y^+ \) values range from 0 to 25.

**Fig. 4a** \( y^+ \) values in 0DOF simulation at model speed 0.976 m/s

**Fig. 4b** \( y^+ \) values in 0DOF simulation at model speed 1.301 m/s
Hydrodynamic Performances of Small Size SWATH Craft

Fig. 4c y+ values in 0DOF simulation at model speed 1.952 m/s

Fig. 4d y+ values in 0DOF simulation at model speed 2.601 m/s

All properties of the numerical solver are reported in Table 3. Boundary conditions are illustrated in Fig. 5 and listed in Table 4.

**Table 3** Numerical simulation set up summary

<table>
<thead>
<tr>
<th>Pressure link</th>
<th>SIMPLE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure</td>
<td>Standard</td>
</tr>
<tr>
<td>Convection Term</td>
<td>2nd Order</td>
</tr>
<tr>
<td>Temporal Discretization</td>
<td>1st Order</td>
</tr>
<tr>
<td>Time-step (s)</td>
<td>0.015</td>
</tr>
<tr>
<td>Iteration per t.s.</td>
<td>5</td>
</tr>
<tr>
<td>Turbulence Model</td>
<td>K-Omega SST</td>
</tr>
</tbody>
</table>

**Table 4** Boundary conditions set up summary

<table>
<thead>
<tr>
<th>Inlet</th>
<th>Velocity inlet condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outlet</td>
<td>Pressure outlet condition</td>
</tr>
<tr>
<td>Bottom/Top</td>
<td>Velocity inlet condition</td>
</tr>
<tr>
<td>Side</td>
<td>Symmetry condition</td>
</tr>
<tr>
<td>Hull</td>
<td>Wall with no-slip condition</td>
</tr>
<tr>
<td>Symmetry plane</td>
<td>Along centreline of the hull</td>
</tr>
</tbody>
</table>
E. Begovic, C. Bertorello, S. Mancini

Hydrodynamic Performances of Small Size SWATH Craft

Fig. 5 Boundaries set up

All calculations have been performed at Calculation Centre of University of Naples Federico II, using 32 processors. For 30 seconds of physical time of resistance simulations, calculation time was about 30 hours.

3.2 Bare hull calm water resistance

Dynamic instability of SWATH craft at higher speeds, resulting in excessive dynamic trim, can be effectively counteracted in service by active stabilizing fins and in some cases by water ballast shifting. To get the whole picture of the possible running conditions, in the CFD simulations three different setups have been considered: hull restrained for trim only, restrained for sinkage and trim, free for sinkage and trim (2DoF). Simulations are performed for model speeds: 0.976, 1.301, 1.952 and 2.603 m/s. All CFD calculations are performed in model scale to compare easily and immediately numerical and experimental results. In coherence the grid sensitivity study was done at model scale. The aims of the paper are to validate CFD as design tool for swath configuration and to explore and identify speed limits for pure SWATH hullforms.

In the simulation with model free to heave and pitch (2DOF), the SWATH dynamic instability is observed at tested speed 1.952 m/s and 2.603 m/s. The result is aft down trim unless the catamaran hulls touch the water and provide restoring moment, as shown in Fig. 6 (lower).

In the second set of simulations (model restrained for pitch, and free only to heave), at higher speed ($v = 1.952$ and $2.603$ m/s), the pitch constrain results in a downward force leading to strut further immersion as can be observed in Fig. 7 (lower). Finally, simulations for model restrained to heave and pitch have been performed and these results are illustrated in Fig. 8. In all simulations SWATH initial trim is zero degree.
Fig. 6 SWATH resistance simulation results with model free for heave and pitch

Fig. 7 SWATH resistance simulation results with model free to heave
Numerical results for total resistance, frictional resistance, trim, sinkage and wetted surface are summarised in Table 5, while in Fig. 9 wave making and frictional resistance coefficients curves for different simulation setups are shown.

**Table 5** Numerical results summary

<table>
<thead>
<tr>
<th>2 DOF</th>
<th>(v_{\text{model}}) (m/s)</th>
<th>(R_T) (N)</th>
<th>(R_F) (N)</th>
<th>Trim (deg)</th>
<th>Sinkage (m)</th>
<th>WS (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.976</td>
<td>3.046</td>
<td>1.184</td>
<td>-0.500</td>
<td>-0.017</td>
<td>0.412</td>
</tr>
<tr>
<td></td>
<td>1.301</td>
<td>3.614</td>
<td>2.004</td>
<td>1.300</td>
<td>-0.025</td>
<td>0.434</td>
</tr>
<tr>
<td></td>
<td>1.952</td>
<td>8.77</td>
<td>4.506</td>
<td>-12.930</td>
<td>-0.004</td>
<td>0.450</td>
</tr>
<tr>
<td></td>
<td>2.603</td>
<td>12.984</td>
<td>6.906</td>
<td>-15.350</td>
<td>0.001</td>
<td>0.444</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>1 DOF</th>
<th>(v_{\text{model}}) (m/s)</th>
<th>(R_T) (N)</th>
<th>(R_F) (N)</th>
<th>Trim (deg)</th>
<th>Sinkage (m)</th>
<th>WS (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.976</td>
<td>3.064</td>
<td>1.184</td>
<td>0.000</td>
<td>-0.017</td>
<td>0.412</td>
</tr>
<tr>
<td></td>
<td>1.301</td>
<td>3.572</td>
<td>1.97</td>
<td>0.000</td>
<td>-0.023</td>
<td>0.424</td>
</tr>
<tr>
<td></td>
<td>1.952</td>
<td>7.402</td>
<td>4.292</td>
<td>0.000</td>
<td>-0.046</td>
<td>0.482</td>
</tr>
<tr>
<td></td>
<td>2.603</td>
<td>12.644</td>
<td>7.368</td>
<td>0.000</td>
<td>-0.059</td>
<td>0.496</td>
</tr>
</tbody>
</table>
From resistance coefficients curves, shown in Fig. 9, it can be observed the effect of trim and sinkage on resistance performances. At first two speeds, the differences in frictional resistance coefficients are minimal as the waterplane area is almost identical for small variation of trim. There is no significant changing in wave pattern and in wetted surface among different simulations. This also means that the possible idea without constrain or stabilization for this SWATH configuration is up to Froude number $Fr_T$ 0.422. At two higher speeds, trim angles obtained in simulation 2DOF are about 13 and 15 degrees, the wetted surface is about 15% higher than the one at rest and difference in total resistance between 2 DOF and 0 DOF is about 30%. As it can be observed in Fig. 9, the frictional resistance coefficient is slightly higher and the wave making resistance coefficient has been almost doubled with respect to 0 DOF.

The obtained results and moreover the sensibility of results with respect to degrees of freedom allowed in simulation are very important as it confirms the capacity of numerical method to predict reliable results. Further achievement is the possibility to use the results from 2DOF and 1DOF simulation as input for trim corrector design, knowing exactly the trimming moment or the downward force to counteract. Furthermore, images and results also indicates that optimizing torpedo hull for wave resistance by potential flow solvers without considering free surface effects would lead to unrealistic conclusions for SWATH design and service capabilities.
3.3 Self-propulsion test simulations

The simulation of the propeller action and the individuation of the “self propelled point” has been performed using a uniform volume force distribution over a cylindrical disk having the same position and diameter of the propeller, called “Virtual Disk Model” (VD). The volume force varies in radial direction and the distribution of the force components, according to Visonneau et al [15] and Bugalski [16] is given by:

\[ f_{\text{ax}} = A_x \cdot r' \sqrt{1 - r'} \]  
(1)

\[ f_{\text{at}} = A_y \cdot \frac{r' \sqrt{1 - r'}}{r' (1 - r'_h) + r'_h} \]  
(2)

\[ r'_h = \frac{R_H}{R_P} \quad r' = \frac{r}{R_P} \]  
(3)

where \( f_{\text{ax}} \) is the body force component in axial direction, \( f_{\text{at}} \) is the body force component in tangential direction, \( r \) is the radial coordinate, \( R_H \) is the hub radius and \( R_P \) the propeller tip radius. The constants \( A_x \) and \( A_y \) are computed as

\[ A_x = \frac{105}{8} \cdot \frac{T}{\pi \cdot d \cdot (3R_H + 4R_P) \cdot (R_P - R_H)} \]  
(4)

and

\[ A_y = \frac{105}{8} \cdot \frac{Q}{\pi \cdot d \cdot R_P \cdot (3R_H + 4R_P) \cdot (R_P - R_H)} \]  
(5)

where \( d \) is the propeller thickness and \( T \) and \( Q \) are thrust and torque respectively. The computation of the body force components necessitates several user inputs. A propeller performance curve needs to be specified, which gives the non-dimensional thrust and torque fractions \( K_T \), \( K_Q \) and the propeller efficiency \( \eta_0 \) as functions of the advance ratio \( J \)

\[ J = \frac{V_A}{n \cdot D} \]  
(6)

where \( V_A \) is the speed of advance of the propeller, \( n \) the rotation rate, and the \( D \) propeller diameter. Further inputs are the position of the propeller within the computational domain, the direction of the propeller rotational axis, and the direction of rotation, shown in Fig. 10.

---

Fig. 10 Position and diameter of Virtual Disc
For the simulations of self propelled model, the restrained model set up is used. The chosen propeller is B Wageningen 5 blades series. All parameters of propeller are reported in Table 6, together with $K_T$, $K_Q$ and $\eta_0$ diagram given in Fig. 11.

**Table 6** Propeller chosen for simulation of self propelled model

<table>
<thead>
<tr>
<th>No. blades</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_E/A_0$</td>
<td>0.85</td>
</tr>
<tr>
<td>$P/D$</td>
<td>0.95</td>
</tr>
<tr>
<td>$D$ (m)</td>
<td>0.05</td>
</tr>
<tr>
<td>$\eta_0$</td>
<td>0.6289</td>
</tr>
</tbody>
</table>

![Propeller characteristics - input data](image)

**Fig. 11** Propeller characteristics - input data

Results of numerical self propulsion test are thrust force and wake coefficient. From thrust force the thrust deduction coefficient $I-t$ is calculated and finally hull efficiency coefficient $\eta_H$. All results are given in Table 7 and in Fig. 12. Axial velocity field on virtual disc for model speeds 0.976, 1.301, 1.952 and 2.600 m/s is shown in Fig. 13, 14, 15 and 16, respectively.

**Table 7** Results of self-propulsion numerical simulations

<table>
<thead>
<tr>
<th>$v_{model}$ (m/s)</th>
<th>$R_{SWATH}$ (N)</th>
<th>$T$ (N)</th>
<th>$I-t$</th>
<th>$w$</th>
<th>$I-w$</th>
<th>$\eta_H$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.976</td>
<td>2.8000</td>
<td>2.8036</td>
<td>0.9987</td>
<td>0.0570</td>
<td>0.9430</td>
<td>1.0591</td>
</tr>
<tr>
<td>1.301</td>
<td>3.1780</td>
<td>3.1818</td>
<td>0.9988</td>
<td>0.0171</td>
<td>0.9829</td>
<td>1.0162</td>
</tr>
<tr>
<td>1.952</td>
<td>5.9232</td>
<td>5.9302</td>
<td>0.9988</td>
<td>0.0077</td>
<td>0.9923</td>
<td>1.0066</td>
</tr>
<tr>
<td>2.603</td>
<td>9.6558</td>
<td>9.6676</td>
<td>0.9988</td>
<td>0.0047</td>
<td>0.9953</td>
<td>1.0035</td>
</tr>
</tbody>
</table>

![Hull efficiency, Thrust deduction and Wake fraction](image)

**Fig. 12** Hull efficiency, Thrust deduction and Wake fraction coefficients
Kennell [17] reports SWATH thrust and wake coefficients experimental data and highlights the wake differences between monohull and SWATH. The major feature of SWATH wake is the strong wake deficit above the centerline of the propeller caused by the retarded flow due to the strut boundary layer. It was pointed out that the wake of SWATH at a particular propeller radius is uniform. Both conclusions have been observed and confirmed by performed CFD simulations. Furthermore, it can be seen that the SWATH hull efficiency coefficient $\eta_H$ is close to 1, indicating considerably higher efficiency than monohull vessels, results obtained also by Kennell [17].

### 3.4 Grid Sensitivity Analysis

The grid sensitivity analysis has been performed for resistance test at model speed of 0.976 m/s for the simulation case 0 DOF, applying RANSE solver with exactly the same numerical model, discretization schemes and parameter settings on three different grids: Fine, Medium and Coarse. Starting from the coarse grid, the mesh size is progressively increased by a refinement ratio equal to $\sqrt{2}$ (called grid refinement index). The grid sizes and the computed model resistance are given in Table 8 and in Fig. 17.

<table>
<thead>
<tr>
<th>Grid</th>
<th>Mesh size</th>
<th>$R_{SWATH}$ (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coarse</td>
<td>0.9330</td>
<td>2.7782</td>
</tr>
<tr>
<td>Medium</td>
<td>1.3200</td>
<td>2.7544</td>
</tr>
<tr>
<td>Fine</td>
<td>1.8670</td>
<td>2.7356</td>
</tr>
</tbody>
</table>
It can be observed from the Fig. 17, that $R_{T-SWATH}$ does not vary significantly with grid refinement and exhibits a sound convergence. The Medium grid has been used for the present work based on a trade-off between resolution and resource demand.

Uncertainty analysis has been performed according to ITTC 7.5-03-01-01 recommendations [18], using the method proposed by Tao and Stern [19]. Starting from the absolute differences among resistance values for Medium and Coarse meshes $\varepsilon_{M-C}$ and Fine and Medium meshes $\varepsilon_{F-M}$: 0.0238 and 0.0188 respectively, the calculation procedure is reported in Table 9.

### Table 9 Uncertainty analysis assessment

<table>
<thead>
<tr>
<th>$R_u = \frac{\varepsilon_{F-M}}{\varepsilon_{M-C}}$</th>
<th>$\ln \frac{\varepsilon_{F-M}}{\varepsilon_{M-C}}$</th>
<th>$P = \frac{p_G}{\ln \sqrt{2}}$</th>
<th>$\sigma^2 = \frac{\varepsilon_{F-M}}{\sqrt{2} \cdot p_G} - 1$</th>
<th>$U_{GC}(%) = 1.25 \cdot \sigma^2$</th>
<th>$U_{GC}(%) = 100 \cdot \frac{U_{GC}}{R_{T-Fine}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.790</td>
<td>0.6805</td>
<td>0.34023</td>
<td>0.0072</td>
<td>0.009011</td>
<td>3.230</td>
</tr>
</tbody>
</table>

### 4. COMPARISON WITH EXPERIMENTAL DATA

CFD analysis reported in this paper follows the experimental campaign concerning SWATH configurations performed at Hydrodynamic Lab of University of Naples Federico II. CFD simulations are compared with the results of the calm water tests performed on this scale model and already presented in [10].

#### 4.1 Resistance set up

Resistance set up for SWATH model is shown in Fig. 18. The towing line has been connected to a load cell HBM with maximum load of 5 kg. The towing line height was the lowest possible that means at waterline with the smallest possible angle to keep the towing line out of the water. Due to the very small value of model resistance, even at the lowest possible acceleration of the towing carriage, the model, free to surge, tended to oscillate longitudinally. Therefore all tests are performed with a counter pull force, (on the left in Figure 18) connected to the model, fixed at the same point of the towing line with opposite direction. A 500 g counter pull made motion steady.

Resistance tests have been performed with trim restrain acting soon after trim angles are recorded. This in order to appreciate the maximum possible free ride speed and then to evaluate the required action of trim correctors. Experimental setup is very close to restrained model numerical simulation set up.
The turbulence of the flow around torpedos has not been stimulated, as the Reynolds number $Re$ value was over $9 \times 10^5$ at the lowest tested speed. The struts at lowest speed values present lower $Re$, but the presence of torpedo heads has been considered an adequate factor for turbulence flow stimulation. The resistance tests are performed for eight model speeds. Tested velocities, measured resistance and measured standard deviations are reported in Table 10, where the values in bold announce the direct comparison with CFD simulation.

The ITTC Seakeeping Committee (2011) in recommended procedures 7.5 adopted ISO-GUM approach to conducting uncertainty analysis of experimental results. The ISO-GUM recognizes two groups of uncertainties, type A and type B based on way in which the uncertainty is evaluated. Type A represents the random category of uncertainty evaluated by using statistical analysis of repeated measurements of the same observation. The results reported in Table 10 are obtained as mean value multiplied bias of load cell (1/1000 N). Type B uncertainty can be estimated from quoted values of uncertainty, assuming statistical distribution of the parameters and factors depending on a level of confidence in the measurement. Generally, type B uncertainty is considered as normally distributed around some mean and for 95% level of confidence, factor 1.96 has to be applied to the standard deviations. Finally it can be seen that total uncertainty $u$, except for the speed 0.651 m/s, is ranging from 3 to 4 % and about the same as the numerical ones.

<table>
<thead>
<tr>
<th>$V$ (m/s)</th>
<th>$\sigma_V$</th>
<th>$Fr_T$</th>
<th>$R_T$</th>
<th>$\sigma_{R_T}$</th>
<th>Type A (N)</th>
<th>Type B (N)</th>
<th>$u$ (N)</th>
<th>$u$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.976</td>
<td>0.002</td>
<td>0.316</td>
<td>2.609</td>
<td>0.059</td>
<td>0.00261</td>
<td>0.115</td>
<td>0.115</td>
<td>4.42</td>
</tr>
<tr>
<td>1.301</td>
<td>0.002</td>
<td>0.422</td>
<td>2.883</td>
<td>0.059</td>
<td>0.00288</td>
<td>0.115</td>
<td>0.115</td>
<td>4.00</td>
</tr>
<tr>
<td>1.627</td>
<td>0.002</td>
<td>0.528</td>
<td>4.276</td>
<td>0.059</td>
<td>0.00428</td>
<td>0.115</td>
<td>0.115</td>
<td>2.70</td>
</tr>
<tr>
<td>1.952</td>
<td>0.002</td>
<td>0.633</td>
<td>5.943</td>
<td>0.088</td>
<td>0.00594</td>
<td>0.173</td>
<td>0.173</td>
<td>2.91</td>
</tr>
<tr>
<td>2.277</td>
<td>0.002</td>
<td>0.738</td>
<td>7.992</td>
<td>0.157</td>
<td>0.00799</td>
<td>0.308</td>
<td>0.308</td>
<td>3.85</td>
</tr>
<tr>
<td>2.603</td>
<td>0.003</td>
<td>0.844</td>
<td>10.091</td>
<td>0.177</td>
<td>0.01009</td>
<td>0.346</td>
<td>0.346</td>
<td>3.43</td>
</tr>
</tbody>
</table>
The measured total resistance was further subdivided into frictional and residual components according to ITTC 57 procedure. Within standard ITTC correlation procedure the measured resistances are reported in nondimensional form through “geosim” coefficients defined as

\[ C_T = \frac{R_T}{0.5 \cdot \rho \cdot v^2 \cdot WS} \]

where

- \( v\) – model velocity, m/s
- \( WS\) – model wet surface, m²
- \( \rho\) – water density, kg/m³

ITTC-57 correlation procedure defines frictional coefficient of a corresponding plate \( C_{F_0}\) as

\[ C_{F_0} = \frac{0.075}{(\log Re - 2)^2} \]

\( Re\) – Reynolds number defined as \( Re = \frac{v \cdot L}{u}\)

\( u\) - kinematic viscosity, m²/s

Due to the different strut and torpedo lengths, their \( Rn\) and \( C_{F_0}\) will be different and therefore the resistance breakdown is done as follows:

\[ R_{F-\text{STRUT}} = 4 \cdot (0.5 \cdot \rho \cdot g \cdot C_{F_0-\text{STRUT}} \cdot WS_{\text{STRUT}}) \]
\[ R_{F-\text{TORPEDO}} = 2 \cdot (0.5 \cdot \rho \cdot g \cdot C_{F_0-\text{TORPEDO}} \cdot WS_{\text{TORPEDO}}) \]
\[ R_{R-\text{SWATH}} = R_{F-\text{SWATH}} - R_{F-\text{TORPEDO}} - R_{F-\text{STRUT}} \]
\[ C_{R-\text{SWATH}} = \frac{R_{R-\text{SWATH}}}{0.5 \cdot \rho \cdot v^2 \cdot (2 \cdot WS_{\text{TORPEDO}} + 4 \cdot WS_{\text{STRUT}})} \]

For the calculation of residual coefficient shown in Fig. 19, the static wetted surface and ITTC 57 correlation procedure have been used. Results shown in Fig. 19 confirm main findings reported before on the necessity of stabilisation device after, in this case, \( F_{RT} 0.422\). The difference between \( C_R\) and \( C_W\) curves at lower speeds is only due to “form factor”, or more correctly, due to the intrinsic difference in the consideration of frictional resistance component by CFD simulation and experimental results based on ITTC 57 frictional line. Beside that the variation of dynamic wetted surface should be taken into account to get fair analysis of experimental resistance components. Nevertheless the obtained results for resistance coefficients can be considered very promising and future experimental campaign will be aimed at form factor and dynamic wetted surface determination.
5. CONCLUSIONS

Aim of this paper is to report the introduction of numerical simulation within a research program focused on small size SWATH vessels and to identify the reliability and potential of CFD analysis within design and development of SWATH hull forms.

The reported results concern calm water resistance analysis and hull efficiency coefficients. They are both of main importance in the preliminary design procedure for a sound assessment of powering performances. The comparison with experimental data reported in this paper allows to consider CFD as a reliable tool for the assessment of further developed similar hull forms.

The numerical investigation identifies the maximum possible free ride speed value without longitudinal instability. CFD results at higher speeds allow to identify and to assess input data for trim correctors and stabilizing fins.

The confirmation of expected very high hull efficiency allows reliable shaft horsepower prediction and highlights one of the peculiar favourable SWATH characteristics.

The further development of this research concerns the improvement of critical aspects in experimental resistance assessment as well as the study on effect of torpedo’s immersion on suction effect both numerically and experimentally; the numerical simulation of rough water SWATH behaviour and the comparison of seakeeping assessments in head and following seas.

ACKNOWLEDGMENTS

This work has been supported by Sommella & Contardi MARINE S.r.l within research commitment to Department of Industrial Engineering, University of Naples Federico II. The Authors are grateful to Mr. Edoardo Contardi and Mr. Carmine Sommella for their encouragements and collaboration. The Authors gratefully acknowledge the availability of 32 processors at Calculation Centre SCoPE, University of Naples, and thanks SCoPE academic staff for the given support.
NOMENCLATURE

- $B$ – maximum breadth (m)
- $CG$ – centre of gravity (m)
- $D$ – propeller diameter (m)
- $DT$ – torpedo diameter (m)
- $DT/T$ – ratio between torpedo diameter and SWATH draft
- $Fr$ – Froude number based on ship length, $Fr = \frac{v}{\sqrt{g \cdot L}}$
- $Fr_T$ – Froude number based on torpedo’s length
- $J$ – advance ratio $J = \frac{v \cdot \lambda}{n \cdot D}$
- $K_Q$ – non-dimensional torque $K_Q = \frac{Q}{\rho \cdot n^2 \cdot D^5}$
- $K_T$ – non-dimensional thrust $K_T = \frac{T}{\rho \cdot n^2 \cdot D^5}$
- $LCG$ – longitudinal position of centre of the gravity (m)
- $LOA$ – length over all (m)
- $L_S$ – strut length (m)
- $L_T$ – torpedo length (m)
- $n$ – propeller rotation rate
- $Q$ – torque (Nm)
- $Re$ – Reynolds number $Re = \frac{v \cdot L}{\nu}$
- $R_{F,SWATH}$ – frictional resistance in calm sea condition (N)
- $R_{T,SWATH}$ – total resistance in calm sea condition (N)
- $t$ – thrust (N)
- $t$ – thrust coefficient
- $v$ – ship, model speed (m/s)
- $w$ – wake coefficient
- $\Delta$ – displacement (N)
- $\lambda$- ship – model scale ratio
- $\eta_0$ – propeller efficiency
- $\eta_H$ – hull efficiency

REFERENCES


Submitted: 06/07/2015

Accepted: 22.09.2015.

Ermina Begovic
Carlo Bertorello
Simone Mancini
University of Naples Federico II
Department of Industrial Engineering
Via Claudio 21, 80125 Naples, Italy

22
A COMPUTATIONAL HYDRODYNAMIC ANALYSIS OF DUISBURG TEST CASE WITH FREE SURFACE AND PROPELLER

Summary

This paper discusses the effects of the free surface and the propeller on a benchmark Post-Panamax Ship, Duisburg Test Case (DTC). The experimental results are already available in the literature. The computational study carried out in this work is verified first with the experiments and then used to explain some of the physical aspects associated with viscous ship flows. There are two interesting outcomes of this work. The first one is, the existence of the propeller contributes to the pressure resistance of the ship by increasing the wave elevations along the hull and the fluid domain substantially. The second outcome is; by changing the pressure distribution along the hull and the propeller, the free surface increases the efficiency of the propulsion system. These specific outcomes are thoroughly discussed in the paper with CFD generated results and physical explanations.

Key words: propulsion efficiency; ship resistance; effect of propeller; effect of free surface; Duisburg Test Case;

1. Introduction

Flow around a bare hull with calm free water surface is one of the fundamental studies in naval architecture. The computational problem is around a century old which started with the foundation of potential theory at the start of 1900s. After maturing itself in time, potential theory also allowed the inclusion of free surface into the problem. With the developments in computer science, RANSE solvers were started to be used widespread which allowed solving for the viscous flows around ships. As the computer capabilities were extended, appendages were also included in the solutions.

Computational flow around a ship is one of the most challenging problems for the computational fluid dynamics (CFD) world. Ships are big structures that have complex geometries. Problems start from the beginning: it is really hard to have a good grid structure around the ship hull. The addition of an even more complex geometry, which is the propeller, complicates the grid problem.

The solution process involves two phases: air and water. The existence of free surface is another problematic issue for the ship hydrodynamicists. The rotating propeller at the aft of the ship needs special care due to its dynamics. Cavitation is a phenomenon that must be taken into
account because it has effect on the propulsion efficiency of the ship. Due to all these problems, naval architects who are working on ship hydrodynamics are usually more concentrated in solving this complicated flow and improving the obtained results. The underlying physics is generally not investigated by the CFD users. However CFD application to real ship flows has matured itself in time. Some results can be found in the open literature and a wide range of CFD results with various methods are given in Larsson et al. [1] for benchmark ships, KRISO Tanker (KVLCC2), KRISO Containership (KCS) and US Navy Combatant (DTMB 5415). Experiments usually reveal the physics behind a surface-piercing and propelling ship. However when used correctly, CFD may provide an invaluable tool to understand the flow around ships. CFD has matured itself in the last couple of decades and it is believed that it is possible to investigate the hydrodynamics of a ship computationally.

This paper discusses the effects of free surface and propeller on the hydrodynamics of a benchmark ship computationally. The ship used in this study is the Duisburg Test Case (DTC) which is a Post-Panamax Container Ship. The experimental results were published in [2]. The CFD generated results are first verified and then used to evaluate the computational outcomes. The free surface and the propeller are included in the solutions but the possible effects of cavitation are neglected.

2. Details of CFD approach

2.1 CFD setup

All the computational analyses are steady state solutions in this paper and $k - \varepsilon$ turbulence model is used with a commercial CFD code, ANSYS Fluent. This turbulence model is applicable when there are not high adverse pressure gradients along the hull. It is not as good as $k - \omega$ turbulence model when capturing the separation points. However $k - \omega$ needs lower $y^+$ values along the hull to demonstrate its abilities and demands more computational power. Considering that the length-to-beam ratio of the benchmark vessel is $L/B = 7$, the vessel is considered to be slender and boundary layer separation along the hull is not playing an important role in the flow. Details of $k - \varepsilon$ turbulence model are given in [3].

To track the free surface boundary, volume of fluid (VOF) method is used. Rhee et al. [4] discusses that VOF “performs well for a wide range of free-surface wave types” and is a better option when compared to other available methods. The theory of VOF is given in [5].

2.2 CFD verification and validation

Three grid types were used to do CFD verification as established in [6, 7]. The total resistance coefficient of the bare hull was taken at $F_r = 0.218$ as the integral variable of the verification. The element numbers in these grids are deviated according to the Richardson extrapolation technique following the methodology of [6]. The properties and the generated results for different grid types are provided in table 1. The total numerical uncertainty $U_N$ is given as

$$U_N = \sqrt{U_I^2 + U_G^2}$$

(1)

Here, $U_I$ and $U_G$ refer to iterative and grid uncertainties. All three cases have achieved oscillatory iterative convergence; therefore, the iterative uncertainty $U_I$ was calculated as

$$U_I = \frac{1}{2} |S_U - S_L|$$

(2)
In table 1, $S_U$ and $S_L$ refer to the upper and lower values of total resistance in the simulations respectively. $S_G$ is calculated as

$$S_G = \frac{1}{2} |S_U + S_L|$$

(3)

Table 1. Grid properties and their uncertainties.

<table>
<thead>
<tr>
<th>Element no.</th>
<th>GRID 3</th>
<th>GRID 2</th>
<th>GRID 1</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>447,000</td>
<td>1,286,000</td>
<td>3,468,000</td>
</tr>
<tr>
<td>$S_U*10^3$</td>
<td>5.236</td>
<td>4.449</td>
<td>4.409</td>
</tr>
<tr>
<td>$S_L*10^3$</td>
<td>5.057</td>
<td>4.394</td>
<td>4.407</td>
</tr>
<tr>
<td>$S_G*10^3$</td>
<td>5.1465</td>
<td>4.4215</td>
<td>4.408</td>
</tr>
<tr>
<td>$U_I*10^3$</td>
<td>0.0895</td>
<td>0.0275</td>
<td>0.001</td>
</tr>
</tbody>
</table>

By taking into account these three grids, the grid uncertainty and grid correction factor as proposed in [6, 7] are calculated as $U_G = 0.0267 \cdot 10^{-3}$ and $C_G = 52.7$ respectively. The grid uncertainty is about 25 times greater than the iterative uncertainty of grid 1. Therefore, it can be said that $U_N \cong U_G$ and,

$$U_N \cong 0.0267 \cdot 10^{-3}$$

The grid correction factor $C_G$ is sufficiently less than or greater than 1, therefore results of grid 1 is used for the rest of this study without making any corrections.

The experimental result provided in [2] at $Fr = 0.218$ is $C_{Texp} = 3.67 \cdot 10^{-3}$ while in grid 1 we get $C_{Tgrid} = 4.408 \cdot 10^{-3}$, which tells that the error is $E = 0.738 \cdot 10^{-3}$. For validation purposes, the error $E$ has to be smaller than the validation uncertainty $U_V$. Validation uncertainty is given as

$$U_V = \sqrt{U_N^2 + U_E^2}$$

(4)

$U_E$ refers to the experimental uncertainty. In the reference paper where the experimental results are published [2], the experimental uncertainty is not provided. However, it must be noted that the numerical uncertainty is very small as compared to the error. To satisfy $|E| < |U_V|$, the experimental uncertainty should be of the order of the error. Here there are two options. Either the experimental values provided in the reference paper have a large experimental uncertainty, or there is a bias of resistance forces calculated by the RANSE solver. To understand which one holds true, grid 1 is tested with other Froude numbers as well. This is covered in the next section and figure 4 plots the experimental and numerical total resistance curves. It is found out that the numerical method deployed in this study has a bias of results with the experimental study. The reason of this difference in the results might be accounted to the fixed body condition applied numerically while in the experiments the hull was free to sink and trim. In 1994, an 8% increase in total resistance coefficient for the fixed body condition for Series 60 was reported in [8]. Toda et al. [9] used fixed body condition for resistance computations around the Series 60 and calibrated their values by reducing 8% of the total resistance. It is possible that the same phenomenon also applies here which would let the resistance values get closer to the experimental results, if such calibration was performed.
2.3 Hydrostatics of the benchmark ship used in calculations

The Duisburg Test Case (DTC), which is a Post-Panamax Container Ship, is used in all the analyses involved in this paper. The experiments are carried out in the model basin SVA Potsdam and the results are given in [2]. The hydrostatic properties of the model ship are given in table 2. The domain size in all the analyses are chosen to meet the minimum requirements recommended by the ITTC [10] and are also listed in the same table.

Table 2. Hydrostatic properties of DTC [2] and the selected computational domain size.

<table>
<thead>
<tr>
<th>Property</th>
<th>Symbol</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length between perpendiculars</td>
<td>$L_{pp}$</td>
<td>m</td>
<td>5.976</td>
</tr>
<tr>
<td>Waterline breadth</td>
<td>$B_{wl}$</td>
<td>m</td>
<td>0.859</td>
</tr>
<tr>
<td>Draft</td>
<td>$T$</td>
<td>m</td>
<td>0.244</td>
</tr>
<tr>
<td>Displacement</td>
<td>$V$</td>
<td>m$^3$</td>
<td>0.827</td>
</tr>
<tr>
<td>Block coefficient</td>
<td>$C_B$</td>
<td>-</td>
<td>0.661</td>
</tr>
<tr>
<td>Wetted area</td>
<td>$S_W$</td>
<td>m$^2$</td>
<td>6.243</td>
</tr>
<tr>
<td>Design speed</td>
<td>$v_d$</td>
<td>knots</td>
<td>3.244</td>
</tr>
<tr>
<td>Computational Domain Length</td>
<td></td>
<td></td>
<td>6$L_{wl}$</td>
</tr>
<tr>
<td>Computational Domain Width</td>
<td></td>
<td></td>
<td>1.5$L_{wl}$</td>
</tr>
<tr>
<td>Computational Domain Depth</td>
<td></td>
<td></td>
<td>$L_{wl}$</td>
</tr>
</tbody>
</table>

3. Comparison of experimental results versus CFD approach

3.1 Bare hull resistance tests

The analyses involving the ship hull in this paper are made two ways. One is the double body flow solution which only covers the underwater hull (and the propeller if it exists). In this case only water exists (single phase) and the free surface is absent. The other covers the region outside the water as well and has the free surface interface between the air and the water (multi-phase). Validations in this study are made for both analysis types. Comparisons with the double body flow solutions are made with the ITTC correlation formula given for the frictional resistance coefficients [11]. Multi-phase solutions (which include the free surface) are validated by the experimentally obtained total resistance values given in [2].

The double body flow solutions are compared with the ITTC Correlation Line dictated by the ITTC57. Only the underwater hull is modeled due to the calm water assumption of the double body flow. The grid around the hull is given in figure 1. The comparison of frictional resistance coefficient, $C_F$, is given in figure 2.

Due to the complexity of the hull hybrid meshing is preferred. In a surrounding block which covers the hull, tetrahedral elements are used for practicality. This may be seen in figure 3. The rest of the domain has hexahedral elements which improve acquiring a better free surface deformation [12]. The computational total resistance and total resistance coefficient comparisons in contrast with the experimental values are given in figure 4. In terms of the total resistance coefficients, greater agreement is achieved with the experimental results at higher $Fr$ numbers. Maki et al. [13] explain this with the nonlinear theory (RANS) being unable to correctly solve for the highly steeped waves which happen in lower $Fr$ numbers. Same phenomenon is also observed in [14] where the discrepancy in low $Fr$ numbers is more significant.
Fig. 1 – The grid around the stern part of the hull (left) and the bow part of the hull (b).

The total resistance coefficient values obtained by double body flow simulations (ignoring free surface) are also plotted in figure 4. It may be quickly noticed that the double body results are in very good accordance with the experimental results. However this is a fluke, because double body flow results do not include the wave resistance. Returning back at figure 2, it will be seen that CFD overpredicts the frictional resistance which compensates for the absence of wave resistance in double body flow simulations.

The obtained results for both the double body flow (which is a single phase analysis due to involving water only) and the multi-phase flow are found to be satisfactory. In both cases, CFD predicts higher resistances but the general shapes of the resistance curves are similar.

Fig. 2 – Comparison of frictional resistance coefficients obtained by ITTC57 and CFD.
A computational hydrodynamic analysis of Duisburg Test Case with free surface and propeller

Fig. 3 – The grid around the hull (a) and stern (b) and the bow (c) for multi-phase flows.

Fig. 4 – Comparison of total resistance and total resistance coefficient obtained by CFD and experiments [2].
3.2 Open-water propeller tests

The validation for resistance tests are explained above using two different analysis methods. However, due to the existence of propeller in some analyses in the following sections, a validation for solving the flow around the propeller is also needed. Open-water experimental results are used to verify the validity of the method used to treat the flow around the propellers. Figure 5 shows the mesh structure on the propeller surface used in the simulations. The diameter of the model propeller is 0.15m to be in accordance with the reference article [2]. The propeller is right-oriented and its parameters in detail are given in the same paper.

Propellers have complex geometries and therefore, once again, tetrahedral elements are preferred for easier meshing. The comparison of the computational results with the experiments is given in figure 6. It is found out that a very good agreement with the experimental results is obtained by CFD. For a detailed analysis on DTC propeller, see [15].

Fig. 5 – The grid on the propeller surface.  
Fig. 6 – Comparison of CFD and experiments in terms of propeller characteristics.

4. Effect of the propeller in ship flow

4.1 Wave cut with and without the propeller

The wave characteristics of the ship were calculated with and without the propeller. In the case of the existence of the propeller, the hull-propeller system was subjected to different propeller rotations calculated by tentative advance coefficients. The advance coefficients of the propeller and the resistance components of the hull-propeller system along with the generated thrust by the propeller are given in table 3.

Due to the rotating nature of the propeller, it creates an asymmetry in the flow around the ship. This may be examined by observing the free surface elevation over the hull. In figure 7, it may be seen that the wave pattern along the hull is not the same at both sides of the ship in the case with the propeller. The wave elevations are equal at both sides of the ship when the propeller is absent because there is nothing in the flow that creates asymmetry. It must be noted here that the advance coefficient of the propeller is \( J = 0.7 \), and the wave elevations are expected to be slightly different in other advance coefficients.
It may be observed from figure 7 that the existence of the propeller increases the wave height along the hull except the stern region. The propeller sucks up the flow to create thrust, decreasing the amount of water at the aft of the ship. Due to the accelerated movement of water at this region, the pressure decreases. When the amount of water is reduced and the pressure is decreased, air replaces the places evacuated by the water; lowering the wave height at this part of the hull. The sucked up water at the aft part of the ship is compensated by the additional water at the bow region. This is one of the reasons of higher wave elevation at the front parts of the ship. The other reason is the additional resistance effect of the existence of the propeller.

![Fig. 7 - Wave elevations on the ship hull with and without the propeller at Fr = 0.218. For the case with the propeller, Js = 0.7.](image)

This additional resistance can numerically be observed from table 3. The table is created by changing the advance coefficient of the propeller and calculating the resistance and thrust values they generate. Lower advance coefficients have higher rotation and when the advance coefficient is infinite, this means that there is practically no existing propeller. In the table it may be seen that while the frictional resistance sails around a fixed value, the pressure resistance consistently drops. This supports the idea that when the propeller speeds up the flow and decreases the pressure at the stern of the ship, the pressure resistance increases. Table 3 also explains why the wave elevations are higher for the case with the propeller because wave resistance is a part of the pressure resistance.

The ejected water from the propeller causes higher waves at the wake of the ship. Figure 8 represents the wave elevations at the upstream and the downstream with and without the existence of the propeller at y = 0. Although there is not a major change at the upstream of the ship hull (which is not expected), there is a significant increase in wave height at the downstream of the hull. It is expected that wave elevations are decreased when the propeller works at higher advance ratios or the submergence depth of the propeller is increased. The free surface contours with and without the propeller in the whole fluid domain are given in figure 9. The propeller changes the generated waves in the fluid domain considerably.
Table 3. Numerical values of resistance and thrust at different advance coefficients for $Fr = 0.218$.

<table>
<thead>
<tr>
<th>$J_s$</th>
<th>$R_p$ (N)</th>
<th>$R_t$ (N)</th>
<th>$T$ (N)</th>
<th>$C_p$</th>
<th>$C_t$</th>
<th>$C_l$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.3</td>
<td>27.433</td>
<td>26.146</td>
<td>53.58</td>
<td>263.98</td>
<td>3.16E-03</td>
<td>3.01E-03</td>
</tr>
<tr>
<td>0.4</td>
<td>22.079</td>
<td>26.081</td>
<td>48.16</td>
<td>136.01</td>
<td>2.55E-03</td>
<td>3.01E-03</td>
</tr>
<tr>
<td>0.5</td>
<td>19.11</td>
<td>26.009</td>
<td>45.12</td>
<td>78.8</td>
<td>2.20E-03</td>
<td>3.00E-03</td>
</tr>
<tr>
<td>0.6</td>
<td>17.435</td>
<td>25.968</td>
<td>43.4</td>
<td>48.69</td>
<td>2.01E-03</td>
<td>2.99E-03</td>
</tr>
<tr>
<td>0.7</td>
<td>16.233</td>
<td>25.934</td>
<td>42.17</td>
<td>31.25</td>
<td>1.87E-03</td>
<td>2.99E-03</td>
</tr>
<tr>
<td>0.8</td>
<td>15.151</td>
<td>25.904</td>
<td>41.06</td>
<td>20.49</td>
<td>1.75E-03</td>
<td>2.99E-03</td>
</tr>
<tr>
<td>0.9</td>
<td>14.761</td>
<td>25.9</td>
<td>40.66</td>
<td>13.43</td>
<td>1.70E-03</td>
<td>2.99E-03</td>
</tr>
<tr>
<td>1</td>
<td>14.37</td>
<td>25.908</td>
<td>40.28</td>
<td>8.55</td>
<td>1.66E-03</td>
<td>2.99E-03</td>
</tr>
<tr>
<td>1.1</td>
<td>14.022</td>
<td>25.903</td>
<td>39.92</td>
<td>5.08</td>
<td>1.62E-03</td>
<td>2.99E-03</td>
</tr>
<tr>
<td>1.2</td>
<td>13.558</td>
<td>25.892</td>
<td>39.45</td>
<td>2.46</td>
<td>1.56E-03</td>
<td>2.99E-03</td>
</tr>
<tr>
<td>infinity</td>
<td>10.584</td>
<td>26.058</td>
<td>36.84</td>
<td>0</td>
<td>1.22E-03</td>
<td>3.00E-03</td>
</tr>
</tbody>
</table>

Fig. 8 – Wave cut at $y = 0$ of the fluid domain for $Fr = 0.218$. 
4.2 Changes in hull pressure due to the existence of the propeller

Possible pressure changes at the stern of the hull are investigated in this section with the existence of the propeller. It is expected that the propeller changes the pressure distribution of the hull because it changes the flow characteristics, especially at the stern of the ship. The computational work carried out in this section has double body condition. The free surface effects are neglected.

Propeller speeds up the flow at the aft of the ship, creating a negative pressure field at the stern. In figure 10, pressure coefficient along the hull is given. From that figure, it may be observed that the propeller changes the pressure distribution along the hull, especially at the stern.

The propeller pushes water away by rotating to propel the ship. This rotation speeds up the flow nearby and the stern of the hull is affected by the increased velocity/decreased pressure. The pressure distribution at the left in figure 10 is an end result of the existence of the propeller. This is also in accordance with the decline of wave elevation shown in figure 7. Propeller sucks up the flow to push the vessel. This, in return, reduces the water nearby and air fills in the places evacuated by water, lowering the wave height at the stern of the ship.
5. **Effect of the free surface in ship flow**

5.1 Effect of free surface on propeller performance

The free surface effect on the propeller is examined including the hull and therefore its interaction with the propeller. The propeller rotation and therefore the advance coefficient is the same for both cases. The double body flow solutions with the multi-phase results are compared in this section. The effects of cavitation are ignored and the existence of the hull for efficiency compared to open water tests is discussed.

The advance coefficients are calculated for a constant ship velocity and by changing the propeller rotation speed. The Froude number is fixed at $Fr = 0.218$ for the results of the CFD analyses in this section and the ship is not in a self-propulsion state.

Figure 12 reveals the comparison made between the efficiencies calculated for the cases behind the ship hull without the free surface and behind the ship hull with the free surface. It is of direct notice that the efficiency with the ship hull majorly increases and the advance coefficient range widens when figure 12 is compared with the open water propeller test results given in figure 6. To understand why such a difference in results exists, we should write down the equations for the advance coefficient and the efficiency:

**Advance coefficient:**

$$J_x = \frac{V_x}{n \cdot D}$$  \hspace{1cm} (5)

**Calculated efficiency:**

$$\eta = \frac{J_x \cdot K_T}{2\pi K_Q}$$ \hspace{1cm} (6)

Here, $n$ is the propeller rotation per second and $D$ is the propeller diameter. $K_T$ is the thrust coefficient while $K_Q$ is the torque coefficient. $J_x$ and $V_x$ refer to advance coefficient and speed of advance respectively. The subscript $x$ may stand for $A$ or $S$. So $V_A$ in that case, is the flow velocity received by the propeller while $V_S$ is the ship velocity. $J_A$ and $J_S$ take into account velocities $V_A$ and $V_S$ respectively. It should be noted that figure 6 is plotted against $J_A$ which is
calculated by $V_A$ and figure 12 uses $J_S$ to plot the propeller performance with respect to $V_S$. The relation between $V_A$ and $V_S$ is:

$$V_A = V_S(1 - w)$$  \hspace{1cm} (7)

So the relation between $J_A$ and $J_S$ is:

$$J_A = J_S(1 - w)$$  \hspace{1cm} (8)

Wake fraction $w$ is always greater than zero. Therefore $J_S$ is greater than $J_A$ which reflects the reason of wider advance coefficient range given in figure 12.

![Figure 12](image)

Fig. 12 – Propeller performance with and without the free surface.

The existence of the free surface increases the efficiency of the propeller at higher advance coefficients. This may be examined from figure 12 as there is a clear increasing trend at the efficiency of the propeller after $J = 0.7$. The swirl created by the propeller’s rotation is limited by an upper barrier (which is the free surface in this case) decreasing the torque. The tangential velocities are decreased and the axial velocities are increased. Refer to figure 13 for an effective propeller wake with and without the free surface effect.
Fig. 13 – Total velocity distribution at the propeller disc with (left) and without (right) the free surface. $J_S = 0.7$, $Fr = 0.218$.

Figure 13 is a proof of the increase in axial velocity when the free surface is present. The free surface behaves like wall in this case and the flow is squeezed between the free surface and the propeller; causing a jet like flow. The increase in thrust and the decrease in torque are very low and hard to perceive from figure 12. The numerical values of $K_T$, $K_Q$ and $\eta$ are provided in table 4 so that a greater understanding could be achieved. Here; $K_T$, $K_Q$ and $\eta$ are propeller characteristics without the free surface while $K_T'$, $K_Q'$ and $\eta'$ are the values obtained with an existing free surface. These small differences reflect on the efficiency of the propeller, especially at higher advance coefficients.

Table 4. Numerical values of $K_T$, $K_Q$ and $\eta$ given in figure 12.

<table>
<thead>
<tr>
<th>$J_S$</th>
<th>$K_T$</th>
<th>$K_T'$</th>
<th>$10^*K_Q$</th>
<th>$10^*K_Q'$</th>
<th>$h$</th>
<th>$h'$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.3</td>
<td>0.3817</td>
<td>0.3802</td>
<td>0.5706</td>
<td>0.5676</td>
<td>0.3194</td>
<td>0.3198</td>
</tr>
<tr>
<td>0.4</td>
<td>0.3503</td>
<td>0.3482</td>
<td>0.5321</td>
<td>0.527</td>
<td>0.4191</td>
<td>0.4207</td>
</tr>
<tr>
<td>0.5</td>
<td>0.3172</td>
<td>0.3152</td>
<td>0.4914</td>
<td>0.4847</td>
<td>0.5137</td>
<td>0.5175</td>
</tr>
<tr>
<td>0.6</td>
<td>0.2826</td>
<td>0.2805</td>
<td>0.4487</td>
<td>0.4397</td>
<td>0.6016</td>
<td>0.6093</td>
</tr>
<tr>
<td>0.7</td>
<td>0.247</td>
<td>0.245</td>
<td>0.4043</td>
<td>0.3931</td>
<td>0.6805</td>
<td>0.6944</td>
</tr>
<tr>
<td>0.8</td>
<td>0.2107</td>
<td>0.2096</td>
<td>0.3591</td>
<td>0.3462</td>
<td>0.7471</td>
<td>0.7709</td>
</tr>
<tr>
<td>0.9</td>
<td>0.1756</td>
<td>0.1741</td>
<td>0.3152</td>
<td>0.2984</td>
<td>0.7982</td>
<td>0.8356</td>
</tr>
<tr>
<td>1</td>
<td>0.1367</td>
<td>0.1367</td>
<td>0.2654</td>
<td>0.2469</td>
<td>0.8199</td>
<td>0.8817</td>
</tr>
<tr>
<td>1.1</td>
<td>0.0975</td>
<td>0.0984</td>
<td>0.2144</td>
<td>0.1926</td>
<td>0.7961</td>
<td>0.8944</td>
</tr>
<tr>
<td>1.2</td>
<td>0.0584</td>
<td>0.0568</td>
<td>0.1599</td>
<td>0.1328</td>
<td>0.6743</td>
<td>0.8166</td>
</tr>
<tr>
<td>1.3</td>
<td>0.0107</td>
<td>0.009</td>
<td>0.0982</td>
<td>0.0632</td>
<td>0.2247</td>
<td>0.2937</td>
</tr>
</tbody>
</table>

The squeeze of the flow between the free surface and the propeller has to be explained further. If this hypothesis is in fact true, then it is expected that the hydrodynamic pressure on the propeller is higher. Figures 14 and 15 show the free surface effect on both sides of the propeller.

Fig. 14 – Effect of free surface on propeller pressure coefficient distribution at the downstream side. $J_S = 0.7$. Hydrostatic pressure is not included.
The free surface increases the pressure on the propeller as can be seen from figures 14 and 15.

Effects of the hydrostatic pressure are not included when calculating the pressure coefficient to understand how the free surface affects the hydrodynamic pressure. In these graphs pressure is non-dimensionalized by $V_S$. When the propeller is deeply submerged in water as it is represented by the double body flow solution, it is not affected by the atmospheric pressure. However, when there is a free surface nearby, it is oppressed by the atmospheric pressure. It was shown in the previous section that the propeller speeds up the flow to decrease the pressure and the wave height at the stern (please see figures 7 and 10). The decline of the wave height above the propeller will squeeze the flow between the propeller and the free surface to increase the pressure over the propeller. Figures 14 and 15 explain the increase of calculated efficiency given in figure 12.

Fig. 15 – Effect of free surface on propeller pressure coefficient distribution at the upstream side. $J_S = 0.7$. Hydrostatic pressure is not included.

Paik et al. [16] experimentally and computationally measured the pressure coefficient distributions of a propeller-ship hull system at ballast and design drafts. They have found out that, when the ship is in ballast draft, the pressure on the propeller is higher. The reason why the pressure on the propeller is lower at design draft is because the propeller is submerged deeper in this case. It is less affected by the free surface than when it is at ballast draft.

5.2 Changes in hull pressure due to the existence of the free surface

The pressure changes in hull with the existence of the free surface are investigated. The propeller is absent in these simulations and has no effect on flow characteristics. The observations are made for the design speed of the vessel.
Fig. 16 – Pressure coefficient distribution along the underwater hull with (right) and without (left) the free surface. Hydrostatic pressure is not included. \( Fr = 0.218 \).

Existence of free surface generally increases the total pressure at the stern of the hull. This may be examined from figure 16 where the effects of the hydrostatic pressure are not included in the total pressure. The reason for this is to accommodate the multiphase solutions with the double body solutions, where the hull is deeply submerged and has no hydrostatic pressure acting upon it. The two ships might seem to have different geometries but this is not the case. For the case without the free surface there is (naturally) no wave elevation while for the case with the free surface, the ship floats between wave crests and troughs. The wave crest at the transom can clearly be seen from the case with the free surface (please see figure 7 for the wave crest at the transom). The effect of the free surface on the hull is compatible with the free surface effect on the propeller. It may be said that the free surface creates an additional pressure over the bodies inside the flow.

6. Conclusion

In this study, hydrodynamics of a benchmark ship, Duisburg Test Case, is computationally investigated. The changes in ship resistance and the propulsion efficiency are observed with the existence of the free surface and the propeller. The computational results are first verified by a widely used methodology within the field and then the generated results are demonstrated. The paper covers the effect of two different changes in the flow. The first one is the existence of the propeller and the second one is the existence of the free surface.

In the first part of the paper it is found out that the existence of the propeller
- lowers the wave elevation at the stern of the hull while increasing it at the bow,
- causes the ship to generate higher waves close to the ship hull and changes the wake field considerably,
- increases the pressure resistance of the ship,
- decreases the pressure at the hull stern and
- does not have a significant effect on vorticity generation along the hull.

The second part covers the effect of the free surface. It is found out that the free surface
- increases the efficiency of the propeller especially at higher advance coefficients,
- increases the pressure on the propeller and
- increases the pressure at hull stern.

Acknowledgments

Authors are grateful to Prof. Fahri Celik from Yildiz Technical University for his valuable recommendations. We also would like to thank the reviewers for their constructive comments and suggestions to improve this paper.
REFERENCES


A REAL TIME DECISION SUPPORT SYSTEM FOR THE ADJUSTMENT OF SAILBOAT RIGGING

UDC 629.5.525.4:629.5.072
Original scientific paper

Summary

The operational complexity and performance requirements of modern racing yachts demand the use of advanced applications, such as a decision support system (DSS) able to assist crew members during navigation. In this article, the authors describe a near-time computational solver as the main piece of a DSS which analyses and monitors the behaviour of sails and rigging. The solver is made up of two different interconnected tools: an iterative Fluid-Structure Interaction algorithm and an advanced Wireless Sensor Network to monitor rigging. The real-time DSS quantifies crew manoeuvres in physical terms, which are reproduced by a simulation program. It can be used in the design phase of sailing yachts and as an aid for real-time boat performance optimisation and accident prevention. This novel DSS is a useful tool for navigation, especially in races.

Key words: racing yacht; trim parameters; decision support system; rigging monitoring;

1. Introduction

The enormous improvements made in yacht design in the last decades have resulted in significantly enhanced performance of these vessels. However, growing competitiveness among designers has increased the demand for detailed experimental and computational research to better understand the behaviour of racing yachts and optimise their design. Moreover, the operational complexity and performance requirements of modern racing yachts require the use of advanced applications.

Some applications are used for optimal selection of a mast and standing rigging [1] whereas others focus on the development of expert decision support systems for ship design [2]. Other contributions have addressed computer-aided design of ship systems automation [3, 4, 5 and 6]. Two more examples of applications are a decision support system (DSS) for vessel fleet scheduling [7] and a knowledge-based DSS for shipboard damage control [8].
The above works have helped build the background to ship design, building and management requirements so that research on DSS applied to ship security and control is increased. However, none of these studies are focused on any specific type of ships, e.g. racing yachts.

Sailing is the art of controlling a boat with large foils called sails. A sailor manages the force of the wind on the sails by adjusting the rigging in order to control the direction and speed of the boat. Sails are designed to be able to take the optimal shape for all sailing conditions. To obtain the best sail shape, the crew adjust the traveller position and sail twist and camber. The fluid-structure interaction of sails and rigging is related to these manoeuvres.

The performance of the sail/rigging configuration can be analysed by two aerodynamic parameters, i.e. lift coefficient and drag coefficient. [9] investigated the relation between changes in sail loads and in trim. The results showed that sail turn (change in traveller position) has an effect on lift and drag whereas camber has no influence on the former and only a slight influence on the latter. In order to study the behaviour of sails and rigging during navigation, it is necessary to consider all manoeuvrability variables. No software for operational computation of structure response to wind conditions considering variations in trim parameters introduced by the crew is currently available. One of the motivations of this work was to develop a novel operational navigation tool using variations in trim parameters to support crew decision making.

Structural failures of racing yachts are not infrequent. Just as an example, the Groupama team broke the mast during the last Volvo Ocean race. A real-time monitoring and structure analysis tool could help prevent damage to the structure and injuries to the crew. The automation of emergency management operations is today driven by complex technology architectures called damage control system (DSC) [10]. A DCS is an information-retrieval and equipment-control system that gives ship personnel the ability to detect, analyse and handle various types of damage situations based on the collection and processing of vast quantities of shipboard information. However, no DCS addressed specifically designed for sailing yachts has yet been developed.

The main goal of this article is to present an operational tool that can be the main piece of a decision support system (DSS) to assist the sailing yacht crew during navigation, as well as to prevent accidents, just as a DCS does.

Our work focuses on the development of a program for monitoring/simulation of the behaviour of upwind sails and rigging which helps the crew optimise real-time yacht performance. Rigging is monitored to obtain the boat structure trim introduced by the crew, and the aerodynamic coefficients and structure response to wind conditions and trim, which is valuable information for the crew, are computed.

The monitoring/simulation program requires the development of a solver composed of two interrelated tools:
- A Wireless Sensor Network (WSN) to monitor rigging and sails and capture trim.
- A Fluid-Structure Interaction (FSI) algorithm to compute the performance of sail/rigging configurations. It integrates two solvers: (1) a fluid dynamics solver based on the Boundary Element Method (BEM) [11] to calculate the aerodynamic forces for a given sail shape in upwind conditions; (2) a Finite Element Analysis (FEA) solver [12 and 13] of the structural behaviour of rigging and sails which considers aerodynamic forces and shroud and stay tension, as well as sheet tension.
The WSN quantifies crew manoeuvres in physical terms. These data is used as boundary conditions for the FSI tool. The model geometry is adapted according to the trim parameters, and then the performance of the new configuration is computed in real time.

Because of difficulties in dynamically measuring boundary conditions with precision and reproducing a manoeuvre in real time, only stationary conditions are analysed. This is an acceptable simplification because it can be assumed that, at the end of a manoeuvre, the sailing yacht operates in a quasi-stationary regime.

The resulting solver, named Sailing, provides real-time information (aerodynamic coefficients) about the trim and rigid structure to avoid problems such as breakage of the mast. Real-time monitoring is an innovation in navigation, especially in races.

Two of the most important requirements for Sailing are reduced computational cost and ability to adapt the analysis to the real position of sails. They are the main guidelines to select the calculation algorithms.

The two interconnected tools that make up Sailing are presented further in this paper.

The most relevant and original aspects of this work are the communication between the sensors and the FSI algorithm, and the methods to adapt the structure to real-time trim parameters. The sensors capture the trim parameters and these are communicated to the FSI algorithm to adjust the boundary conditions for structural analysis. That is, there is real-time consideration of the manoeuvre parameters in the determination of fluid-structure interaction. Near-time analysis is then conducted so that the results are immediately available to assist skippers in their decisions.

Next, simulation data are used to adjust an Artificial Neural Network in order to know in advance the best trim angle for certain wind conditions.

The FSI solver is coded in C++ and then the resulting code is integrated with the pre/post-process system GiD\(^1\). The FSI code is extended with a TCL\(^2\) interface to provide the necessary communication routines with the sensor network.

The outline of this paper is as follows. In section 2, the FSI solver is presented. Section 3 focuses on the sensor network. The interface that connects the FSI program with the sensor network data is described in section 4. Section 5 gives some application examples. The paper ends with a discussion section and some conclusions.

2. The fluid structure interaction solver

Sails are air foils that work by using an airflow set up by the wind and the motion of the boat. The performance of a specific sail configuration is obtained by the coupled effect of the airflow around the sails and structure response to the generated forces.

Rigging adjustments to achieve the desired amount of camber and twist are interrelated. That is the reason why structural analysis of all rigging elements in a fluid-structure interaction algorithm is necessary for the study of sail performance.

As said before, the FSI simulation algorithm integrates two solvers: (1) a fluid dynamics solver to calculate the aerodynamic forces for a given sail shape in upwind conditions; (2) a Finite Element Analysis (FEA) solver of the structural behaviour of rigging and sails. The aerodynamic forces and shroud and stay tension, as well as sheet tension, are taken into account.

\(^1\) GiD: The universal, adaptive and user-friendly pre and postprocessor http://gid.cimne.upc.es/

\(^2\) TCL: Tool Command Language http://www.tcl.tk/
In order to study sail and rigging performance of a boat, the first necessary step is the definition of the flying sail shape to be analysed. In this work, the flying sail is obtained by the fluid-structure interaction solver, but the iterative solution process starts with the designed sail plan geometry. A Graphic User Interface (GUI) based on GiD is developed to create or import the structure (sail and rigging) and define analysis data.

Flow and structural analysis, validation cases and interaction algorithm are detailed below.

2.1 Flow Analysis

The first numerical simulation of sails was conducted at the Massachusetts Institute of Technology in the 60s, when [14] developed Vortex Lattice Method (VLM) and flat wakes to investigate upwind sails. Since potential flow approximation is a reasonable approach for predicting upwind sail performance, provided that separation is restricted to small areas within the vicinity of the mast or leading edge, VLM is one of the computational methods commonly used to predict flow over sails [15, 16]. This method is relatively fast and suited for computing flow over combinations of highly cambered, thin lifting surfaces like sails.

Provided that separation of the flow over the sail is restricted to small areas within the vicinity of the mast or leading edge, potential flow approximation is a reasonable approach. Thus, fluid flow must fulfill the Laplace equation as long as the boundary conditions are a zero normal velocity component and the Kutta condition at each trailing segment. The Laplace equation solution can be obtained by distributing elementary solutions over the problem boundaries (body surface $S_B$ and wake surface $S_w$). In our case of study, the sail surface carrying the continuous distribution of doublicity is discretised into a set of vortex ring elements, each one carrying a locally constant value of doublicity ($\Gamma$). The Biot-Savart law is used to evaluate the velocity at each collocation point by each ring element. The boundary condition of zero normal velocity at each of the collocation points results in a set of algebraic equations. The solution of this set of equations results in the vector $(\Gamma_1, \Gamma_2, \ldots, \Gamma_m)$, which represents the doublicity of each ring element.

A preconditioned Bi-Conjugate Gradient method (BiCG) is used to solve the system of linear equations. Once the circulation of each singularity element is known, it is possible to calculate total flow velocity, pressures and loads.

[17] presented an alternative interpretation and implementation of the Vortex Lattice Method studied by [15]. The difference between this method and a general one, such as that proposed by [11], lies in the calculation of pressures and forces. The velocity induced by the doublicity distribution at the collocation point is split into two components: one normal $\left( u_{n,ind} \right)$ to the surface (where $n$ indicates the normal direction) and the other $\left( u_{m,ind}, u_{l,ind} \right)$ in the tangent plane of the surface at the collocation point (where $m$ and $l$ indicate the two directions in the tangent plane of the surface). The free stream velocity, $Q_{\infty}$, is transformed into the coordinates of the vortex ring element $Q_{\infty} = \left( q_{m,\infty}, q_{l,\infty}, q_{n,\infty} \right)$.

In order to obtain the velocity at the collocation point on both sides of the sail, half of the velocity jump is added to the mean velocity to give the upper surface velocity, and half of the velocity jump is subtracted to give the lower surface velocity. On the other hand, the velocity jump along direction $s$, which is $\frac{\partial \Gamma}{\partial s}$, is obtained using a simple finite difference approximation.
The free stream and the velocity induced by the doublicity distribution are combined with the velocity jump to obtain the velocity at each collocation point on both sides of the sail.

\[
V_{\text{up}} = \left( q_{m,\infty} + u_{m,\text{ind}} + \frac{1}{2} \frac{\partial \Delta \Gamma}{\partial m}, q_{l,\infty} + u_{l,\text{ind}} + \frac{1}{2} \frac{\partial \Delta \Gamma}{\partial l} \right)
\]

\[
V_{\text{down}} = \left( q_{m,\infty} + u_{m,\text{ind}} - \frac{1}{2} \frac{\partial \Delta \Gamma}{\partial m}, q_{l,\infty} + u_{l,\text{ind}} - \frac{1}{2} \frac{\partial \Delta \Gamma}{\partial l} \right)
\]

In this way, the method computes pressure at each side of the sail:

\[
P_{\text{up}} = p_{\infty} - \frac{1}{2} \left( V_{\text{up}}^2 - Q_{\infty}^2 \right)
\]

\[
P_{\text{down}} = p_{\infty} - \frac{1}{2} \left( V_{\text{down}}^2 - Q_{\infty}^2 \right)
\]

\[
C_{\text{up}} = 1 - \left( \frac{V_{\text{up}}}{Q_{\infty}} \right)^2
\]

\[
C_{\text{down}} = 1 - \left( \frac{V_{\text{down}}}{Q_{\infty}} \right)^2
\]

The method was codified into the solver to calculate coefficients and forces generated by a flow around a thin lifting surface.

Some validation cases were made to test the flow analysis. An example was the reproduction of an experiment carried out in the wind tunnel of the Chunghnam National University of Korea [18]. The same geometry was tested using the flow analysis presented in this paper achieving an accuracy measurement of 90% for the lift and 75% for the drag.

2.2 Structural Analysis

The description of flow is supplemented by considering fluid-structure interaction. The particularity of sails is due to their high flexibility, and therefore a non-linear theory must be developed for the structural calculation [19, 20]. Rigging adjustments to achieve the desired amount of camber and twist are interrelated. That is why structural analysis of all rigging elements in a fluid-structure interaction algorithm is necessary for the study of sail performance.

Once the aerodynamic forces are evaluated, the next step is to compute the structure response to them. The method selected to analyse the structure and compute its response is the well-known Finite Element Method (FEM).

The rigging structure of a typical sailing yacht mainly consists of mainsail, jib, mast, spreaders, forestay, backstay, shrouds, main sheet and jib sheet. The different parts of the rigging and sails are modelled with membrane, truss and beam structural elements.

The solution algorithm is based on the minimisation of the potential energy. Potential energy \( E_{\text{TOT}} \) includes (a) the strain energy of elastic distortion \( \phi(x) \), which is calculated depending on the type of element (membrane, cables and beams), and (b) the potential of applied loads \( \Omega \), by which they have the capacity to do work in case of displacement of the structure. The structures are analysed after each configuration change made by the crew, but the
resulting configuration is considered in a stationary state. Therefore, the equilibrium configuration is defined by the principle of stationary potential energy.

A description of all the elements is provided in the following subsections.

2.2.1 Sails

The behaviour of sails requires large displacement analysis of very thin structures. The models used to study sails are geometrically nonlinear, and since strains in modern sails remain low, constitutive laws of the material can be considered as linear, and stresses on the structure as linear functions of local strains. It is assumed that sails can be accurately modelled as membranes. Three-node triangular elements are used to discretise sails. The calculation method was presented by [21].

One of the validation tests was the reproduction of the Hencky’s problem, which considers the deformation of an initially flat circular membrane with fixed edges, loaded with a constant pressure. The maximum deformation of the membrane reached 0.0330m in the centre part. This value almost matches the maximum deformation of 0.0331m obtained by Pauletti [22].

2.2.2 Cables

A sailboat’s rigging is composed of many ropes and cables, mainly main sheet, jib sheet, forestay, backstay and shrouds. These elements can be modelled using bar finite elements. They can only transmit axial forces, which mean that the nodes of bar elements only have translational degrees of freedom. Therefore, it is possible to discretise a cable with a set of articulated bar elements. The resulting truss will be a cable that only transmits traction forces.

The model implemented in this work was presented by [23]. It includes a total Lagrangian description using the standard strain definition and assumes an elastic material. The cables are discretised with two-node linear elements, where the strain is assumed constant along the element and the material is considered to be homogeneous and isotropic.

The cables used in our structure are pre-stressed, as a result of the stress applied to trim the sail. This action leads to a cable strain. By exerting a constant stress on a cable element and considering a linear stress strain relationship, the potential strain energy and its gradient can be easily computed.

One of the validation cases considered a cable with fixed ends loaded with its own weight. Initially the cable had a V-shape, and the expected deformation was to obtain a U-shape. The reactions at the ends should be equal to the weight of the cable.

The properties of the material considered were \( E=5.01\cdot10^6 \text{N/m}^2 \) and \( \rho=100 \text{kg/m}^2 \), the length of the cable was \( L=14.1421 \text{ m} \) and the area of the cable was \( A=0.0005 \text{ m}^2 \). Considering the density of the material and the dimensions of the cable, the weight of the cable was \( P=69.36 \text{N} \). The reactions in each end were \( 34.681 \text{N} \) and the total reaction was \( R=2\cdot34.681=69.372 \text{N} \), obtaining the expected results.

2.2.3 Beams

The other elements of the structure (mast, spreaders and boom) resist axial and transversal forces, and bending moments. Therefore, they are modelled using beam theory [12, 24]. The beams are modelled as a two-node beam element with six degrees of freedom per node. The mast and the boom are connected by a hinge to allow rotation of the boom around the mast. To introduce the capability of releasing a degree of freedom, the algorithm presented in [25] to obtain the condensed stiffness matrix is adapted to simulate the hinge between the mast and the boom.
One of the validation cases was the simulation of a vertical cantilever with a horizontal load. The problem had the following parameters: length \( L = 14 \text{m} \), \( A = 21.36 \text{cm}^2 \), \( I_x = 5.8 \times 10^{-6} \text{m}^4 \), \( I_y = 1.35 \times 10^{-5} \text{m}^4 \), \( J = 1.9 \times 10^{-5} \text{m}^4 \), \( G = 3.946 \times 10^{10} \text{N/m}^2 \) and \( E = 1.105 \times 10^{11} \text{N/m}^2 \).

The lowest node had restricted translations and rotations in all directions, and the top node had a punctual load of 500 N applied in the \( x \) direction.

It was possible to calculate the displacement of the top node with the following analytical expression:

\[
y_{\text{max}} = \frac{F \cdot L^3}{3EI_y} = \frac{500 \times 14^3}{3 \times 1.105 \times 10^{11} \times 1.35 \times 10^{-5}} = 30.7 \text{cm}
\]

(7)

The displacement of the top node obtained was 30.6 cm and the reaction was \( R = 500 \text{N} \), which are both expected results.

2.2.4 Equilibrium configuration

The solution procedure involves generating the structure model with the above elements, imposing boundary or support conditions and solving the equations to obtain nodal quantities by imposing the minimisation of the potential energy.

A conservative mechanical system has an energy potential, \( E_{\text{TOT}} \), which includes the strain energy of elastic distortion \( \phi(x) \) and the potential of applied loads \( \Omega \).

In the problem at hand, the airflow applies external loads on the structure, causing the displacement and strain of the elements. Given the nodal displacement vector \( \mathbf{x} \) and nodal flow forces \( \mathbf{f} \), the potential of applied loads is \( \Omega = -f^T \mathbf{x} \).

The strain energy \( \phi(x) \) is calculated according to the finite element analysed (membrane, truss or beam).

The total potential energy is

\[
E_{\text{TOT}} = \sum_{\text{elements}} \phi(x) - f^T \mathbf{x}
\]

(8)

And thus the equilibrium configuration is found from the stationary value of the total potential energy:

\[
\frac{\partial E_{\text{TOT}}}{\partial \mathbf{x}} = \sum_{\text{elements}} \nabla \phi(x) - \mathbf{f} = 0
\]

(9)

This non-linear system of equations is solved by a Quasi-Newton iterative procedure. The computer code uses the limited memory Broyden-Fletcher-Goldfarb-Shanno method (BFGS) to tackle large scale problems, as described by [26]. A line search procedure as described by [27] is implemented, too.

2.3 Fluid-Structure Interaction Algorithm

Complete modelling of sail steady equilibrium involves a fluid-structure interaction analysis. The presence of the sail modifies the flow, whilst the latter applies aerodynamic forces on the former, changing its geometry.

[28, 29, 30, 31] dealt with the numerical problem by coupling FEM and VLM solvers. [32] pointed out that one of the reasons for poor convergence of the FEM-VLM is the use of the same discretisation scheme for the FEM and VLM. That is why this work uses different discretisation schemes for these solvers.
The VLM selected to compute flow forces on the sails uses quadrilateral elements, even though these elements can create both structured and unstructured meshes. The membrane element selected to compute sail response is based on three-node triangular elements. In order to fulfill the requirements of both methods, a structured mesh of quadrilateral elements is generated and split internally by the program into triangular elements (Fig. 1). This procedure allows the two meshes to have coincident nodes facilitating data transfer between the two meshes.

![Elements for the VLM](image1.png) ![Elements for the FEM](image2.png)

Fig. 1 Fluid Solver mesh and Structural Solver mesh

This fluid-structure interaction method is iterative: starting from an initial geometry, the flow forces are calculated, from which sail displacements are then evaluated, and finally the flow simulation is repeated with the updated geometry and the structural calculation is in turn repeated with the updated flow forces until convergence is achieved. The grid is thus modified at each fluid-structure iteration. Lagrangian advance techniques are used to fit the sail grid surface from the deformed sail calculated in the previous iteration. Lagrangian updating of the grid is done automatically by the algorithm as a hidden procedure for users.

The first step of the calculation is to apply the trim parameters to the basic structure as initial loads and boundary conditions. In order to modify these parameters in execution time, a TCL interface is implemented. This interface includes several communication routines allowing solvers to be connected with a network of sensors located on the rigging to obtain the trim parameters in real time.

From the interaction method in [33], an embedded iteration scheme is used to achieve convergence of the iterative procedure of the potential flow solution process, as well as the solution of the non-linear system of equations to calculate equilibrium in the structural code. External flow is computed by taking into account the sail configuration (aerodynamic step). Once the flow field is given, the new configuration is computed (structural step). This leads to a new external flow because of the updated geometry of the sail and so on. The sequence of aerodynamic/structural steps is repeated until the stopping condition is satisfied.

If the difference between the nodal forces computed at two consecutive iterations of the flow solver is less than a tolerance, it is sufficient to consider that the change in geometry will be small compared to the previous step. Then, the stopping criteria will be met. Since \( k \) and \( k+1 \) are two consecutive iterations between the fluid solver and the structural solver, the stopping criteria will be

\[
\forall \text{node } \frac{|f_{k+1} - f_k|}{|f_k|} < \text{tol}
\]  

(10)

Different numerical tests showed that a tolerance of 0.1 has little influence on the result. In practice, most nodes have small change \((1 \cdot 10^{-7} - 1 \cdot 10^{-8})\). The error norm is usually defined by the nodes at the upper end of the mainsail, where the results of two consecutive iterations differ more than for the other sail nodes.
The scheme of the calculation process, including the grid updating algorithm inserted in the iteration process between the fluid solver and the structural solver, is shown in Fig. 2.

![Fig. 2 Fluid-Structure interaction algorithm](image)

3. Monitoring System

Despite the efforts invested in the development of monitoring systems for racing yachts in the last decades, to the author's knowledge, no flexible system able to be adapted to any rigging design and measure the structural response of elements and identify current operating configurations in real time is yet available. Therefore, a new monitoring system was developed [34].

![Fig. 3 Sensor element for monitoring traveller position (3DRRM)](image)
The electronics of the device are based on the platform Waspmote, commercialised by Libelium. The sensor, named ‘3D Remote Rigging Monitor’ or 3DRRM (Fig. 3), was designed to measure stress on any rigging element.

Experimental tests were designed to demonstrate that standard instrumentation using strain gauges is sensitive enough to accurately monitor current rigging elements. Furthermore, it was necessary to proof whether the output obtained by these means was also sensitive to the orientation between the gauge and the principal strain directions.

Eight strain gauges configured in two full Wheatstone bridges were installed in the 3DRRM. Different tensile tests were carried out to evaluate the operation of the device, showing a positive performance in all tested cases. Experimental setup for these two configurations is shown in Figure 4.

The designed sensors could process measured strain data by applying Artificial Neural Networks (ANN) algorithms to unambiguously establish the direction and magnitude of the traction force acting on the element. The function to be approximated by the ANN in this case had two inputs (strains $\varepsilon_1$, $\varepsilon_2$, measured on each side of the device) and two outputs; load and pull angle of the sheet $\{F, \alpha\} = f(\varepsilon_1, \varepsilon_2)$.

The ANN structure used had one hidden layer of perceptron neurons. The learning/validation was carried out for different number of neurons in the hidden layer. The best fitting with the validation data was the criteria used to select the optimum number.

To test the neural network, 320 FEM analysis and 76 points obtained in the laboratory tests were made available. For the validation task, 80% of the simulation data and 46 points of the experimental data were selected randomly as training data. The remainder data was used for validation purposes. The training data was used to adjust the hidden internal parameters of the neural network. Once the neural network was adjusted to the data, it was executed with the inputs of the validation data, and these results were compared with the validation data results. This comparison was done by creating a linear regression and calculating the coefficient of determination.

---

3 Waspmote http://www.libelium.com/products/waspmote
A real Time Decision Support System for the adjustment of Sailboat Rigging

Immaculada Ortigosa
Julio García-Espinosa
Marcel·la Castells

Several network configurations were used but the optimal fitting was obtained with four neurons at the hidden layer. The neural network fitting was almost perfect, obtaining a coefficient of determination of 0.996.

This way, it was possible to identify the operating conditions of the rigging.

4. Integration of a monitoring system with the FSI Algorithm

As stated above, our main goal was to develop a near-time simulation tool of the behaviour of sails and rigging to help the crew optimise real-time yacht performance. To do this, the actual configuration of the sail structure must be known in advance.

The parameters defining this configuration are the so-called trim parameters. The trim parameters considered in this work are angle to the wind, backstay load, forestay load, shroud load, main sheet load and angle, and jib sheet load and angle.

Shroud and stay stress is adjusted at dock. During sailing, the crew trim the sails by varying traveller position (twist) and adjusting sheet stress (camber). Structural and potential flow calculation depends on these adjustments. Their initial values are inserted into the GUI as initial data. This information is used to update the geometry and then compute the performance of this new configuration with the coupled fluid-structure interaction algorithm.

To measure the trim parameters, a 3DRRM [34] element is attached to any rope or cable of the sailing yacht (main sheet, jib sheet, shrouds or stays) for monitoring. The wireless monitoring elements provide the trim parameters (shroud load, stay load, main sheet load and angle, jib sheet load and angle, main traveller and jib traveller position) applied by the crew and the wind intensity and angle.

Data acquired by the WSN are transmitted to perform the aerodynamics/structural calculation via a C++ interface to interpret TCL scripts. This information is used as inputs and boundary conditions to the calculation/simulation system. The TCL interface reads the ‘external initial conditions’ and calls the C++ functions required to update the geometry. Furthermore, this system can access ‘real time’ data obtained by the WSN and update the boundary conditions to be used by the FSI solver accordingly.

Shroud load, stay load, main sheet load and jib sheet load exert a constant stress on cable elements. In addition, when the crew move the travellers (main traveller and jib traveller), their angle must be ‘virtually’ reproduced by our geometry. In order to introduce in our solver provide our solver with the capability to adapt sail position into the solver, an algorithm was created using the ideas in [34]. A particular sail trim is carried out by wrapping the jib sheet around the forestay and the mainsail around the mast.

Trim performance is analysed for a certain wind condition and structure configuration. If the wind condition changes or the crew change the structure configuration, a new performance analysis will be made. A rigging monitoring tool able to communicate in real time with the FSI solver is designed. This tool allows finding the boundary conditions which define the new structure configuration. Using these data, the performance of this trim is computed. Some validation cases were carried out to verify the results of this tool.

5. Results. An application case

One of the original aspects of the program is the communication between the sensors and the FSI algorithm. The sensors capture the trim parameters, which are communicated to the FSI algorithm for their adjustment. However, the performance of the complete tool cannot be validated with experimental data because these are very scarce. This is mainly due to two factors: (a) the difficulty in measuring the deformed shape of sails and the wind pattern producing the strain, and (b) the transient character of the phenomena involved, which makes
the measurement of the exact instantaneous sail shape/wind pattern difficult. Performing a full scale test of sails and rigging is out of the question due to the dimensions necessary for the wind tunnel. Also, a model test would not be reliable for full scale extrapolation due to the difficulty in simulating sail and rigging elasticity. Thus, the benchmark validation performed in this work is the only way to check results. In the case under study, the accuracy of this model could be sufficient for practical purposes.

The application examples presented in this section are based on the Totalboats GP42 yacht, where the WSN was tested. The sail plan geometry and rigging of this boat were reproduced in our pre-processor system GiD (see Fig. 6). Next, the required structural properties of every element, the boundary conditions (desk nodes, free rotating nodes such as the join between the mast and cable element loads) and other data were entered.

**Case 1**

The following parameters were used for rigging case:

- Apparent wind velocity is 8 m/s and apparent angle is 15°.
- Main traveller position is (6.22, 0.34, -0.59). Hence, mainsail trim angle is $\delta_F = 3.25^\circ$.
- Jib traveller position is (0.41, 0.88, -0.50). Hence, jib trim angle is $\delta_M = 8.9^\circ$.
- Main sheet and jib sheet loads are 1500 N and 700 N, respectively.
- Backstay load is 10000 N.
- Forestay load is 20000 N.
- Shroud load is 1000 N.

The results for this first configuration with an apparent wind angle of 15° are shown in Fig. 7.
Areas where the pressure coefficient has an abnormal value, i.e. a negative value, are worth noting. This abnormal value can be attributed to two reasons: flow is turbulent or separated in these areas or these areas are flapping.

In order to study the effect of apparent wind angle, in the second configuration, the apparent wind angle is 20°. The wind condition then changes, and therefore the crew trim the sails. The new trim parameters are detected by the integrated WSN:

- Apparent wind velocity is 9 m/s and apparent angle is 20°.
- Main traveller position is (6.22, 0.63, -0.59).
- Jib traveller position is (0.37, 1.095, -0.5).
- Main sheet and jib sheet loads are 2000N and 1000N, respectively.
- Shroud load is 1200 N.
- Backstay load is 15000 N.
- Forestay load is 22000 N.

These changes are introduced into Sailing by the TCL interface and the ‘virtual geometry’ is adjusted to these trim parameters. The performance of this new configuration is then computed.

Fig. 7. Results for the first configuration: Displacements, Cp and abnormal area for an apparent wind angle of 15°
The response of this structure is shown in Fig. 8.

Fig. 8 Results for the second configuration: Displacements, Cp and abnormal area for an apparent angle of 20°

It is possible to see the changes in the pressure coefficient and the areas with abnormal values.

Case 1 proves the versatility of the 3DRRM. The different configurations (shrouds, backstay, forestay, main sheet and jib sheet) were presented and analysed to obtain loads at each one of them by the 3DRRM. In this way, all the rigging elements could be monitored.

Data obtained by the 3DRRM were communicated through the TCL interface, and the geometry could be virtually trimmed. FSI software calculation was executed with the new parameters to determine the appropriateness of the model.

Case 2

Separated or turbulent flow around the head of the sails is related to the effect of jib trim angle $\delta_j$ and mainsail trim angle $\delta_M$. Marchaj [37] suggested that the best position for the foresail relative to the mainsail and its trim angle varies with the apparent wind angle. The recommended close-hauled angle of the foresail trim was $7^\circ < \delta_j < 20^\circ$. After several tests at Southampton University wind tunnel, Marchaj proved that, in order to obtain high aerodynamic efficiency, flow on the leeward side of the sail must be attached and steady. These tests demonstrated that at the small apparent wind angle of 20°, when $\delta_j = 10^\circ$ and $\delta_M = 5^\circ$, attached flow is around most of both sails, and when the apparent wind angle increases to 25° and the trim angles are the same, turbulent or separated flow regions increase.
Considering this and maintaining the values of the rest of variables and varying the apparent wind angle, the influence of jib trim angle, \( \delta_j \), is evaluated:

- Apparent wind velocity is 8 m/s.
- New main traveller position is (6.22, 0.63, -0.59). Mainsail trim angle is \( \delta_M = 5.78^\circ \).
- Main sheet and jib sheet loads are 2000N and 1500N, respectively.
- Shroud load is 1200 N.
- Backstay load is 15000 N.
- Forestay load is 22000 N.

The driving force for different apparent wind angles is shown in Fig. 9.

As can be seen in Fig. 9, higher driving forces are obtained for jib trim angles closer to the centre line of the ship and larger apparent wind angles. It is important to emphasise that the program is intended for use in upwind conditions; therefore, the angles of incidence must be less than 35°. Apparent wind angles of 50° and 60° are too large to be considered as close hauled sailing. For these angles, flow is usually not attached to the sail and the approximation of potential flow is not correct.

Moreover, when \( \delta_j \) is larger than 10°, the driving force decreases, which could be explained by Marchaj’s theory.

Example 2 shows calculations to generate a driving force database based on trim parameters and wind conditions for the development of a Decision Support System (DSS) to help skippers during navigation.
6. Conclusions

The main objective of our work was to develop a simulation program of the behaviour of sails and rigging to help the crew optimise real-time yacht performance and prevent accidents. For this purpose, a tool named *Sailing* integrating a fluid solver, a structural solver and a TCL communication interface with a wireless rigging monitoring system was designed. This tool is able to compute the performance of sail/rigging configurations and monitor the state of the rigid structure.

A flexible sensor, 3DRRM, was developed to measure stress on any rigging element of the boat. The concept of the sensor is based on measuring strain at different points of the device. These strain values are processed to unambiguously establish the direction and magnitude of the traction force acting on the element.

A network of these sensors can be integrated with the FSI solver by means of a TCL communication interface. Using the information from the WSN, the trim parameters are entered into the solver as boundary conditions.

*Sailing* provides the crew with real-time information (aerodynamic coefficients) to help them find the best trimming conditions, just like a DSS. Additionally, it informs about the state of the rigid structure to prevent accidents. Real-time monitoring is an innovation for navigation, especially for races.

*Sailing* can also perform different simulations by systematically varying trim parameters to obtain a database of structure and driving force responses depending on wind conditions and trim parameters. This database would be the first step to develop a decision support system. With this DSS, sailing yacht skippers would be able to decide in advance the optimum trimming for a specific wind condition.

7. ACKNOWLEDGEMENTS

This study was partially supported by the Ministry for Industry of Spain in the DSSAIL project TSI-020100-2008-647. This R&D project was promoted and developed by Totalmar, Compass IS, Barcelona School of Nautical Studies and the International Centre for Numerical Methods in Engineering.

The authors also want to acknowledge Juan Miguel Bone, Francesc Campà, Alberto Fernández, Clara García, Jordi Jiménez, Albert Montserrat, and Alberto Tena for their support and collaboration in this work.
REFERENCES


Inmaculada Ortigosa

Julio García-Espinosa

Marcel la Castells


STEADY STATE PERFORMANCES ANALYSIS OF MODERN MARINE TWO-STROKE LOW SPEED DIESEL ENGINE USING MLP NEURAL NETWORK MODEL

UDC 629.5
Original scientific paper

Summary

Compared to the other marine engines for ship propulsion, turbocharged two-stroke low speed diesel engines have advantages due to their high efficiency and reliability. Modern low speed "intelligent" marine diesel engines have a flexibility in its operation due to the variable fuel injection strategy and management of the exhaust valve drive. This paper carried out verified zerodimensional numerical simulations which have been used for MLP (Multilayer Perceptron) neural network predictions of marine two-stroke low speed diesel engine steady state performances. The developed MLP neural network was used for marine engine optimized operation control. The paper presents an example of achieving lowest specific fuel consumption and for minimization of the cylinder process highest temperature for reducing NOx emission. Also, the developed neural network was used to achieve optimal exhaust gases heat flow for utilization. The obtained data maps give insight into the optimal working areas of simulated marine diesel engine, depending on the selected start of the fuel injection (SOI) and the time of the exhaust valve opening (EVO).

Key words: Marine two-stroke diesel engine; MLP neural network; Numerical simulation; Utilization; Start of fuel injection; Time of exhaust valve open;

1. Introduction

Two-stroke diesel engines are the main component of ship propulsion. They are applied for propulsion of different ship types and classes due to their low price (regarding other propulsion machines), reliability, high efficiency and their very simple maintenance and servicing [1].

Turbocharging provides an increase in engine power and a modest reduction of specific fuel consumption [2]. Turbocharging causes an increase of medium effective pressure and maximum temperature of the in-cylinder process. This has an influence on the strain of engine components (as a result of differing thermal expansions) and also on the emissions of pollutants [3].
The diesel engine, as the main ship propulsion device, has to maintain very high reliability of its operation, even with the allowed degradation of performance when a failure occurs [4]. Precisely for this reason it is necessary to continuously monitor all the engine major operating parameters. Intelligent control system of the engine must have access to all diagnostic data and be able to adapt the engine to the optimal mode for desired operation [5].

In this paper, the main observed points were the engine steady states, although the numerical simulation model was not limited to steady state engine operation only. Standard engine simulations rarely include an analysis of engine transients and engine behaviour in exchanged working conditions. Simulation models based on neural networks in marine propulsion systems can achieve a number of objectives, such as the optimization of the propulsion system by changing the configuration or customizing the engine control settings [6], [7].

2. Engine specifications

Two-stroke low speed marine diesel engine 6S50MC MAN B&W, whose data were used for numerical simulations, Table 1, is originally not designed for variable settings in fuel injection and exhaust valve opening. This can be done with the same manufacturer modified engine design, which has a new designation MCE for "intelligent" engine variant (electronically controlled electro-hydraulic drives for exhaust valves and fuel injection). Manufacturer set the basic angle settings for the start of fuel injection and the opening of the exhaust valve, so different settings of these angles may worsen or improve the engine operating parameters.

<table>
<thead>
<tr>
<th>Data description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Process type</td>
<td>two-stroke, direct injection</td>
</tr>
<tr>
<td>Number of cylinders</td>
<td>6 in line</td>
</tr>
<tr>
<td>Cylinder bore</td>
<td>500 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>1910 mm</td>
</tr>
<tr>
<td>Ignition sequence</td>
<td>1-5-3-4-2-6</td>
</tr>
<tr>
<td>Maximum continuous rating (MCR)</td>
<td>8580 kW</td>
</tr>
<tr>
<td>Engine speed at MCR</td>
<td>127 min⁻¹</td>
</tr>
<tr>
<td>Maximal mean effective pressure</td>
<td>18 bar</td>
</tr>
<tr>
<td>Maximal combustion pressure</td>
<td>143 bar</td>
</tr>
<tr>
<td>Specific fuel consumption (with high efficiency turbocharger)</td>
<td>171 g/kWh, on 100% load</td>
</tr>
<tr>
<td>Compression ratio (obtained by calculation)</td>
<td>17.2</td>
</tr>
<tr>
<td>Crank mechanism ratio</td>
<td>0.436</td>
</tr>
<tr>
<td>Exhaust manifold volume</td>
<td>6.13 m³</td>
</tr>
<tr>
<td>Inlet manifold volume (with intercooler)</td>
<td>7.179 m³</td>
</tr>
</tbody>
</table>

2.1 Engine available data from test bed

The main data of the marine diesel engine are obtained by measurement [8]. Such measurements are performed during the testing of the new engine on the test bed. Table 2 presents the measured values for the selected engine steady operation points at 25%, 50%,
75%, 93.5%, 100% and 110% of engine load. The engine was produced at the Shipyard Split under the MAN B&W license.

The examination was performed at the following environment state:
- Ambient temperature 30 °C,
- Ambient pressure 1005 mbar,
- Relative humidity 50%.

The engine was tested on diesel fuel D-2, whose features are, according to a supplier report:
- Density 844.7 kg/m³,
- Kinematic viscosity 3.03 mm²/s,
- Sulfur content 0.45%,
- Net calorific value 42.625 MJ/kg.

Table 2 6S50MC MAN B&W measured data [8]

<table>
<thead>
<tr>
<th>Engine load (regarding MCR)</th>
<th>25%</th>
<th>50%</th>
<th>75%</th>
<th>93.5%</th>
<th>100%</th>
<th>110%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Indicated power (kW)</td>
<td>2401</td>
<td>4406</td>
<td>6580</td>
<td>8170</td>
<td>8656</td>
<td>9499</td>
</tr>
<tr>
<td>Effective power (kW)</td>
<td>2142</td>
<td>4099</td>
<td>6160</td>
<td>7667</td>
<td>8182</td>
<td>9014</td>
</tr>
<tr>
<td>Engine speed (min⁻¹)</td>
<td>76.5</td>
<td>96</td>
<td>110.4</td>
<td>118.5</td>
<td>121.4</td>
<td>125.2</td>
</tr>
<tr>
<td>Controller Index</td>
<td>44.3</td>
<td>55.4</td>
<td>68.1</td>
<td>77.3</td>
<td>79.2</td>
<td>85.8</td>
</tr>
<tr>
<td>Compression pressure (bar)</td>
<td>46.2</td>
<td>70.3</td>
<td>97.5</td>
<td>117.6</td>
<td>123.7</td>
<td>137.8</td>
</tr>
<tr>
<td>Maximal combustion pressure (bar)</td>
<td>66.6</td>
<td>97.4</td>
<td>129.6</td>
<td>143.3</td>
<td>141.4</td>
<td>139.3</td>
</tr>
<tr>
<td>Mean indicated pressure (bar)</td>
<td>8.37</td>
<td>12.24</td>
<td>15.89</td>
<td>18.38</td>
<td>19.01</td>
<td>20.23</td>
</tr>
<tr>
<td>Fuel rack position (mm)</td>
<td>39.7</td>
<td>50.3</td>
<td>63.3</td>
<td>73</td>
<td>75</td>
<td>81.8</td>
</tr>
<tr>
<td>Intake manifold pressure (bar)</td>
<td>1.3</td>
<td>2.03</td>
<td>2.76</td>
<td>3.33</td>
<td>3.55</td>
<td>3.93</td>
</tr>
<tr>
<td>Intake manifold temperature (°C)</td>
<td>25</td>
<td>29</td>
<td>34</td>
<td>40</td>
<td>41</td>
<td>45</td>
</tr>
<tr>
<td>Exhaust manifold pressure (bar)</td>
<td>1.3</td>
<td>1.86</td>
<td>2.51</td>
<td>3.06</td>
<td>3.26</td>
<td>3.64</td>
</tr>
<tr>
<td>Temperature before turbine (°C)</td>
<td>308</td>
<td>327</td>
<td>346</td>
<td>384</td>
<td>404</td>
<td>458</td>
</tr>
<tr>
<td>Turbocharger rotational speed (min⁻¹)</td>
<td>7290</td>
<td>11360</td>
<td>13870</td>
<td>15360</td>
<td>15895</td>
<td>17110</td>
</tr>
<tr>
<td>Specific fuel consumption (g/(kW·h))</td>
<td>186.83</td>
<td>174.06</td>
<td>171.18</td>
<td>171.82</td>
<td>174.66</td>
<td>180.5</td>
</tr>
</tbody>
</table>

2.2 Available data from simulation

The data and mathematical model, used in the simulations, are the results of scientific research project "Numerical simulation and optimization of marine diesel engines" (069-0691668-1725, Croatian Ministry of Science, Education and Sports). The developed MATLAB-SIMULINK simulation model gives satisfactory results, but unfortunately not sufficiently fast for engine real-time control and it is impractical for quick analysis. The accuracy provided by MATLAB-SIMULINK numerical simulations gave a relative error less than 3% in the interior and 5% on the borders of the engine operation field. This was a prerequisite for the high-quality neural network learning process in order to obtain her predictions at the same accuracy level. Numerical simulations can also investigate the engine working conditions outside the domain covered by the producer warranty. This is another reason why the developed MLP (Multilayer Perceptron) neural network used the results obtained by MATLAB-SIMULINK numerical simulations.

Original MATLAB-SIMULINK simulation [8] were performed by means of the engine controller acting to fuel rack regarding the load set by the propeller at engine constant speed. In the presented research, a set of engine data for randomly distributed operation points was
obtained by simulations. Before each simulation, input parameters were selected randomly. Simulations were stopped after convergence was reached. The convergence criterion was the convergence of air to fuel ratio in the cylinder. If this criterion was not met in 30 successively iterated engine cycles, convergence for that input point was not reached. For each simulation, a file of input and output data is kept. From that file, converged data points are selected and filtered. The filtering results in removal of data if the specific fuel consumption is outside the expected range (those points were not the real steady state points), Table 3.

Modern marine diesel engines with electro-hydraulic control of fuel injection and exhaust valve opening allow a very large area of engine customization in various modes. This entire area is usually too large for complete engine testing, and detailed measurements are not publicly available. This was precisely the reason due to which the development of the neural network was performed by using data obtained by numerical MATLAB-SIMULINK simulations.

### Table 3 Simulated data from MATLAB-SIMULINK [8]

<table>
<thead>
<tr>
<th>Engine load (regarding MCR)</th>
<th>25%</th>
<th>50%</th>
<th>75%</th>
<th>93.5%</th>
<th>100%</th>
<th>110%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Indicated power (kW)</td>
<td>2401.5</td>
<td>4407.5</td>
<td>6581</td>
<td>8169.5</td>
<td>8658.2</td>
<td>9499.7</td>
</tr>
<tr>
<td>Effective power (kW)</td>
<td>2141.7</td>
<td>4098.6</td>
<td>6159.7</td>
<td>7666.7</td>
<td>8181.8</td>
<td>9014</td>
</tr>
<tr>
<td>Engine speed (min⁻¹)</td>
<td>76.5</td>
<td>96</td>
<td>110.4</td>
<td>118.5</td>
<td>121.4</td>
<td>125.2</td>
</tr>
<tr>
<td>Compression pressure (bar)</td>
<td>47</td>
<td>70.06</td>
<td>97</td>
<td>116.5</td>
<td>124.1</td>
<td>137.52</td>
</tr>
<tr>
<td>Maximal combustion pressure (bar)</td>
<td>69.9</td>
<td>94</td>
<td>126</td>
<td>144</td>
<td>142</td>
<td>140.1</td>
</tr>
<tr>
<td>Mean indicated pressure (bar)</td>
<td>8.37</td>
<td>12.24</td>
<td>15.89</td>
<td>18.383</td>
<td>19.01</td>
<td>20.23</td>
</tr>
<tr>
<td>Fuel rack position (mm)</td>
<td>39.23</td>
<td>50.83</td>
<td>63.73</td>
<td>72.37</td>
<td>75</td>
<td>81.9</td>
</tr>
<tr>
<td>Intake manifold pressure (bar)</td>
<td>1.38</td>
<td>2.075</td>
<td>2.83</td>
<td>3.36</td>
<td>3.554</td>
<td>3.925</td>
</tr>
<tr>
<td>Intake manifold temperature (°C)</td>
<td>23</td>
<td>26</td>
<td>33.55</td>
<td>39.15</td>
<td>40.95</td>
<td>44.5</td>
</tr>
<tr>
<td>Exhaust manifold pressure (bar)</td>
<td>1.3</td>
<td>1.89</td>
<td>2.58</td>
<td>3.05</td>
<td>3.24</td>
<td>3.6</td>
</tr>
<tr>
<td>Temperature before turbine (°C)</td>
<td>307</td>
<td>326</td>
<td>347</td>
<td>377</td>
<td>397</td>
<td>447</td>
</tr>
<tr>
<td>Turbocharger rotational speed (min⁻¹)</td>
<td>7450</td>
<td>11356</td>
<td>13868</td>
<td>15360</td>
<td>15896</td>
<td>17113</td>
</tr>
<tr>
<td>Specific fuel consumption (g/(kW·h))</td>
<td>180</td>
<td>173.6</td>
<td>171.12</td>
<td>171.9</td>
<td>174.598</td>
<td>179.9</td>
</tr>
</tbody>
</table>

### 3. Neural network model

The power of the neural network is due to massively parallel distributed structure, and ability to learn, therefore to generalize. Generalization means that the neural network can produce "reasonable" outputs for inputs not seen during training or "learning", [9]. The smallest unit of an artificial neural network is the artificial neuron. The neuron makes the basic unit for processing the input to the output. The word artificially must be emphasized, because even though artificial neurons mimic biological neuron, it is different from biological and represents only its simplified model.
Each artificial neuron has the following elements: inputs to neuron $x_i$, connection weights $w_{kj}$, summation operator $\Sigma$, activation function $f$, bias $\theta_k$ and output from neuron $y_k$, Figure 1.

Linear sum of neuron inputs $u_k$ is defined in following equation:

$$u_k = \sum_{j=1}^{p} w_{kj} - \theta_k$$ (1)

where $w_{kj}$ are connection weights of $k$ neuron with $j$ input, and $p$ is a number of neuron inputs. The output value from neuron $y_k$ is defined in following equation:

$$y_k = f(u_k)$$ (2)

An MLP neural network with one hidden layer was chosen in this paper. The MLP neural network can have many hidden layers, Figure 2, but one layer is enough for output functions with continuous values.

The hidden layer has a sigmoid function as activation function, one of the most used activation functions. The output layer also has a sigmoid function, although it is common to have a linear function in the output layer for the problems with continuous values in outputs. The linear function in the output layer was tried but for this problem a more stable convergence was achieved with the sigmoid function. Sigmoid function is defined as:

$$y(x) = \frac{1}{1 + e^{-x}}$$ (3)

The shape of training data dictates the number of neurons in the input layer and in the output layer. The number of neurons in the hidden layer has to be set. The strategy for finding number of hidden neurons was trying numbers in ascending order (2, 4, 10, 20, 40 and 80). The number of 40 neurons was chosen. Using more neurons would result in increased difficulties in finding weights without increasing the performance of neural network, FANN (Fast Artificial Neural Network) [10].

FANN library was used for neural network learning because of the performance it gives. Learning was achieved by a custom made application developed in C programming language. Post processing was performed in scripting programming language python with pyfann library (which uses the same FANN library).
Ozren Bukovac, Vladimir Medica, Vedran Mrzljak

Study State Performances Analysis of Modern Marine Two-Stroke Low Speed Diesel Engine Using MLP Neural Network Model

Table 4 Input parameters range

<table>
<thead>
<tr>
<th>Input parameter*</th>
<th>Value range**</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>( n_{\text{M}} )</td>
<td>75 … 130</td>
<td>min(^{-1})</td>
</tr>
<tr>
<td>( x_{\text{reg}} )</td>
<td>12 … 82.7</td>
<td>mm</td>
</tr>
<tr>
<td>( \text{SOI} )</td>
<td>-10 … +10</td>
<td>°CA</td>
</tr>
<tr>
<td>( EVO )</td>
<td>-20 … +20</td>
<td>°CA</td>
</tr>
</tbody>
</table>

* For details, see Input parameters in the Table 5
** regarding the reference value of the engine

Table 5 Input and output variables list

<table>
<thead>
<tr>
<th>Ord.</th>
<th>Variable label</th>
<th>Variable description</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Konverg</td>
<td>Simulation convergence</td>
<td>1-Yes, 0-No</td>
</tr>
<tr>
<td>2</td>
<td>( n_{\text{M}} )</td>
<td>Engine speed</td>
<td>min(^{-1})</td>
</tr>
<tr>
<td>3</td>
<td>( x_{\text{reg}} )</td>
<td>Fuel rack position</td>
<td>mm</td>
</tr>
<tr>
<td>4</td>
<td>SOI</td>
<td>Start of injection</td>
<td>°CA</td>
</tr>
<tr>
<td>5</td>
<td>EVO</td>
<td>Exhaust valve open</td>
<td>°CA</td>
</tr>
<tr>
<td>6</td>
<td>( M_{\text{M}} )</td>
<td>Engine torque</td>
<td>Nm</td>
</tr>
<tr>
<td>7</td>
<td>( P_{\text{ef}} )</td>
<td>Engine power</td>
<td>kW</td>
</tr>
<tr>
<td>8</td>
<td>( b_{s} )</td>
<td>Specific fuel consumption</td>
<td>g/kWh</td>
</tr>
<tr>
<td>9</td>
<td>( T_{\text{EM}} )</td>
<td>Exhaust manifold temperature</td>
<td>K</td>
</tr>
<tr>
<td>10</td>
<td>( T_{\text{out,T}} )</td>
<td>Turbine outlet temperature</td>
<td>K</td>
</tr>
<tr>
<td>11</td>
<td>( T_{\text{IM}} )</td>
<td>Intake manifold temperature</td>
<td>K</td>
</tr>
<tr>
<td>12</td>
<td>( m_{\text{flow,T}} )</td>
<td>Mass flow on the turbine</td>
<td>kg/s</td>
</tr>
<tr>
<td>13</td>
<td>( \lambda_{\text{EM}} )</td>
<td>Air excess ratio</td>
<td>-</td>
</tr>
<tr>
<td>14</td>
<td>( p_{\text{IM}} )</td>
<td>Intake manifold pressure</td>
<td>Pa</td>
</tr>
<tr>
<td>15</td>
<td>( p_{\text{EM}} )</td>
<td>Exhaust manifold pressure</td>
<td>Pa</td>
</tr>
<tr>
<td>16</td>
<td>( n_{\text{TC}} )</td>
<td>Turbocharger rotational speed</td>
<td>min(^{-1})</td>
</tr>
<tr>
<td>17</td>
<td>( m_{\text{flow,C}} )</td>
<td>Mass flow on the compressor</td>
<td>kg/s</td>
</tr>
<tr>
<td>18</td>
<td>( p_{\text{max}} )</td>
<td>Maximum cylinder pressure</td>
<td>Pa</td>
</tr>
<tr>
<td>19</td>
<td>( T_{\text{max}} )</td>
<td>Maximum cylinder process temperature</td>
<td>K</td>
</tr>
<tr>
<td>20</td>
<td>( Q_{w} )</td>
<td>Heat transferred to the cylinder walls</td>
<td>J</td>
</tr>
</tbody>
</table>

The data available from simulation, organized as records, were divided into three data sets: for training, validation and testing. The training data set was used for training of network, validation data set was used to decide when to stop the training and testing data set for evaluation of the trained neural network performance. The size of the training data set is about 70% of data, and size of validation and testing data sets were 15% each. The lists of inputs and outputs with respective ranges and units are given in Table 4 and Table 5.
Input and output data were scaled in the range [0, 1]. Scaling is not mandatory for all MLP neural networks but the scaling of input data helps in finding initial weights giving all inputs of one record the equal importance.

Scaling of output values is important because of sigmoid activation function in output neurons. Data scaling can influence on training performance [11].

The MLP is a neural network that can be trained using supervised learning. In supervised learning, training data consist of input-output pairs, and the neural network is trying to find a mapping function that will generate the output for given input values. Through the learning process, the network changes their weights. In the training of the artificial neural network convergence depends on the initial start weights vector. So, to test some structure, it is necessary to repeat the process of learning with different randomly selected weights. Each parameter has influence on the performance. In this paper, a more advanced batch training algorithm iRPROP (improved resilient backpropagation) [12] was used, which is a variety of the standard RPROP (resilient backpropagation) training algorithm [13]. It achieves good results for many problems, the training algorithm is adaptive and the learning rate does not have to be specified.

The tanh error function is an error function that makes large deviations of stand-outs, by altering the error value used during the training of the network. This is the default error function in FANN. Usually it performs better, but, however it can give poor results with high learning rates, [10].

<table>
<thead>
<tr>
<th>Table 6</th>
<th>Error level by number of data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Training data (the number of data)</td>
<td>Validation data (the number of data)</td>
</tr>
<tr>
<td>&lt;5%</td>
<td>&lt;10%</td>
</tr>
<tr>
<td>$M_M$</td>
<td>783</td>
</tr>
<tr>
<td>$b_e$</td>
<td>809</td>
</tr>
<tr>
<td>$T_{EM}$</td>
<td>784</td>
</tr>
<tr>
<td>$T_{out,T}$</td>
<td>772</td>
</tr>
<tr>
<td>$T_{IM}$</td>
<td>785</td>
</tr>
<tr>
<td>$m_{flow,T}$</td>
<td>765</td>
</tr>
<tr>
<td>$p_{IM}$</td>
<td>803</td>
</tr>
<tr>
<td>$p_{EM}$</td>
<td>806</td>
</tr>
<tr>
<td>$n_{TC}$</td>
<td>797</td>
</tr>
<tr>
<td>$m_{flow,C}$</td>
<td>775</td>
</tr>
<tr>
<td>$p_{max}$</td>
<td>760</td>
</tr>
<tr>
<td>$T_{inax}$</td>
<td>804</td>
</tr>
</tbody>
</table>

The learning of the neural network repeats from epoch to epoch by using the training algorithm. After each epoch the errors on training and on validation data are determined. The process of learning is repeated while errors are decreasing. Training with train data is stopped when the error on validation data set starts to increase. Usually the error values tend to oscillate so it is important not to stop the iterations immediately, but to try training for a number of epochs and keep the track of weights with the minimum error. Table 6 presents the errors of the most important output variables after achieving minimum error.
When displaying simulation results of the MLP neural network, it should be considered that the angle delay of the start of fuel injection (SOI) has a negative sign for earlier shift and positive sign for the later shift in regard to reference settings by the engine manufacturer. The same angle delay principle was used also for the exhaust valve opening (EVO).

4. The ANN model applications in engine operation optimization

After the neural network learning, some analyses have been carried out. The developed neural network model allows a fast calculation of the engine steady state operation parameters. In a very short time it is possible to cross the whole area of solutions for a given SOI and EVO and to calculate all the necessary characteristics [14]. The area solutions for searched SOI and EVO are available in discrete steps of 0.5 °CA. The decision making for the best SOI – EVO combination depends mostly on the specific fuel consumption \( b_c \), the maximum cylinder pressure \( p_{\text{max}} \) and the maximum cylinder process temperature \( T_{\text{max}} \) [15], [16].

4.1 ANN results for a full engine load and for 50% engine load

![Fig. 3 Engine effective power, \( P_{\text{ef}} \) [MW] - for a full load](image)
![Fig. 4 Engine effective power, \( P_{\text{ef}} \) [MW] - for a 50% load](image)

Effective power shows the same trend, regardless of whether it is at full load or at 50% engine load, Figure 3 and Figure 4. SOI has a decisive influence on the engine effective power. The maximum effective power was reached with the start of fuel injection just before the engine factory settings. Moving the SOI for later reduces the effective power, and later injection causes the proportional decrease in effective power. EVO has an almost constant effect on the engine effective power for the selected SOI.

![Fig. 5 Specific fuel consumption, \( b_c \) [g/kWh] - for a full load](image)
![Fig. 6 Specific fuel consumption, \( b_c \) [g/kWh] - for a 50% load](image)

The specific fuel consumption has been the lowest for the SOI just before the factory settings, and the same is optimal for engine operation regarding the specific fuel consumption and effective power. Also in this situation, for specific fuel consumption, EVO has an almost
constant effect for the selected SOI. The specific fuel consumption change has a similar trend at full and at 50% engine loads, Figure 5 and Figure 6.

The exhaust gases thermal flow at the turbine outlet leads to similar conclusions as for the exhaust gases temperature after the turbine, regardless of the engine load, Figure 7 and Figure 8. Although it is not explicitly visible in Figure 8, this picture also points to the engine operation instability at a very early SOI and late EVO at 50% load, which is reflected in the changes of the engine operation area borders.

At full engine load, Figure 9 shows that the maximum pressure in the cylinder is obtained for very early SOI shift and very late EVO shift. This working area at this engine load is stable and there is no danger of falling out of operation. Also, in this same area the effective power decreases and the specific fuel consumption increases, and surely this working area is not preferred for selection. The maximum cylinder pressure rapidly decreases for later SOI shift, and earlier EVO shift regarding to the engine referent values.
Maximum cylinder pressure at 50% load shows the same trend as for a full engine load, Figure 10. The only exception is the area of early SOI shift and late EVO shift, which also at this engine load shows unstable engine operation due to worsened scavenging process.

The highest temperature in the engine cylinder, for both of the observed loads was achieved at a very early SOI shift and proportionally early EVO shift, Figure 11 and Figure 12. However, it should be noted that too high temperatures in the engine cylinder cause high emissions, primarily emissions of nitrogen oxides, and under these conditions the engine certainly could not provide the required environmental standards. Regarding the maximum temperature of the engine process, usually a compromise between the achieved emissions and the produced heat necessary for utilization has to be found.

4.2 The optimization of SOI and EVO for maximum thermal flow of exhaust gases for utilization

The simulation results indicated that the SOI shift for 3.5 °CA later, and EVO shift for 20 °CA later, ensure the highest exhaust gas thermal flow, 9.5% higher than the reference one, Figure 13, with an increase in specific fuel consumption, Figure 14. The increase in specific fuel consumption is small enough, that increased fuel cost will be very quickly paid off by using higher obtained exhaust thermal flow for the utilization process. At the same time the engine power (ie. engine torque at the constant engine speed) decreased by approximately 7%, Figure 15.

Presented simulation results show justifiability for SOI and EVO shifts, in order to obtain a sufficient additional thermal flow, which can be effectively applied in the utilization process. In that way, it is possible to achieve significant savings in the ship propulsion plant with such a diesel engine and with the possibility of achieving multi-criteria optimization.
In this mode of engine operation, excessive pressures and temperatures in the engine cylinder can be avoided, Figure 16 and Figure 17. This fact proves that the displayed change of operating parameters would not lead to significant thermal load increase, or to high increase in emissions. Such operating parameters change during the actual engine operation surely will result in a substantial impact on the entire propulsion plant efficiency.

Fig. 16 Maximum pressure in the engine cylinder $p_{\text{max}}$ [MPa]

Fig. 17 Maximum temperature in the engine process $T_{\text{max}}$ [$^\circ$C]

Graphs in figures 5, 7, 9, 11 and figures 13, 14, 16, 17 are similar in shape but the ranges are not the same. The figures show various operating modes.

4.3 Satisfying the required thermal capacity at constant torque and engine speed

In this case, the engine operating point was given with engine speed $n_M = 118.5 \text{ min}^{-1}$ and the required torque $M_M = 465 \text{ kNm}$. The minimal required exhaust thermal flow after turbine $\dot{Q}_\text{min} = 3200 \text{ kW}$ was also given. The limits on the position of the fuel rack were between 40 and 75 mm. Additionally, the limits for the highest maximum temperature of the engine process $T_{\text{max,lim}} = 1900 \, ^\circ\text{C}$ and the maximal allowable pressure in the cylinder $p_{\text{max,lim}} = 14 \, \text{ MPa}$ were also set. The pressure values from the simulation can exceed the limited value of 14 MPa but those working points are constrained in optimisation algorithm because they are not used in real engine operation.

The initial idea was to find the engine operation point at which all given conditions are met. For referent settings of $SOI$ and $EVO$ the given value of exhaust thermal flow was not achieved. In order to achieve the operating point where the parameters are equal or the nearest possible to default ones, $SOI$ and $EVO$ shifts were allowed.

In order to achieve and maintain the engine torque, since it varies by $SOI$ and $EVO$ shifts, it was necessary to make a fuel rack position correction. After this step, the engine has reached a working point where the desired thermal flow was satisfied. In that working point, the desired engine torque was also reached.

Simulation passes the entire field of $SOI$ and $EVO$ shifts, and for each of the shifts a new fuel rack position was calculated. With the new position of the fuel rack, the predefined engine torque and engine speed ($n_M = 118.5 \, \text{min}^{-1}, M_M = 465 \, \text{kNm}$) must be satisfied. Then it is checked if the new operation point satisfies a predetermined minimum exhaust thermal flow on the turbine outlet. Finally, the simulation checks if the newly determined operation point has a lower specific fuel consumption than the previous one. If at least one point satisfies all these conditions, the system has a solution.
At given operating conditions, engine torque shows almost constant value, but stable change for almost the entire working area, for all SOI and EVO shifts, Figure 18. The only exceptions are the areas of large SOI shift later than the reference value, with intense loss of engine torque, and thus engine power, due to the worsened scavenging conditions. A major SOI shift to later shows late fuel injection, so in this area incomplete combustion can be expected, which results in a huge loss of engine torque. It is necessary to avoid the area where these phenomena occur, because it is impossible to achieve a stable operating point.

Even in this operation mode, specific fuel consumption was the lowest at the reference (factory) engine settings, Figure 19. Large increases in specific fuel consumption occurred only at intense shift of SOI for later, where a huge exhaust thermal flow at the turbine outlet was available, Figure 20, but it is necessary to avoid this operating area due to the large reduction in engine torque and highly probable fall-out from the drive, Figure 18. Maximum exhaust thermal flow, in the engine stable operation area, is presented in Table 7.

Table 7 Exhaust heat flow maximizing for a given $n_M = 118.5 \text{ min}^{-1}$ and $M_M = 465 \text{ kNm}$

<table>
<thead>
<tr>
<th>SOI (^\circ\text{CA})</th>
<th>EVO (^\circ\text{CA})</th>
<th>$M_M$ \text{kNm}</th>
<th>$b_e$ \text{g/kWh}</th>
<th>$x_{\text{reg}}$ \text{mm}</th>
<th>$P_{\text{et}}$ \text{MW}</th>
<th>$T_{\text{max}}$ \text{°C}</th>
<th>$p_{\text{max}}$ \text{MPa}</th>
<th>$\dot{Q}$ \text{[kW]}</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>457.7</td>
<td>165.7</td>
<td>55</td>
<td>5.68</td>
<td>1452.2</td>
<td>11.67</td>
<td>2984.4</td>
</tr>
<tr>
<td>+4.5</td>
<td>-2.5</td>
<td>465</td>
<td>175.7</td>
<td>59</td>
<td>5.77</td>
<td>1404.6</td>
<td>10.28</td>
<td>3300.9</td>
</tr>
</tbody>
</table>
Maximum cylinder pressure occurs in areas of very early SOI shift, and very late EVO shift, Figure 21. Therefore, for the maximum cylinder pressure it is optimal to hold SOI and EVO parameters to reference values, with the recommended EVO shift to earlier, in order to avoid excessively high pressures. In this area, other operating parameters do not indicate a sudden or unexpected change, so this engine operating area would be advisable for given conditions. EVO shift to earlier, while retaining the referent SOI, would be recommended also for maximum engine process temperature, Figure 22, because the maximal temperature would be optimal for utilization, and thermal load of engine working parts or emissions remain acceptable. The optimal solution could be achieved also with more complex methods of optimization (multi-criteria optimization, multi-objective optimization, etc.).

5. Conclusion

In this paper, the changes in the characteristics of "intelligent" marine two-stroke diesel engine were studied, when crank angles for the start of fuel injection (SOI) and for the opening of the exhaust valve (EVO) were shifted. The fuel injection strategy (fuel injection flow) and the exhaust valve opening curve did not change, which was left for future research. The investigations have pointed to the great potential that provides electro-hydraulic control of fuel injection and exhaust valve drive to bring the modern marine diesel engine in the desired working conditions.

Some aexamples of the described neural network applications in optimization of marine diesel engine were presented, in order to achieve the desired exhaust heat flow for the utilization purposes, along with minimum specific fuel consumption, as well as to maintain maximum engine process temperature as low as possible in order to reduce NOx emission.

The developed neural network model is fully prepared for the reception of new data, measured during the engine operation. With comparisons of measured data and data obtained by the neural network, it will be possible to evaluate the quality of measured data and the entire measuring system. This was already proven on various sets of measured data.

The neural network model was developed using data obtained from numerical simulations, for the engine steady state operation, with verification from available data measured on the test bed. Therefore, the existence of high-quality numerical simulation model in neural network development was very important. Also, the resulting neural network model has limitations (eg. the model is valid for engine steady state operation, for the same type of engine, the same selected turbocharger etc., but the obtained structure can effectively learn on the data for a new engine type).

Data for the neural network learning and testing must be within all steady state regimes of engine operation. But once the requested data were obtained, and neural network optimized, learned and operational, it is capable to give the required engine data almost 3000 times faster in regards to conventional numerical simulation.
References


M.M. Moustafa

ISSN 0007-215X
eISSN 1845-5859

OPTIMIZATION PROCEDURE FOR PRELIMINARY DESIGN STAGE OF CAIRO-DAMIETTA SELF-PROPELLED GRAIN BULK SHIPS

UDC 629.5.012:546.2
Professional paper

Summary

The global logistics center for the storage and handling of grain which will be constructed at Damietta port will extremely increase the annual movement of grain through Cairo-Damietta waterway. Therefore, the demand for inland grain bulk ships has increased significantly in the recent years.

This paper introduces a procedure to find out the fleet size and optimum characteristics of self-propelled grain bulk ships working between Cairo and Damietta through River Nile. The characteristics of the Cairo–Damietta waterway are investigated to define the constraints on dimensions and speed for such ship type. Also, mathematical model for the objective function was developed considering: powering, voyage, weight, stability and cost calculation. In this research, Specific cost (Sc), cost of transporting one ton of cargo a distance of one kilometre, is considered as the objective function for this optimization process.

This optimization problem is handled as a single objective nonlinear constrained optimization problem using a specially developed computer program. Solutions are generated by varying design variables systematically in certain steps. The best of these solutions is then taken as the estimated optimum. Finally, the problem is presented, the main constrains analyzed and the optimum solution shown.

Key words: optimum ship design; inland grain bulk ship; Cairo-Damietta waterway;

1. Introduction

Egypt is one of the largest importers of grain in the world. Where, Egypt and the surrounding region import about 50% of the size of the grain trade in the world. Hence the importance of the global logistics center for the storage and handling of grain, which its technical studies began in May, 2014. This center not only will cover the needs of the local market, but also will supply the surrounding regional markets by their needs. Therefore, the global logistics center for the storage and handling of grain will make Egypt the most important international hub for handling and storage of grain.
The Egyptian government was found that the Damietta port is best suited place for a number of key elements in the project. One of them it’s excellent location on the Mediterranean Sea and near the northern entrance of Suez Canal. Also, Damietta port connects the Mediterranean Sea with the Egyptian inland waterways network.

On the other hand, 600 million tons of goods will need to be transported inside Egypt annually [1]. Therefore, the inland water transportation system has to be further developed if the country wants to cope with such a large rise in traffic of cargo. Nowadays, serious and prompt action is taken by the Ministry of Transport to increase river transport share in the transport of goods and planned to be up to 10% (60 million tons) of the transport volume in Egypt during the next ten years, instead of a 0.5% today. For this reason, River Transport Authority (RTA) implements a plan for the periodic maintenance and purges of the Egyptian inland waterways network, and thus contributing to facilitate transportation movement throughout the year.

The possible increase in the amount of goods, which will be transported through the River Nile, requires an increase in the size of the inland water transportation fleet. Therefore, the aim of the present work is to find out the optimum fleet size and dimensions of self-propelled grain bulk ships working between Cairo and Damietta through the River Nile.

2. Statement of the problem

The Egyptian government, in March 2015, said that a contract had been signed to construct the global logistics center for the storage and handling of grain at Damietta port. This project will extremely increase the annual movement of grain through Cairo-Damietta waterway. Therefore, a fleet of self-propelled grain bulk ships is to be designed and constructed. The fleet is to carry grain from Damietta port, where the global logistics center for the storage and handling of grain already exists, to Cairo which is considered as the most populated city in Egypt.

The fleet is to be designed according to the economic criterion discussed below which will deliver a total of 0.5 million tons of grain each year. Thus, a small saving in the cost of transportation one ton of cargo on this navigation route means a huge saving in money in the ship life. Therefore, it becomes mandatory that the ship’s major characteristics have to be chosen very carefully. The design problem is to optimize the dimensions and speed of the vessel taking into account the existing navigation restrictions along Cairo-Damietta waterway.

3. Cairo-Damietta waterway

Cairo-Damietta waterway can be considered as the most important transportation path between Cairo and the outer exposed waterways to the open sea at Damietta Port. This waterway is classified as a first class waterway according to the classification of the General Egyptian River Transport Authority [2]. Characteristics of the first class waterways in Egypt are listed in Table1.

<table>
<thead>
<tr>
<th>Table1</th>
<th>Characteristics of first class waterways in Egypt</th>
</tr>
</thead>
<tbody>
<tr>
<td>Criterion</td>
<td>River Nile and its Branches</td>
</tr>
<tr>
<td>Min. bridge height</td>
<td>13 m</td>
</tr>
<tr>
<td>Min. water depth</td>
<td>2.5 m</td>
</tr>
<tr>
<td>Min. canal width</td>
<td>35 m</td>
</tr>
<tr>
<td>Max. ship draft</td>
<td>1.8 m</td>
</tr>
</tbody>
</table>
Cairo-Damietta waterway is about 260 Km long and mainly used for transportation of goods and raw materials by different ship types. The final 12 km of the navigation route to Damietta Port is via an artificial canal. This canal has a water depth of 4 m and width of 40 m at bottom level. Cairo-Damietta waterway has many restrictions that greatly influence navigation. The most important restrictions are the water depth, bridge height (air clearance) and the size of the locks. This waterway contains three locks (Delta barrages, Zifta and Faraskour), nine movable bridges and two fixed bridges. Faraskour lock is the smallest lock in Cairo-Damietta waterway where it has a length and width of 170 m and 17 m, respectively. Damietta high bridge is the lowest bridge in Cairo-Damietta waterway where its height is 8m above water surface at the highest tide level. Cairo-Damietta waterway is shown in Figure1.

4. Optimization Problem

The problem under consideration is handled as a single objective nonlinear constrained optimization problem. This optimization problem can be formulated as follows, [3];

\[
\begin{align*}
\text{Find } \quad & X = \begin{bmatrix} X_1 \\
X_2 \\
\vdots \\
X_m \end{bmatrix} \\
\text{which maximizes } & F(X) \text{ subjected to the following constraints;}
\end{align*}
\]

\[
\begin{align*}
& g_j(X) \leq 0, \quad j = 1, 2, \ldots, q \\
& h_j(X) = 0, \quad j = 1, 2, \ldots, p
\end{align*}
\]

where, “m” is the number of design variables. Also, “q” and “p” are the number of inequality and the equality constraints, respectively.
Objective function

The broad approach to choosing the best ship is quite easy to understand. Simply, it is to optimize a cost function which has been appropriately chosen. Specific cost (Sc), cost of transporting one ton of cargo a distance of one kilometre, is considered as the objective function for this optimization process. Thus, this optimization procedure leads to a minimum cost of shipping each ton of cargo each year. For each ship, specific cost can be calculated as follows [4]:

\[
Sc = \frac{Average\ Annual\ Cost\ (AAC)}{Annual\ Transported\ Cargo\ *\ Route\ Distance}
\]

\[
AAC = (P - \frac{S_v}{(1-i)^N} + C_{tau}) \cdot CR
\]

Specific cost (Sc) can be obtained by a series of calculations. In order to do that it is most important that all cost considerations be given as functions of the physical variables [5]. If this can be accomplished prior to any decisions regarding the physical characteristics of the ships, then it will be possible to evaluate the objective function for any values of those variables.

Design variables

The problem under consideration involves eight design variables. These variables are shown in Table (2).

<table>
<thead>
<tr>
<th>No.</th>
<th>Design Variables</th>
<th>Explanation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Ns (ship)</td>
<td>Fleet size</td>
</tr>
<tr>
<td>2</td>
<td>Loa (m)</td>
<td>Length overall</td>
</tr>
<tr>
<td>3</td>
<td>B (m)</td>
<td>Molded breadth</td>
</tr>
<tr>
<td>4</td>
<td>Bst (m)</td>
<td>Width of side tank</td>
</tr>
<tr>
<td>5</td>
<td>T (m)</td>
<td>Design draft</td>
</tr>
<tr>
<td>6</td>
<td>D (m)</td>
<td>Molded depth</td>
</tr>
<tr>
<td>7</td>
<td>Hdb (m)</td>
<td>Height of double bottom</td>
</tr>
<tr>
<td>8</td>
<td>Vs (km/h)</td>
<td>Ship speed</td>
</tr>
</tbody>
</table>

Design constraints

The constraints are functional relationship between the design variables. These constraints define the feasible region from which the optimum solution has to be found.

Geometry constraints

Relationships between principal dimensions of self-propelled River Nile ships are investigated to clarify the acceptable limits on the dimensional ratios (Loa/B, Loa/D and B/T) for such ships. These constraints are formulated as follows [1]:
4.3.2 Navigation constraints

Shallow water nature and the presence of locks and bridges along Cairo-Damietta waterway; represent several constraints on the speed and dimensions of Cairo-Damietta grain bulk ships. These constraints are formulated as follows:

1. The breadth of Cairo-Damietta grain bulk ships is dictated by the dimensions of Faraskour lock. Where, its dimensions are smaller than the dimensions of other locks. This constraint can be formulated as follows;

\[ B \leq 15.4 \, m \]  

Maximum ship breadth is obtained after subtracting 1.6 m from the width of Faraskour lock (for each side, 0.3 m fender and 0.5 m clearance).

2. The draft of Cairo-Damietta grain bulk ships is often dictated by the shallow water nature of such navigation route. This constraint can be formulated as follows;

\[ T \leq 1.5 \, m \]  

3. The speed of Cairo-Damietta grain bulk ships is often dictated by the shallow water nature of such navigation route. The right choice of ship’s speed should be decided in the preliminary design stage based on the Froude depth number (Fnh) to avoid the critical region. In this study the Froude depth number is taken equal to 0.7, [1]. This constraint can be formulated as follows;

\[ \frac{V_s}{\sqrt{g \, h_w}} \leq 0.7 \]  

4. The air draft of Cairo-Damietta grain bulk ships is often dictated by the existing bridges. Damietta high bridge is the lowest bridge in Cairo-Damietta waterway where its height is 8 m above water surface [2].

4.3.3 Constraint on block coefficient

Most inland vessels are characterized by large values of the hull block coefficients in order to achieve a larger displacement at low draught and decrease their building cost. Therefore, most inland vessels have a block coefficient \( C_B \) that varies from 0.8 to 0.9, [6]. This constraint can be formulated as follows;

\[ 0.8 \leq C_B \leq 0.9 \]
4.3.4 Weight Balance Constraint

This equality constraint is handled to enforce the balance between ship weight and displacement. This constraint can be formulated as follows:

$$\Delta = W_{light} + D_{wt}$$ (13)

4.3.5 Stability Constraint

The transverse metacentric height ($GM_T$) must be greater than 5% of ship breadth, [7]. This constraint can be formulated as follows:

$$GM_T \geq 0.05 B$$ (14)

4.3.6 Freeboard Constraint

The freeboard ($F_B$) of the inland navigation ships must be greater than 0.5 meters, [8]. This constraint can be formulated as follows:

$$D - T \geq 0.5 m$$ (15)

The required freeboard is compared, in the computer program, to the freeboard arrived at from volumetric requirements, and the larger of the two is used.

5. Model formulation

The subsequent sections provide the details of the modeling parameters and their mathematical formulation. The estimation methods that have been used to model the problem are also detailed.

5.1 Power evaluation

In this study, Equation (16) is used in the preliminary design stage to calculate the power ($P_B$) of a self-propelled Cairo-Damietta grain bulk ships, [1].

$$P_B = 0.02 [ V_S^3 \Delta^{2/3} ]^{0.841}$$ (16)

This equation (a form of Admiralty Equation) has been developed using curve fitting with low regression coefficient [1]. It can be used also to calculate the preliminary power of self-propelled Cairo-Damietta grain bulk ships as it covers a lot of similar ships. Also, it is naturally that some points fall out of the average field covered by the curve fitted powers.

5.2 Trip calculation

The number of round trips per year ($N_{trip}$) mainly depends on the vessel operational days ($VO_d$), which is defined as the number of days per year in which the vessel is actively being used, either traveling between ports or loading or unloading at ports. For the purposes of this work, operational days ($VO_d$) are assumed to be 350, allowing 15 days per year for vessel maintenance and repair. The number of operational days divided by the round trip time ($T_{trip}$) determines the number of potential trips a vessel can make in a year.

$$N_{trip} = \frac{VO_d}{T_{trip}}$$ (17)
Round trip time \( T_{\text{trip}} \) is the taken time by a vessel to travel between two ports, including loading, unloading and any other eventuality. It can be calculated as follows:

\[
T_{\text{trip}} = T_{\text{nav}} + T_{\text{port}}
\]  

(18)

Navigation time \( T_{\text{nav}} \) is the time the vessel spends traveling between ports, which depends on round trip distance and vessel speed. River Nile ships are prevented from sailing at night as a result of the currently improper navigational conditions on the waterway itself. Therefore, sailing time is taken equal to 12 hours per day. Navigation time can be calculated as follows:

\[
T_{\text{nav}} = \frac{(1 + Na) \cdot Ru}{12 \cdot V_s}
\]  

(19)

Port time is a function of vessel cargo capacity and the port cargo handling rate. Cargo handling operations can be continued for 24 hours per day in the River Nile terminals. In this paper, it is assumed that, loading time equals unloading time. Also, loading and unloading operation are carried out by terminal equipment. Port time is multiplied by two to reflect the loading and unloading activities in both origin and destination ports. Port time can be calculated as follows:

\[
T_{\text{port}} = \frac{2 \cdot (1 + Pa) \cdot Dwt}{24 \cdot CHR}
\]  

(20)

The fuel and diesel consumption per trip \( FC_{\text{trip}} \) may be calculated according to the following equations:

\[
FC_{\text{trip}} = FC_{\text{nav}} + DC_{\text{port}} + DC_{\text{nav}}
\]  

(21)

\[
FC_{\text{nav}} = \frac{SFC \cdot P_B \cdot (12 \cdot T_{\text{nav}})}{10^{-6}}
\]  

(22)

In the present study, the daily diesel oil consumption is taken equal to 2 tons for both port and navigation times.

5.3 Weight calculation

Ship weight \( \Delta \) can be calculated according to the following equation:

\[
\Delta = W_{\text{light}} + Dwt
\]  

(23)

Ship deadweight \( Dwt \) is a notation of the ship carrying capacity. Ship deadweight \( Dwt \) is taken as the weight of the cargo, fresh water, fuel and miscellaneous weights. Ship lightweight \( W_{\text{light}} \), the weight of the complete ship, being ready for service but empty, is obtained by adding outfitting and machinery weights to the steel weight. In this study, 2% of light ship weight is taken as a margin [5]. The purpose of this margin is giving an allowance to ensure the attainment of the specified deadweight in case of underestimating the lightship weight, and also to compensate for possible departures from the initial weight design during construction.

Steel weight \( W_{\text{steel}} \) of Cairo-Damietta self-propelled grain bulk ships is calculated according to equation (24), [9]. In this paper, for transversely framed dry bulk vessels with draft equal to 1.5m, the coefficients \( C_1 \) and \( C_2 \) are taken equal to 1.8E-5 and 2.37E-1, respectively [9].
\[ W_{steel} = C_1 \left[ LBT \right]^2 + C_2 LBT \] (24)

Outfitting weight \(W_{out}\) of Cairo-Damietta self-propelled grain bulk ships is calculated according to equation (25), [10].

\[ W_{outfit} = ko \ L \ B \] (25)

where, \(ko\) is a coefficient based on ship types. \(ko\) is taken as 0.028 tons/m\(^2\) for inland cargo ships, [11]. The first step towards assessing the machinery weight \(W_{mc}\) is the calculation of the required power to drive a ship. The second step involves taking a decision on the type of machinery best suited to the service conditions of the ship under consideration. In the absence of manufacturers’ specifications, a value between \((0.012 – 0.02 \text{ t/kw})\) can be used as approximate unit weight for medium speed diesel engines, [10]. In this study a value of \(0.015 \text{ t/kw}\) is used as approximate unit weights for diesel engines of Cairo-Damietta self-propelled grain bulk ships.

5.4 Stability calculation

The computer program alters the breadth of the proposed ship until the transverse metacentric height \(GM_T\) is at least 5% of ship breadth. The transverse metacentric height \(GM_T\) can be calculated according to the following formula:

\[ GM_T = KB + BM_T - KG \] (26)

The vertical center of buoyancy \(KB\), for inland ships, can be calculated according to the following formula, [12];

\[ KB = 0.535 \ T \] (27)

The transverse metacentric radius \(BM_T\), can be calculated according to the following formula, [10];

\[ BM_T = \frac{B^2}{24 T C_B} \left[ 3 C_{WL} - 1 \right] \] (28)

The general assumption regarding the vertical center of gravity (KG) is that its position is 57% of the depth above keel.

5.5 Cost assessment

5.5.1 Ship capital cost

Ship capital Cost \(P\) is broken down into steel cost \(C_{steel}\), outfitting cost \(C_{out}\) and machinery cost \(C_{mc}\).

\[ P = C_{steel} + C_{out} + C_{mc} \] (29)

In this paper, 10% of ship capital cost \(P\) is taken as additional cost to cover the other costs (classification society, model testing, docking, external services, tugs ...etc). Also, it is assumed that, 50% of ship capital cost is equity capital and the rest is lend from a bank for 10 years annual equal payments at 10% interest.
Hull steel cost ($C_{\text{steel}}$) is calculated by multiplying the steel weight by a fixed value for manufacturing of one ton of steel. An average value of 8000 LE has been taken for the evaluation as a valid present figure.

Outfitting cost ($C_{\text{out}}$), being generally recognized as one of the most difficult and design-specific factors to calculate, is determined as a function of outfitting weight to the $2/3$ power [5]. In this paper, based on a reference vessel, outfitting costs ($C_{\text{out}}$) can be calculated using the following formulae, [9]:

$$C_{\text{out}} = 40000 W_{\text{out}}^{2/3}$$ (30)

In this paper, a value of 220 €/kW is used to determine the cost of the propulsion system ($C_{\text{mc}}$) for modern inland ship engines of 330 to 500 kW of rated power [9]. The main engine is the most expensive item in the ship’s equipment and its share in the total costs of a ship can be up to 15%, [13]. Hence, minimizing the main engine power is of great importance [13]. Ship outfitting costs ($C_{\text{out}}$) and machinery costs ($C_{\text{mc}}$) are calculated in Euro. Therefore, in this paper, a factor of 8.5 is used to convert these costs to Egyptian pound.

5.5.2 Annual Operating Cost

Figure (2) shows the elements of ship annual operating cost. In the present work, the annual operating costs ($C_{\text{ao}}$) are allowed to escalate with a rate of 5% throughout the life span and projected again to the first year of ship’s life using the present value techniques. Therefore, Equations (31) and (32) can be used to calculate ship annual operating cost at any year and present worth of life span operating expenses, respectively.

![Fig.2 Annual operating cost](image)

$$\left[ C_{\text{ao}} \right]_n = C_{\text{ao}} \cdot \left[ 1.05^{n-1} \right]$$ (31)

$$C_{\text{tao}} = \sum_{n=1}^{N} \left( \frac{\left[ C_{\text{ao}} \right]_n}{(1 + i)^n} \right)$$ (32)
The crew cost ($C_{wages}$) may be calculated according to the following formula;

$$C_{wages} = 12 A_{wage} \cdot N_{crew}$$  \hspace{1cm} (33)$$

The annual fuel cost ($C_{fuel}$) may be calculated according to the equation;

$$C_{fuel} = FC_{trip} \cdot F_{price} \cdot N_{trip}$$  \hspace{1cm} (34)$$

Victualling are usually bought locally at the ship’s trading ports and the annual cost is calculated on a per person per day basis. Victualling cost ($C_{vict}$) may be calculated according to the following equation;

$$C_{vict} = 350 C_{day} \cdot N_{crew}$$  \hspace{1cm} (35)$$

Ship carrying capacity are handled two times (loading and unloading) in each round trip. Cargo handling cost ($C_{ch}$) may be calculated according to the following equation;

$$C_{ch} = 2 D_{wt} \cdot N_{trip} \cdot C_{hot}$$  \hspace{1cm} (36)$$

The port expenses ($C_{port}$) may be calculated according to the following equation;

$$C_{port} = 2 D_{wt} \cdot N_{trip} \cdot f_{port}$$  \hspace{1cm} (37)$$

The maintenance and repair costs ($C_{mar}$) may be calculated according to the equations, [9];

$$C_{mar} = 5 L B T + 0.009 \cdot \frac{T_{nav}}{T_{trip}} \cdot (12 V_{O_d}) \cdot P_B$$  \hspace{1cm} (38)$$

This cost is calculated in Euro. Therefore, in this paper, a factor of 8.5 is used to convert this cost to Egyptian pound. Insurance costs ($C_{insu}$) may be calculated according to the following equation, [11];

$$C_{insu} = 0.11 P$$  \hspace{1cm} (39)$$

Administration cost ($C_{admin}$) is a contribution to the office expenses of a shipping company or the fees payable to a management company plus a considerable sum for communications and sundries. It can be taken equal to 10% of the annual operating costs, [11]. Financing cost ($C_{fin}$), during loan period, may be calculated according to the following equations [5];

$$R_b = P_b \cdot CR$$  \hspace{1cm} (40)$$

$$CR = \frac{i (1 + i)^{n_b}}{(1 + i)^{n_b} - 1}$$  \hspace{1cm} (41)$$

$$C_{fin} = R_b - \frac{P_b}{n_b}$$  \hspace{1cm} (42)$$

where, $P_b$, $R_b$, $n_b$ and $CR$ are borrowed capital, bank annual payment, loan period and capital recovery factor, respectively.
6. Developed computer program

The present optimization problem is carried out by using a specially developed Visual Fortran computer program. This program is illustrated by the flow chart shown in Figure (3).

In this program the design variables are varied in a sequential manner over a range of different step sizes. Thus, this program deals with a multi-dimensional problem whose size is a function of the number of variables, the step size and the specified range of each variable. The lower limit, upper limit and step size of the design variables are shown in Table 3.
Table 3: Lower limit, upper limit and step size of the design variables

<table>
<thead>
<tr>
<th>No.</th>
<th>Design Variable</th>
<th>Lower Limit</th>
<th>Upper Limit</th>
<th>Step Size</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>N_s</td>
<td>1 ship</td>
<td>15 ship</td>
<td>1 ship</td>
</tr>
<tr>
<td>2</td>
<td>Loa</td>
<td>30.0 m</td>
<td>100.0 m</td>
<td>0.10 m</td>
</tr>
<tr>
<td>3</td>
<td>B</td>
<td>6.0 m</td>
<td>15.6 m</td>
<td>0.05 m</td>
</tr>
<tr>
<td>4</td>
<td>Bst</td>
<td>0.8 m</td>
<td>1.5 m</td>
<td>0.05 m</td>
</tr>
<tr>
<td>5</td>
<td>T</td>
<td>1.0 m</td>
<td>1.5 m</td>
<td>0.05 m</td>
</tr>
<tr>
<td>6</td>
<td>D</td>
<td>2.5 m</td>
<td>3.5 m</td>
<td>0.05 m</td>
</tr>
<tr>
<td>7</td>
<td>Hdb</td>
<td>0.8 m</td>
<td>1.0 m</td>
<td>0.05 m</td>
</tr>
<tr>
<td>8</td>
<td>V_s</td>
<td>10 km/h</td>
<td>18 km/h</td>
<td>0.2 km/h</td>
</tr>
</tbody>
</table>

In this paper, lower limits of double bottom height and width of side tanks are taken equal to 0.8 m as required by the Egyptian River Transport Authority (RTA). Also, double bottom height and width of side tanks are chosen to provide enough volume for the required ballast quantity (at least 60% of cargo weight). Table (4) contains the input data of the developed program while Table (5) contains the output results.

Table 4: Input Data – Developed Program

<table>
<thead>
<tr>
<th>No.</th>
<th>Item</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Water depth (h_w)</td>
<td>2.5 m</td>
</tr>
<tr>
<td>2</td>
<td>Round trip distance (Ru)</td>
<td>520 km</td>
</tr>
<tr>
<td>3</td>
<td>Nile allowance (Na)</td>
<td>0.10</td>
</tr>
<tr>
<td>4</td>
<td>Port allowance (Pa)</td>
<td>0.20</td>
</tr>
<tr>
<td>5</td>
<td>Ship life (N)</td>
<td>25 years</td>
</tr>
<tr>
<td>6</td>
<td>Scrap value (S_v)</td>
<td>0.15</td>
</tr>
<tr>
<td>7</td>
<td>Interest rate (i)</td>
<td>10%</td>
</tr>
<tr>
<td>8</td>
<td>Cargo handling rate (CHR)</td>
<td>50 ton/h</td>
</tr>
<tr>
<td>9</td>
<td>Cargo handling cost (C_hot)</td>
<td>5 LE/ton</td>
</tr>
<tr>
<td>10</td>
<td>Port expenses (f_port)</td>
<td>1.0 LE/ton</td>
</tr>
<tr>
<td>11</td>
<td>Number of crew (N_crew)</td>
<td>8 crew</td>
</tr>
<tr>
<td>12</td>
<td>Average wage per person (A_wage)</td>
<td>2000 LE/month</td>
</tr>
<tr>
<td>13</td>
<td>Fuel price (F_price)</td>
<td>2000 LE/ton</td>
</tr>
<tr>
<td>14</td>
<td>Specific fuel consumption (SFC)</td>
<td>150 gr/hp/h</td>
</tr>
<tr>
<td>15</td>
<td>Daily accommodating cost per person (C_day)</td>
<td>25 LE/day</td>
</tr>
<tr>
<td>16</td>
<td>Minimum specific cost (S_c_min)</td>
<td>0.35 LE/(ton.km)</td>
</tr>
<tr>
<td>17</td>
<td>Fleet required annual grain capacity (F_{RC})</td>
<td>0.5 million tons/year</td>
</tr>
</tbody>
</table>
### Table 5 Output results - developed program

<table>
<thead>
<tr>
<th>No.</th>
<th>Items</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Objective Function ((Sc))</td>
<td>0.285 LE/(ton.km)</td>
</tr>
<tr>
<td>2</td>
<td>Fleet Size ((N_S))</td>
<td>11 Ship</td>
</tr>
<tr>
<td>3</td>
<td>Length Over All ((L_oa))</td>
<td>67.8 m</td>
</tr>
<tr>
<td>4</td>
<td>Ship Breadth ((B))</td>
<td>11.75 m</td>
</tr>
<tr>
<td>5</td>
<td>Ship Depth ((D))</td>
<td>3.5 m</td>
</tr>
<tr>
<td>6</td>
<td>Ship Draft ((T))</td>
<td>1.5 m</td>
</tr>
<tr>
<td>7</td>
<td>Width of Side Tank ((B_{st}))</td>
<td>1.25 m</td>
</tr>
<tr>
<td>8</td>
<td>Double Bottom Height ((H_{db}))</td>
<td>1.0 m</td>
</tr>
<tr>
<td>9</td>
<td>Ship Speed ((V_S))</td>
<td>12.4 km/h</td>
</tr>
<tr>
<td>10</td>
<td>Air Clearance</td>
<td>1.0 m</td>
</tr>
<tr>
<td>11</td>
<td>Block Coefficient ((C_B))</td>
<td>0.889</td>
</tr>
<tr>
<td>12</td>
<td>Ship Capital Cost ((P))</td>
<td>6,521,139 LE</td>
</tr>
<tr>
<td>13</td>
<td>Ship Deadweight ((D_{wt}))</td>
<td>706.88 tons</td>
</tr>
<tr>
<td>14</td>
<td>Ship Lightweight ((W_{light}))</td>
<td>344 tons</td>
</tr>
<tr>
<td>15</td>
<td>Engine Brake Power ((P_B))</td>
<td>554.77 hp</td>
</tr>
<tr>
<td>16</td>
<td>Capacity of Fuel Tanks</td>
<td>30 m³</td>
</tr>
<tr>
<td>17</td>
<td>Capacity of Ballast Tanks</td>
<td>620 m³</td>
</tr>
</tbody>
</table>

#### 7. Conclusions

1. The global logistics center for the storage and handling of grain at Damietta port will extremely increase the annual movement of grain through Cairo-Damietta waterway and consequently, will increase River transport share in Egypt during the next years, instead of a half per cent today.

2. The developed computer program represents a tailored and simple tool to find out the optimum fleet size and dimensions of self-propelled grain bulk ships working between Cairo and Damietta through the River Nile. This program may be simply modified to suit not only the other navigation routes but also the other River Nile ship types.

3. The output results of the developed program may be taken as a standard dimensions for any new self-propelled grain bulk ship working through Cairo-Damietta waterway. Also, according to the characteristics of Cairo-Aswan waterway, this fleet can be navigated from Damietta to Aswan.

4. Increasing the sailing time per day through the River Nile, will decrease the cost of cargo transportation and consequently, will highly encourage the transportation companies to shift their activities to the River Nile transportation mode.
REFERENCES


NOMENCLATURE

AAC Average annual cost, LE/year
Awage Average wage per person, LE/month
B Ship breadth, m
Bst Width of side tank, m
BMt Transverse metacentric radius, m
CHR Cargo handling rate, ton/h
CR Capital recovery factor, “”
CB Block coefficient, “”
Cadmin Administration cost, LE/year
Cao Annual operating cost, LE/year
Cch Cargo handling cost, LE/year
Cday Accommodating cost, LE/day/person
Cfin Financing cost, LE/year
Cfuel Fuel cost, LE/year
Chot Cargo handling cost, LE/ton
Cinsu Insurance cost, LE/year
Cmar Maintenance and repair costs, LE/year
Cmc Machinery cost, LE
Cout Outfitting cost, LE
Cport Port expenses, LE/year
Csteel Hull steel cost, LE
Cvict Victualling cost, LE/year
Ctao Present worth of life span operating expenses, LE
CWL Water plan area coefficient, “"
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>C\textsubscript{wages}</td>
<td>Crew cost, LE/year</td>
</tr>
<tr>
<td>D</td>
<td>Ship depth, m</td>
</tr>
<tr>
<td>DC\textsubscript{nav}</td>
<td>Diesel oil consumption at sea, tons/trip</td>
</tr>
<tr>
<td>Dwt</td>
<td>Dead weight, tons</td>
</tr>
<tr>
<td>F\textsubscript{B}</td>
<td>Freeboard, m</td>
</tr>
<tr>
<td>F\textsubscript{RC}</td>
<td>Fleet required annual grain capacity, tons/year</td>
</tr>
<tr>
<td>F\textsubscript{price}</td>
<td>Fuel price, LE/ton</td>
</tr>
<tr>
<td>FC\textsubscript{nav}</td>
<td>Fuel consumption at sea, tons/trip</td>
</tr>
<tr>
<td>FC\textsubscript{port}</td>
<td>Fuel consumption at port, tons/trip</td>
</tr>
<tr>
<td>FC\textsubscript{trip}</td>
<td>Total fuel consumption, tons/trip</td>
</tr>
<tr>
<td>F\textsubscript{nh}</td>
<td>Froude depth number, “__”</td>
</tr>
<tr>
<td>f\textsubscript{port}</td>
<td>Port expenses, LE/ton</td>
</tr>
<tr>
<td>GM\textsubscript{T}</td>
<td>Transverse metacentric height, m</td>
</tr>
<tr>
<td>Hdb</td>
<td>Double bottom height, m</td>
</tr>
<tr>
<td>h\textsubscript{w}</td>
<td>Water depth, m</td>
</tr>
<tr>
<td>i</td>
<td>Interest rate, “__”</td>
</tr>
<tr>
<td>K\textsubscript{B}</td>
<td>Vertical center of buoyancy, m</td>
</tr>
<tr>
<td>K\textsubscript{G}</td>
<td>Vertical center of gravity, m</td>
</tr>
<tr>
<td>LE</td>
<td>Egyptian pound, “__”</td>
</tr>
<tr>
<td>Loa</td>
<td>Length over all, m</td>
</tr>
<tr>
<td>N</td>
<td>Ship life, years</td>
</tr>
<tr>
<td>Na</td>
<td>Nile allowance, “__”</td>
</tr>
<tr>
<td>Ns</td>
<td>Fleet size, ship</td>
</tr>
<tr>
<td>N\textsubscript{crew}</td>
<td>Number of crew, crew</td>
</tr>
<tr>
<td>N\textsubscript{trip}</td>
<td>Number of round trips per year, trip/year</td>
</tr>
<tr>
<td>n\textsubscript{p}</td>
<td>Loan period, years</td>
</tr>
<tr>
<td>P</td>
<td>Ship capital cost, LE</td>
</tr>
<tr>
<td>Pa</td>
<td>Port allowance, “__”</td>
</tr>
<tr>
<td>P\textsubscript{B}</td>
<td>Brake power, hp</td>
</tr>
<tr>
<td>R\textsubscript{b}</td>
<td>Bank annual payment, LE/year</td>
</tr>
<tr>
<td>Ru</td>
<td>Round trip distance, km</td>
</tr>
<tr>
<td>Sc</td>
<td>Specific cost, LE/(ton.km)</td>
</tr>
<tr>
<td>SFC</td>
<td>Specific fuel consumption, gr/hp/h</td>
</tr>
<tr>
<td>SHP</td>
<td>Shaft power, hp</td>
</tr>
<tr>
<td>Sv</td>
<td>Scrap value, LE</td>
</tr>
<tr>
<td>T\textsubscript{nav}</td>
<td>Sea time, days/trip</td>
</tr>
<tr>
<td>T\textsubscript{port}</td>
<td>Port time, days/trip</td>
</tr>
<tr>
<td>T\textsubscript{trip}</td>
<td>Total trip time, days</td>
</tr>
<tr>
<td>VO\textsubscript{d}</td>
<td>Operation time per year, days/year</td>
</tr>
<tr>
<td>V\textsubscript{S}</td>
<td>Ship speed, km/h</td>
</tr>
<tr>
<td>W\textsubscript{light}</td>
<td>Light ship weight, tons</td>
</tr>
<tr>
<td>W\textsubscript{m/c}</td>
<td>Machinery weight, tons</td>
</tr>
<tr>
<td>W\textsubscript{out}</td>
<td>Outfitting weight, tons</td>
</tr>
<tr>
<td>W\textsubscript{steel}</td>
<td>Net steel weight, tons</td>
</tr>
<tr>
<td>\Delta</td>
<td>Ship displacement, tons</td>
</tr>
</tbody>
</table>

Submitted: 15.08.2015.  
Accepted: 13.12.2015.
TURKISH SHIPBUILDING INDUSTRY – CHALLENGES AND POTENTIAL

Summary

Shipbuilding industry was developed and grown in the late 19th and early 20th centuries. The Turkish shipbuilding industry started to develop in an international level around 15 years ago with specialization in small tonnage vessels. The global economic crisis that started in 2008 has affected the enhancement of shipbuilding industry in Turkey. This study addresses the important factors affecting the development of Turkish shipbuilding industry by focusing on the global shipbuilding during the crisis and the current state of the industry with the application of a survey to the key decision makers. The data was obtained from the survey collected from the Turkish shipyards and maritime companies that represent the considerable share of the industry. The surveys were conducted between 2012 and 2013. The current state of shipbuilding industry concerning the most relevant factors for the development of the industry according to the respondents has been designated, and it was seen that the highlighted factors were lower in the current state of the industry. Moreover, the strongest and weakest factors for the Turkish shipbuilding industry were pointed out. Herein, general views of the respondents regarding the development of Turkish shipbuilding industry were presented.

Keywords: Turkish shipbuilding; shipyard; Turkey; shipbuilding; sectoral analysis; survey;

1. Introduction

The late 19th and early 20th centuries are the periods that the shipbuilding industry was developed and grown globally as well as all the other heavy industries. The development and competitiveness of the Turkish shipbuilding industry in international a manner was started 15 years ago with particular specialization in the building of small tonnage vessels such as chemical tankers and yachts (Celik, Erturk, & Turan, 2013). The global economic crisis also affected the shipbuilding industry in Turkey. However, the Turkish shipbuilding industry has an increasing leadership in the international trade of new ships that is a niche market that are mainly small tonnage chemical/oil tankers (up to 10 thousand deadweight (DWT) (OECD, 2011). The ship repair and conversion activities have been increasingly growing in the shipyards in contrast to the shipbuilding activities.
The general location of Turkish shipyards is shown in Fig.1. The shipyards are mainly located in Tuzla and Yalova regions in Turkey. The epicentre of shipbuilding activities in Turkey is Tuzla Bay, situated some 50 km east of Istanbul. Since Tuzla area is overcrowded, it can no longer offer suitable places for new yards. Some entrepreneurs have focused on nearby inland locations, such as Yalova-Altınova and İzmit. In particular, some enterprises in these nearby facilities specialise in the manufacture of individual hull blocks that are then transported to other shipyards, where they are assembled (OECD, 2011).

In addition, recently, the industry has been expanding beyond its traditional zone, and diversifying into new areas throughout Turkey, including Yalova, İzmit, the regions of Black and Mediterranean Sea. After Tuzla, the second largest shipbuilding location the Yalova-Altınova Shipyard Region was founded in 2004 in order to increase the capacity of Turkish yards to meet the growing demand for new vessel both nationally and internationally (OECD, 2011).

The Turkish shipbuilding industry has enhanced its capacity in production and export, including a significant product range in last decade when the global market tended to show development. According to the order books, Turkey has been in the top ten countries on the basis of its DWT production, and in the top five countries on the basis of the number of ships (OECD, 2011). Therefore, the Turkish shipbuilding also reflects accurately the members of the entire World shipbuilding industry. On the other hand, Turkey had 316,000 GT (Gross Tonnage) and 582,000 Compensated Gross Tonnes according to Order book by countries 1st July 2015 (SEAEurope, 2015).

In the literature, there are various studies regarding sectoral analysis. Some of those studies use input-output analysis while others employ surveys. Sartaş (Sartaş, 2010) analyzed the growth dynamics in Turkish commercial shipbuilding sector and its prospects between 1992-2008 and presented the reasons for the growth of the industry in Turkey. Savsar (Savsar, 1998) investigated the past, present and future of the maritime transportation sector in Saudi Arabia, and presented the current situation and forecasts on the future shipping capacity requirement. As a result, general steps that could be followed in analysing a country's maritime industry and its shipping fleet requirement have also been illustrated in
the study. Benito et al. (Benito, Berger, De la Forest, & Shum, 2003) analyzed maritime sector in Norway and they reported that Norwegian maritime sector is in a better situation compared to the other sectors. In the study of Managi (Managi, 2007), total productivity of Japan’s shipping industry was analyzed with the application of Luenberger productivity indicator and found large productivity increases in three significant maritime firms in Japan. They emphasized that although the shipping industry average is declining over the periods, the increasing trends of the maritime shipping firms in the study might reflect the restructuring of the organisation together with the adoption of new technologies. Cai et al. (Cai et al., 2009) made a sectoral analysis for international technology development coal-fired power generation, cement and aluminum sectors in China and compared the characteristics of these sectors as a result. Zakaria et al. (Zakaria, Iqbal, & Hossain, 2010) evaluated the shipbuilding industries in Bangladesh. They took an overall picture of shipbuilding industries in both public and private sector through an extensive review of the literature, field visits, interacting with shipyards and ship owners using structured, unstructured and open-ended questionnaires. According to the results of the study, the shipbuilding industry has been found as an attractive industry for Bangladesh with respect to long heritage and cheap labor costs. However, it is also pointed out that the productivity in shipbuilding should be upgraded since it is at the lowest level in the World.

Additionally, Zsuzsanna et al. (Zsuzsanna, Marian, & Sándor, 2014) investigated the ceramic sector in Romania by using a survey in order to find out the current situation of the enterprises, to see their current financial situation and to determine the importance given to competitiveness, flexibility, adaptability and reactivity by these companies. It was shown in the results that in order to improve the performances of companies, an increase of these mentioned parameters and efficient management is required. Alrashed and Asif (Alrashed & Asif, 2014) reflected the findings of a survey applied to architects, engineers, project managers, construction contractors, developer and investors in Saudi Building Industry in their study. The findings of the study according to survey results indicate that the industry has to be aware of the importance of sustainability, education and work experience. Hadžić et al. (Hadžić, Tomić, Vladimir, Ostojić, & Senjanović, 2015) carried out a study on the current state of the shipbuilding industry in Croatian and defined a significant role within national economy, and EU since it produces custom made and relatively complex products with the considerable perspective of their value and complexity enhancement. They gave recommendations for its future development in the results of the study.

Kwak et al. (Kwak, Yoo, & Chang, 2005) performed an input-output (I-O) analysis to the maritime industry in Korea and figured out the role of maritime industry in the national economy. They have concluded that maritime industry has a low forward linkage effect, a high backward linkage effect, a high production-inducing effect, a low supply shortage cost, a low pervasive effect of price changes and a high employment-inducing effect. Morrissey and O'Donoghue (Morrissey & O’Donoghue, 2013) also conducted a study on the role of marine sector in the Irish national economy by using an input-output methodology and pointed out that especially maritime transportation sector has a significant position in the economy of Ireland. Additionally, they also presented that the marine industry has a low forward linkage effect, a relatively high backward linkage effect, a high production-inducing effect and a high employment- inducing the effect.

Apart from above studies, Ćagalj (Ćagalj, 2009) suggested a new organizational approach for shipbuilding industry relying on scientifically based organizational theories in order to provide for decentralization, flexibility, innovativeness. Moreover, Luttenberger et al. (Runko Luttenberger, Ančić, & Šestan, 2013) analyzed the advantages of short-sea shipping (SSS) sector in Croatia and environmental concerns sourced by shipping. They undertook a strengths-weaknesses-opportunities-threats (SWOT) study with regard to the
strategy of SSS development in Croatia. As a consequence, suggestions for diminishing the weaknesses and threats have been given for ships, ports and maritime procedures.

In today's increasing competition among all the shipyards in the world, and depressed market conditions, improvement of Turkey's competitiveness may be possible with the determination of the current situation of the industry and taking precautions for the unsuccessful points. In this context, the key players in the Turkish shipbuilding industry were surveyed, and the results derived from the surveys were presented herein. To the best knowledge of the authors, this is the only study in the literature with such comprehensive analysis applied to shipbuilding industry. It is known that there are some reports both confidential and commercial prepared by private research companies. However, these are not public and intended for particular trade purpose. Therefore, this study provides important insights on challenges and potential of Turkish shipbuilding industry.

2. Methodology

The analysis in this study is based on the data from a survey that was applied to the key decision makers in the shipbuilding industry in Turkey to measure the importance of the development and current success of the industry. There are around 100 leading companies that 69 of them are the member of Turkish Shipbuilders’ Association ("Turkish Shipbuilders’ Association," 2015). These companies entirely represent the shipbuilding industry in Turkey and mainly located in the Tuzla and Yalova Shipyard Zones, as well as in Korfez (Izmit), Karadeniz Ereğlisi, Gelibolu, and Çanakkale (OECD, 2011). Other companies are marine equipment manufacturers and suppliers, classification societies, design companies and consultants. The surveys were conducted between 2012 and 2013 with the 47 of the 69 member companies that responded positively to participate this study.

The survey is divided into five sections exploring the development of the Turkish shipbuilding industry. In the first and second part of the survey, the respondents were asked about their position in the company, and how long they have worked in the shipbuilding industry. In the third part, 35 factors on the sectoral development of the industry were given to define both the importance of the development and current success of shipbuilding industry. The respondents were requested to rate the factors based on their level of optimism with a scale from 0 to 100, with 0 being not optimistic at factor and 100 being very optimistic. Moreover, in the fourth part, participants were asked to express their opinion on when the industry will attain the uptrend period again. The final section is comprised of open-ended question that is their opinion on the development of Turkish shipbuilding industry in the future.

3. Data Analysis

Data was analysed using SPSS predictive analytics software (SPSS 15.0). The variables were arranged based on the mean values, maximum - minimum values, ± standard deviation (SD), and frequency distributions. The differences between sectoral experiences of the respondents were determined using the one-way analysis of variance (ANOVA) test. The differences between positions in the company of the respondents were determined using an independent sample t-test. Using these tests, all the responses to the inquiries were statistically analysed for significant differences between the groups: type of position in the company, experience in the sector.

The reliability of all responses to the inquiries was checked with Cronbach’s Alpha. Participants were requested to rate inquiries in respect to both the importance of the development and current success of shipbuilding industry. Thus, the reliability statistics was given for these two groups. Responses to the development of shipbuilding industry
(Cronbach’s $\alpha = 0.967$) and for the current state of the shipbuilding industry (Cronbach’s $\alpha = 0.942$) has high internal reliability.

The sectoral experiences of the 47 the respondents are listed in Table 1. The majority of the respondents have more than 20 years of experience in the shipbuilding industry.

### Table 1 Sectoral experience of the respondents ($n = 47$).

<table>
<thead>
<tr>
<th>Experience (years)</th>
<th>Percent of the respondents (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>≤5</td>
<td>12.8</td>
</tr>
<tr>
<td>6-10</td>
<td>19.1</td>
</tr>
<tr>
<td>11-15</td>
<td>19.1</td>
</tr>
<tr>
<td>16-20</td>
<td>14.9</td>
</tr>
<tr>
<td>&gt;20</td>
<td>34.0</td>
</tr>
</tbody>
</table>

As shown in Fig. 2, 96% of the respondents are men and 4% of them are women. All respondents are key players being 49% of them are top management and 51% of them are qualified employees and consultants.

### Fig. 2 Descriptive statistics for the respondents

#### 4. Results and Discussion

The global crisis that started in 2008 affected the Turkish shipbuilding industry, and the growth of the industry decreased gradually. Thereby, the Turkish shipyards had difficulties to have new orders, and were struggling to continue their activities. When the respondents were asked “when the sector will attain the uptrend period again?”, 26% of the respondents stated that it was impossible; 29% of the respondents replied that it could be possible in a period of 3-5 years; the 23% replied that it could be possible in a period of 6-10 years; the 11% replied that it could be possible in a period of 11-15 years, and the 2% replied that it could be possible in a period of 0-2 years. Opinions of the respondents for the inquiry of when the sector will attain the uptrend period again have been illustrated in Fig. 3. The results obtained from the survey showed that the majority of the respondents do not think that the sector will attain the uptrend period in a short-term time zone. Rest of inquiries were asked to determine the important factors for the development and current state of the shipbuilding industry.
Fig. 3 Opinions of the respondents for the inquiry of when the sector will attain the uptrend period again

4.1 Results on the Important Factors for the Development of Shipbuilding Industry

The respondents were requested to give a grade to each of the 35 factors with a range from 0 as the lowest point to 100 as the highest point. The statistical data about the importance of the development of shipbuilding industry is given in Table 2. Large differences within the responses were found to be for three factors: (i) success in marketing, (ii) national customer loyalty, (iii) international customer loyalty (Fig. 4). It was seen that the respondents gave minimum 5, maximum 100 points to those three factors above. This large difference indicates that there are problems with marketing activities in the sector.

The respondents gave 93 points in average with a maximum of 100 points and a minimum of 40 points for the factor related human resources capability profile to which the highest average point was given. It shows that the respondents think that the human resource is sufficient.

Table 2 Descriptive statistics for the data of the factors that were graded between 0 and 100 by the respondents for importance of the factors for having a developed industry.

<table>
<thead>
<tr>
<th>Factors on Sectoral Development</th>
<th>Mean</th>
<th>Minimum</th>
<th>Maximum</th>
<th>Standard Deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>*Infrastructure of human resources capability profile</td>
<td>92.9</td>
<td>40.0</td>
<td>100.0</td>
<td>12.0</td>
</tr>
<tr>
<td>*Quality level of the national sector</td>
<td>88.5</td>
<td>50.0</td>
<td>100.0</td>
<td>14.3</td>
</tr>
<tr>
<td>*Success in delivery time of the products or services</td>
<td>88.3</td>
<td>0.0</td>
<td>100.0</td>
<td>21.5</td>
</tr>
<tr>
<td>*Opportunities and technological level in the industry</td>
<td>87.6</td>
<td>50.0</td>
<td>100.0</td>
<td>13.6</td>
</tr>
<tr>
<td>*Availability of the qualified managers</td>
<td>87.5</td>
<td>20.0</td>
<td>100.0</td>
<td>15.8</td>
</tr>
<tr>
<td>Incentives for global competitors</td>
<td>87.0</td>
<td>50.0</td>
<td>100.0</td>
<td>15.0</td>
</tr>
<tr>
<td>Effective collaboration with domestic suppliers</td>
<td>87.0</td>
<td>50.0</td>
<td>100.0</td>
<td>14.0</td>
</tr>
<tr>
<td>Competence of the domestic suppliers</td>
<td>87.0</td>
<td>50.0</td>
<td>100.0</td>
<td>15.0</td>
</tr>
<tr>
<td>Success in marketing</td>
<td>87.0</td>
<td>5.0</td>
<td>100.0</td>
<td>18.0</td>
</tr>
<tr>
<td>Global sectoral reputation and recognition</td>
<td>86.0</td>
<td>20.0</td>
<td>100.0</td>
<td>17.0</td>
</tr>
</tbody>
</table>
A t-test was used to compare the difference between answers of the top management and employees. Significant differences were found in four inquiries. The respondents were requested to give points for the factors of “availability of qualified foreman”. Top management gave 87 points in average; qualified employees and consultants gave 78 points in average for this inquiry. (μ_{manager} = 86.74, μ_{employee} = 77.92, p = 0.038). Top management also gave 76 points in average, qualified employees and consultants gave 89 points in average for the inquiry of “sectoral collaboration and partnerships”. (μ_{manager} = 75.65, μ_{employee} = 88.75, p = 0.042). Moreover, the top management gave 91 points in average, qualified employees and consultants gave 81 points in average for the inquiry of “price competition level of the national sector”. (μ_{manager} = 90.65, μ_{employee} = 80.83, p = 0.073). Additionally, top management gave 84 points in average, qualified employees and consultants gave 73 points in average for the inquiry of “updated software usage” (Fig. 4). (μ_{manager} = 83.70, μ_{employee} = 73.33, p = 0.090)

The ANOVA test was used to determine whether there are any significant differences between the means of sectoral experiences of the respondents. According to the sectoral experiences, statistically significant responses are found. Different points were given to the inquiry of “intersectoral collaboration” according to the sectoral experience. The respondents who have ≤5 years of experience gave 84 points in average, while the respondents who have 16-20 years of experience gave 51 points in average for this inquiry (μ_{≤5 years} = 84.16, μ_{6-10 years} = 71.66, μ_{11-15 years} = 68.33, μ_{16-20 years} = 51.42, μ_{>20 years} = 72.33, p = 0.097). On the other hand, the respondents who have ≤5 years of experience gave 79 points in average for the inquiry of “global leadership in the sector”, while the respondents
who have 16-20 years of experience gave 51 points in average for the same inquiry (Fig. 4). 
\( \mu \leq 5 \text{ years} = 79.17, \mu 6-10 \text{ years} = 76.67, \mu 11-15 \text{ years} = 78.89, \mu 16-20 \text{ years} = 47.29, \mu >20 \text{ years} = 74.69, p = 0.08 \)

![Diagram](image)

**Fig. 4** Main factors for the development of Turkish shipbuilding industry

### 4.2 Results on the Current State of Shipbuilding Industry

In the second part of the survey, the views of the respondents were asked regarding the current state of the shipbuilding industry. The respondents gave grades with a range from 0 as the lowest point to 100 as the highest point for the 36 inquiries. Table 3 shows descriptive statistics for the data of the factors that were graded between 0 and 100 for their development status in the current state of the industry. Moreover, strongest and weakest factors for Turkish shipbuilding industry have been given in Table 3 and Fig. 5.

![Table 3](image)

<table>
<thead>
<tr>
<th>Factors on Sectoral Development</th>
<th>Mean</th>
<th>Minimum</th>
<th>Maximum</th>
<th>Standard Deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td><em>Incentives for global competitors</em></td>
<td>69.4</td>
<td>0.0</td>
<td>100.0</td>
<td>33.2</td>
</tr>
<tr>
<td><em>Potential of the global competitors</em></td>
<td>67.3</td>
<td>0.0</td>
<td>100.0</td>
<td>32.2</td>
</tr>
<tr>
<td><em>Strength of the global competitors</em></td>
<td>61.1</td>
<td>0.0</td>
<td>100.0</td>
<td>33.6</td>
</tr>
<tr>
<td><em>Quality level of the national sector</em></td>
<td>58.6</td>
<td>20.0</td>
<td>100.0</td>
<td>21.3</td>
</tr>
<tr>
<td><em>Availability of the qualified foreman</em></td>
<td>57.9</td>
<td>10.0</td>
<td>100.0</td>
<td>20.2</td>
</tr>
<tr>
<td>Infrastructure of human resources capability profile</td>
<td>52.7</td>
<td>5.0</td>
<td>100.0</td>
<td>20.4</td>
</tr>
<tr>
<td>Capacity of the current facilities</td>
<td>51.8</td>
<td>5.0</td>
<td>100.0</td>
<td>23.6</td>
</tr>
<tr>
<td>Competence of the domestic suppliers</td>
<td>51.0</td>
<td>10.0</td>
<td>100.0</td>
<td>21.4</td>
</tr>
<tr>
<td>Effective collaboration with domestic suppliers</td>
<td>51.0</td>
<td>0.0</td>
<td>100.0</td>
<td>25.2</td>
</tr>
<tr>
<td>Price competition level of the national sector</td>
<td>49.9</td>
<td>10.0</td>
<td>100.0</td>
<td>22.9</td>
</tr>
<tr>
<td>Opportunities and technological level in the industry</td>
<td>49.3</td>
<td>10.0</td>
<td>100.0</td>
<td>22.2</td>
</tr>
<tr>
<td>Updated software usage</td>
<td>49.2</td>
<td>0.0</td>
<td>100.0</td>
<td>24.6</td>
</tr>
</tbody>
</table>
Results from a Survey

<table>
<thead>
<tr>
<th>Factor</th>
<th>Score (Top Management)</th>
<th>Score (Employees)</th>
<th>Difference</th>
<th>Significance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Customer focality</td>
<td>46.9</td>
<td>0.0</td>
<td>100.0</td>
<td>24.8</td>
</tr>
<tr>
<td>Availability of the qualified managers</td>
<td>45.3</td>
<td>10.0</td>
<td>100.0</td>
<td>20.4</td>
</tr>
<tr>
<td>Success in delivery time of the products or services</td>
<td>44.7</td>
<td>0.0</td>
<td>100.0</td>
<td>26.1</td>
</tr>
<tr>
<td>Product/service differentiation</td>
<td>44.1</td>
<td>0.0</td>
<td>100.0</td>
<td>26.0</td>
</tr>
<tr>
<td>Global sectoral reputation and recognition</td>
<td>43.6</td>
<td>0.0</td>
<td>100.0</td>
<td>21.1</td>
</tr>
<tr>
<td>Availability of financial instruments in the sector</td>
<td>42.1</td>
<td>10.0</td>
<td>90.0</td>
<td>20.5</td>
</tr>
<tr>
<td>Guarantee letters</td>
<td>41.2</td>
<td>10.0</td>
<td>80.0</td>
<td>19.0</td>
</tr>
<tr>
<td>Success in marketing</td>
<td>39.8</td>
<td>0.0</td>
<td>100.0</td>
<td>23.8</td>
</tr>
<tr>
<td>Dynamism of the global markets</td>
<td>39.5</td>
<td>0.0</td>
<td>100.0</td>
<td>21.4</td>
</tr>
<tr>
<td>Loyal customer’s purchasing power</td>
<td>37.9</td>
<td>0.0</td>
<td>100.0</td>
<td>24.2</td>
</tr>
<tr>
<td>National customer loyalty</td>
<td>36.4</td>
<td>0.0</td>
<td>90.0</td>
<td>23.7</td>
</tr>
<tr>
<td>Management information systems</td>
<td>36.1</td>
<td>0.0</td>
<td>100.0</td>
<td>22.8</td>
</tr>
<tr>
<td>Collaboration with the universities</td>
<td>35.6</td>
<td>0.0</td>
<td>75.0</td>
<td>20.1</td>
</tr>
<tr>
<td>Global follower in the shipbuilding market</td>
<td>33.5</td>
<td>0.0</td>
<td>100.0</td>
<td>24.7</td>
</tr>
<tr>
<td>International customer loyalty</td>
<td>31.3</td>
<td>0.0</td>
<td>75.0</td>
<td>18.3</td>
</tr>
<tr>
<td>Intersectoral collaboration</td>
<td>30.6</td>
<td>0.0</td>
<td>100.0</td>
<td>23.4</td>
</tr>
<tr>
<td>Incentives and legislation</td>
<td>29.2</td>
<td>0.0</td>
<td>80.0</td>
<td>21.6</td>
</tr>
<tr>
<td>Effective lobbies for local and international sectorial trade</td>
<td>27.8</td>
<td>0.0</td>
<td>75.0</td>
<td>18.9</td>
</tr>
<tr>
<td>**Global leadership in the sector</td>
<td>27.6</td>
<td>0.0</td>
<td>80.0</td>
<td>21.1</td>
</tr>
<tr>
<td>**Sectoral collaboration and partnerships</td>
<td>26.8</td>
<td>0.0</td>
<td>75.0</td>
<td>22.1</td>
</tr>
<tr>
<td>**Effective consultancy utilization</td>
<td>25.5</td>
<td>0.0</td>
<td>60.0</td>
<td>17.0</td>
</tr>
<tr>
<td>**Availability of sectoral strategies and plans</td>
<td>25.4</td>
<td>0.0</td>
<td>100.0</td>
<td>24.4</td>
</tr>
<tr>
<td>**The level of the research and development activities</td>
<td>22.2</td>
<td>0.0</td>
<td>50.0</td>
<td>17.5</td>
</tr>
</tbody>
</table>

* High-scored factors which represent strong and weak points in the Turkish shipbuilding industry.

**Low-scored factors which represent weak points in the Turkish shipbuilding industry.

A $t$-test was performed to compare the difference between the responses of top management and employees. Significant differences were found in some inquiries and shown in Fig. 5.

According to $t$-test results, top management gave 33 points in average, qualified employees and consultants gave 20 points in average for the inquiry of “availability of sectoral strategies and plans”. ($\mu_{\text{manager}} = 33.41$, $\mu_{\text{employee}} = 18.04$, $p = 0.037$). Additionally, top management gave 71 points in average, qualified employees and consultants gave 52 points in average for the inquiry of “strength of global competitors”. ($\mu_{\text{manager}} = 70.91$, $\mu_{\text{employee}} = 52.17$, $p = 0.058$)

The ANOVA test was used to determine whether there are any significant differences between the means of the sectoral experiences of the respondents. As regards to sectoral experience, statistically significant responses are listed in Table 4 and shown in Fig. 5.
Table 4 Statistically significant relations between the sectoral experience and factors.

<table>
<thead>
<tr>
<th>Factors on the Sectoral Development</th>
<th>Mean of the Factors</th>
<th>Significance (p)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Work Experience (years)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>≤5</td>
<td>6-10</td>
</tr>
<tr>
<td>Intersectoral collaboration</td>
<td>58.3</td>
<td>24.4</td>
</tr>
<tr>
<td>Global leadership in the sector</td>
<td>44.2</td>
<td>25.0</td>
</tr>
<tr>
<td>Price competition level of the national sector</td>
<td>65.0</td>
<td>36.7</td>
</tr>
</tbody>
</table>

Fig. 5 Main factors for the current state of Turkish shipbuilding industry

The ANOVA test was also conducted to determine whether there are any significant differences in the inquiry of the expectations of “attaining the uptrend period again” with all other inquiries. According to expectations, statistically significant answers are showed Table 5 below.

Table 5 Statistically significant relations between the expectations and factors.

<table>
<thead>
<tr>
<th>Factors on Sectoral Development</th>
<th>Expectations of Attaining Uptrend Period</th>
<th>Significance</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Unpredictable</td>
<td>In 0-2 years</td>
</tr>
<tr>
<td>Global leadership in the sector</td>
<td>15.0</td>
<td>10.0</td>
</tr>
<tr>
<td>Success in delivery time of products or services</td>
<td>39.2</td>
<td>10.0</td>
</tr>
<tr>
<td>Product/service differentiation</td>
<td>30.0</td>
<td>10.0</td>
</tr>
<tr>
<td>Potential of global competitors</td>
<td>50.8</td>
<td>10.0</td>
</tr>
</tbody>
</table>

4.3 Comparison of the Results of the Survey

The evaluation of the current state of the shipbuilding industry concerning the five top graded factors for the development of the industry according to the respondents was indicated with an asterisk in Table 2. The significance of human resources, quality, on-time delivery, technology usage and qualified managers has been highlighted for the development of the Turkish shipbuilding industry. However, these highlighted factors were found to be lower in the current state of the industry. According to the results, the industry should set up
strategic plans for improving the position of the industry as well as for competing with the leading shipbuilder countries.

The high-graded factors that represent the strength and weakest issues of the Turkish shipbuilding industry were indicated with an asterisk in Table 3. According to the results of the survey, the respondents think that international competitors have incentives and opportunities compared to their current state in the industry. Although the quality level of the national sector and foreman is high, the deficiency of incentives related to the shipbuilding industry is seen as a threat for having new projects. The incentives facilitate the global competitors to be more powerful during the tendering stage.

The low-scored factors that represent weakness issues of the Turkish shipbuilding industry were indicated with a double asterisk in Table 3. The answers of the respondents indicate that the level of Turkish shipbuilding industry for global leadership is relatively low. Furthermore, it was inferred from the respondents that the industry does not have strategic plans, future directions, and actions. The industry leaders should attach a particular importance for spectacular success.

4.4 Comments and Opinions of the Respondents for the Development of Turkish Shipbuilding Industry

In the survey, the respondents were asked to share their opinions and suggestions regarding the development of Turkish shipbuilding industry. General opinions and suggestions were presented below:

- Joint ventures of the shipyards should be encouraged for specialisation and costs should be reduced,
- Value-added vessels should be built and tailor-made production should be adopted,
- The global market should be followed,
- Companies should get into new markets with the adoption of R&D,
- Actual reports should be prepared comprising of capacity, characteristics of the employee, costs and steel processing capacity of the industry, efficient use of available sources,
- Financial instruments should be diversified and access to foreign sources should be improved,
- Government incentives for the shipbuilding industry should be increased,
- Development plans of shipbuilding industry should be prepared,
- Shipbuilding should be adopted as a national policy,
- Local contribution should be increased in the shipbuilding processes,
- Leading countries should be taken as a role model,
- More professional staff should be employed,
- Investments should be made on qualified and experienced employee,
- Technology usage should be increased,
- Marketing activities should be increased,
- Tax reduction should be sustained,
- Strategic alliance of university and industry should be improved,
- Companies should adopt institutionalization,
- Quality should be improved and companies should be customer-focused,
- Flexible production should be established,
- The sub-industry should be strengthened,
- A platform should be formed having representatives from all maritime companies.

5. Conclusion

This study addressed the important factors for the development of the Turkish shipbuilding industry and the current state of the industry. Accordingly, the key decision makers were surveyed between 2012 and 2013. The results indicated that human resources capability profile was an advantage for the development of the shipbuilding industry. In contrast, intersectoral collaboration was a weakness for the sectoral development. The top management and qualified employees and consultants gave statistically significant answers to the inquiry of availability of qualified foreman, sectoral collaboration and partnerships, price competition level of the domestic sector and updated software usage. Besides, the sectoral experience was found statistically significant for the inquiry of intersectoral collaboration and global leadership in the shipbuilding market.

The respondents expressed their opinion that incentives for the global competitors were higher than their current state, and this situation was a risk factor for new business contracts. On the other hand, the level of research and development activities was specified as a weakness for the current state of the industry. The responses of top management, and consultants and qualified employee were statistically significant for the inquiry of sectoral strategic plans and power of global competitors. Moreover, the sectoral experiences of the respondents were statistically significant for the inquiry of intersectoral collaboration, global leadership in the shipbuilding market and, price competition level of the domestic sector.

This study showed that the majority of the respondents do not believe the sector will attain uptrend period in a short-term time period. The respondents think that the level of the Turkish shipbuilding industry for global leadership is quite low, and the industry does not have strategic plans for the future. Therefore, this issue should be considered for the future development of industry, and shipbuilding companies need to take precautions to achieve success.

Acknowledgements

The authors thank Mr. Osman Kaya Turan, a professional manager in the Turkish shipbuilding industry, for the data collection and his contributions to this study.

REFERENCES

Results from a Survey


Appendix A. A Survey to Development of Turkish Shipbuilding Industry

1. What is your position in the company?
   - ☐ Manager
   - ☐ Employee

2. How long have you been working in the shipbuilding industry?
   - ☐ ≤5 years
   - ☐ 6-10 years
   - ☐ 11-15 years
   - ☐ 16-20 years
   - ☐ >20 years

3. Please grade the importance of the development and current success of shipbuilding industry between 0 and 100 where 0 is the minimum point, and 100 is the maximum point.

<table>
<thead>
<tr>
<th>Factors on the Sectoral Development</th>
<th>Importance of the Development of Shipbuilding Industry</th>
<th>Current Success of Shipbuilding Industry</th>
</tr>
</thead>
<tbody>
<tr>
<td>Infrastructure of human resources</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Availability of qualified managers</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Availability of qualified foreman</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dynamism of the global market</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Availability of financial instruments in the sector</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Guarantee letters</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Incentives and legislation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sectoral collaboration and partnerships</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Effective lobbies for local and international sectoral trade</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Global sectoral reputation and recognition</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Availability of sectoral strategies and plans</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Intersectoral collaboration</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Capacity of the current facilities</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Success in marketing</td>
<td></td>
<td></td>
</tr>
<tr>
<td>National customer loyalty</td>
<td></td>
<td></td>
</tr>
<tr>
<td>International customer loyalty</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Loyal customer’s purchasing power</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Customer focality</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Global leadership in the sector</td>
<td></td>
<td></td>
</tr>
<tr>
<td>--------------------------------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Global follower in the shipbuilding market</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Strength of global competitors</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Potential of the global competitors</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Incentives for global competitors</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Quality level of the national sector</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Price competition level of the national sector</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Success in delivery time of products or services</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Product/service differentiation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Opportunities and technological level in the industry</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Management information systems</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Updated software usage</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Competence of domestic suppliers</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Effective collaboration with domestic suppliers</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Collaboration with the universities</td>
<td></td>
<td></td>
</tr>
<tr>
<td>The level of research and development activities</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Effective consultancy utilization</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

4. When do you think the industry will attain the uptrend period again?

- Unpredictable
- In 0-2 years
- In 3-5 years
- In 6-10 years
- In 11-15 years
- Never

5. What is your opinion about the development of Turkish shipbuilding industry?
Aims and scopes of journal Brodogradnja

The journal is devoted to multidisciplinary researches in the fields of theoretical and experimental naval architecture and oceanology as well as to challenging problems in shipping, shipbuilding, offshore and related industries worldwide. The aim of the journal is to integrate interests in shipbuilding, ocean engineering, sea and ocean shipping, inland navigation and intermodal transportation as well as environmental issues, overall safety, wind, marine and hydrokinetic renewable energy production and sustainable transportation development at seas and oceans. The journal focuses on hydrodynamics, structures, reliability, materials, construction, design, optimization, production engineering, building and organization of building, project management, repair and maintenance planning, information systems in shipyards, quality assurance as well as outfitting, powering, power plants and equipment onboard.

https://www.fsb.unizg.hr/brodogradnja/
EDITORIAL BOARD

Editor in Chief
Professor emeritus Kalman Žiha (Department of Naval Architecture and Ocean Engineering) kziha@fsb.hr

Editor of Sciences
Professor Nastia Degiuli (Department of Naval Architecture and Ocean Engineering) nastia.degiuli@fsb.hr

Assistant Editor
PhD Student Ivana Martić (Department of Naval Architecture and Ocean Engineering) ivana.martic@fsb.hr

Editorial Board Secretary
Nikolina Zmijarević Đugum (Department of Naval Architecture and Ocean Engineering) nzmijarevic@fsb.hr

Advisor for Bibliographic Databases and Libraries
Tamara Krajna, PhD Head librarian (Faculty of Mechanical Engineering and Naval Architecture tkrajna@fsb.hr)

Senior Editors (Department of Naval Architecture and Ocean Engineering)
Professor Nastia Degiuli nastia.degiuli@fsb.hr (197130)
Professor Joško Parunov jparunov@fsb.hr (206782)
Professor Ante Šestan ante.sestan@fsb.hr (155886)
Associate Professor Smiljko Rudan smiljko.rudan@fsb.hr (216136)
Associate Professor Jerolim Andrić jandric@fsb.hr (219630)
Associate Professor Vedran Slapničar vslapnic@fsb.hr (203716)
Assistant Professor Ivan Ćatipović, PhD ivan.catipovic@fsb.hr (275224)
Assistant Professor Neven Hadžić, PhD neven.hadzic@fsb.hr (320461)
Assistant Professor Nikola Vladimir, PhD nikola.vladimir@fsb.hr (305872)
Assistant Professor Pero Prebeg, PhD pero.prebeg@fsb.hr (257590)
Professor emeritus academician Ivo Senjanovic, PhD ivo.senjanovic@fsb.hr (57053)

Senior Advisors (Department of Naval Architecture and Ocean Engineering)
Retired professor Andrea Werner andreja.werner@fsb.hr (54173)
Retired professor Izvor Grubišić izvor.grubisic@fsb.hr (76234)
Retired professor Vedran Žanić izvor.grubisic@fsb.hr (76234)
Retired professor Rajko Grubišić izvor.grubisic@fsb.hr (76234)
Retired professor Večeslav Čorić izvor.grubisic@fsb.hr (76234)

External Editors (Faculty of Mechanical Engineering and Naval Architecture)
Professor Ivan Juraga (FSB) Zagreb ijuraga@fsb.hr
Professor Joško Petrić (FSB) Zagreb josko.petric@fsb.hr
Professor Neven Duič (FSB) Zagreb neven.duic@fsb.hr
Professor Nedeljko Štefanić (FSB) Zagreb nedeljko.stefanic@fsb.hr
Professor Zvonimir Guzović (FSB) Zagreb zvonimir.guzovic@fsb.hr
Professor Gojko Magazinović (FESB) Split gojko.magazinovic@fesb.hr
Associate Professor Dražen Lončar (FSB) Zagreb drazen.loncar@fsb.hr
Associate Professor Severino Krizmanić (FSB) Zagre severino.krizmanic@fsb.hr
Assistant professor Ivan Stojanović, PhD (FSB) Zagreb ivan.stojanovic@fsb.hr

External Editors (Others)
Professor Josip Brnić (RITEH, Rijeka) josip.brnic@riteh.hr (148822)
Professor Jasna Prpić-Oršić (RITEH, Rijeka) jasnapo@riteh.hr (211853)
Professor Roko Dejhalla (RITEH, Rijeka) roko@riteh.hr (161532)
Professor Albert Zamarin (RITEH, Rijeka) zamarin@riteh.hr (229065)
Professor Roko Markovina (FESB, Split) roko@fesb.hr (207162)
Professor Branko Blagovjević (FESB, Split) bblag@fesb.hr (218434)
Professor Zoran Vukić (FER, Zagreb) Zoran.Vukic@fer.hr (74412)
Professor Vladimir Andročec (GRAD, Zagreb) androcec@grad.hr (96701)

International Editorial Scientific Advisory Board
Professor Carlos Guedes Soares (Instituto Superior Tecnico, Lisbon) guedess@mar.ist.utl.pt
Professor Jose Manuel Gordo (Instituto Superior Tecnico, Lisbon) jose.gordo@ist.utl.pt
Professor Angelo Palos Teixeira (Instituto Superior Tecnico, Lisbon) teixeira@mar.ist.utl.pt
Professor Galal Younis (University of Suez Canal, Port Fouad, Port Said) dr.galal@gyounis.net
Professor Jerzy Matusiak (Ship Laboratory, Helsinki) Jerzy.Matusiak@tkk.fi
Anatolij Branko Togunjac, PhD (Institut Gibrorybflot St. Petersburg, Russia) togunjac@grf.spb.ru
Professor Ahmet Dursun Alkan (Yildiz Technical University, Istanbul) alkanad@yildiz.edu.tr
Professor Omer Goren (Istanbul Technical University) ogoren@itu.edu.tr
Professor Radoslav Nabergoj (University of Trieste) nabergoj@units.it
Professor Giorgio Trincas (University of Trieste) trinkas@units.it
Professor Igor Zotti (University of Trieste) zotti@units.it
Professor Alberto Francescutto (University of Trieste) francesc@units.it
Professor Ermina Begović (University of Naples Federico II) begovic@unina.it
Professor Šime Malenica (Bureau Veritas, Marine Division, Paris) sime.malenica@bureauveritas.com
Professor Dae Seung Cho, (Pusan National University, Busan, Korea) daecho@pusan.ac.kr
Professor Sun Hong Kwon (Pusan National University, Busan, Korea) shkwon@pusan.ac.kr
Byung Ki Choi, PhD (Hyundai Heavy Industries, Research Department, Ulsan, Korea) bkchoi@hhi.co.kr
Dongsheng Qiao, PhD (Dalian University of Technology, Dalian, China) gds903@163.com; qiaods@dlut.edu.cn
Tino Stanković, PhD (ETH, Zurich) tinos@ethz.ch
Professor Prasanta K. Sahoo, PhD (Department of Marine and Environmental Systems, 150 W University Boulevard, Melbourne, FL 32901, USA) psahoo@fit.edu
Professor Carlo Francesko Mario Bertrorello, PhD (Department of Industrial Engineering Marine Section – Via Claudio 19, 80125 Napoli, Italy) bertorel@unina.com
List of reviewers in 2015

Professor Gojko Magazinović
Professor Mladen Šercer
Professor Tatjana Haramina
Professor Peter Vidmar
PhD Michael Puskar
Professor Jamel Bessrour
Nikola Brnardić
Professor Zoran Nikolić
Professor Roko Dejhalla
PhD Sandro Ianniello
PhD Ali Hosseini Kordkheili
Professor Nikša Fafandjel
Professor Duško Pavletić
Professor Večeslav Čorić
PhD Alexander Phillips
PhD Ankica Đukić
Professor Ahmed Elshazly
Professor Ahmed Ibrahim
Professor Dario Bruzzone
Professor Tin Matulja
PhD Sara Muggiasca
Professor Jasna Prpić-Oršić
Professor Boris Ljubenkov
Professor Neven Hadžić
Professor Stefano Gaggero
Professor Hamid Mehdigholi
Professor Jorge Freiria
Professor Cristobal Mariscal Diaz
Professor Ivo Džijan
Professor Zoran Mrša
Professor Miguel Angel Celis Carbajal
PhD Damir Kolić
Professor Nenad Vulić
Professor Nikola Vladimir
Professor Radoslav Nabergoj
Professor Ivan Čatipović
PhD Sa-Young Hong
Professor N. M. Golam Zakaria
Professor Serdo Kos
Professor Zvonimir Lušić
Professor Prasanta Sahoo
Professor Mashud Karim
Professor Xianwu Luo
Professor Michele Viviani
Professor Serkan Ekinci
Professor Sun Hong Kwon
Professor Gregory Grigoropoulos
Professor Jose Manuel Gordo