About journal BRODOGRADNJA(SHIPBUILDING)

Indexed by Web of Sciences (WoS) Science Citation Index Expanded since 2008. 2019 Journal Citation Reports® Science Edition impact factor 0.968 (5 year 0.926) ranked Q3 10th/14 in category Marine Engineering.

2018 SCImago ranking Q2 in Mechanical Engineering and Q2 in Ocean Engineering SJR=0.399 H Index=11, SNIP=0.941.

Journal BRODOGRADNJA(SHIPBUILDING) was launched in 1950 as an expression of growing enthusiasm and ambition for promotion of the shipping and shipbuilding tradition of Croatia.

Its primary focus was on research and development serving the fast developing shipbuilding industry.

The permanently growing shipbuilding industry took over the publishing of BRODOGRADNJA in 1981.

In 1982 BRODOGRADNJA became a bilingual international journal, i.e. contributions were written either in Croatian or English.

The Croatian shipbuilding research institute — Brodarski Institute became the publisher of BRODOGRADNJA in 1992.

During this successful period of growth, BRODOGRADNJA covered the following regularly scheduled sections: peer reviewed scientific and professional papers, current topics and news from the shipbuilding industry, reviews, and worldwide maritime news.

Since the beginning of the year 2014 BRODOGRADNJA has been edited and published by the Faculty of Mechanical Engineering and Naval Architecture of the University of Zagreb.

BRODOGRADNJA is now an international scientific peer reviewed open-access journal with focus on the Theory and Practice of Naval Architecture, Marine Engineering and Ocean Engineering serving the Shipbuilding, Shipping and Offshore Industry.


BRODOGRADNJA(SHIPBUILDING) has been an open access journal from its early beginnings.

Papers are available free of charge to everyone in accordance with the Budapest Open Access Initiative from 2002 and Croatian declaration on open access to scientific information from the year 2012.

Open and fully free access is particularly supported after the inclusion of BRODOGRADNJA into the Central Portal of Croatian Scientific Journals called Hrčak (Hamster) in 2005 and following the launching of the Web edition of BRODOGRADNJA in 2007.

Publisher copyright policies & self-archiving (green policies) information is available here http://www.sherpa.ac.uk/romeo/pub/2294/ and is searchable in searching the RoMEO database - http://www.sherpa.ac.uk/romeo.

Copyright policy of BRODOGRADNJA – authors are permitted to retain copyright and publishing rights without restrictions.

Manuscripts are licenced under Creative Commons Attribution CC BY 3.0 Unported Licence.

BRODOGRADNJA(SHIPBUILDING) is in Directory of Open Access Journals (DOAJ) since 2015.

Faculty of Mechanical Engineering and Naval Architecture of the University of Zagreb is the member of Publisher International Linking Association (PILA) using CROSSREF services Digital Object Identifiers DOI:10.21278/brod.

Journal Brodogradnja is the member of the Committee on Publishing Ethics (COPE) since 2017.

Journal Brodogradnja is funded by the Ministry of Science and Education of the republic of Croatia as well as by the Faculty of Mechanical Engineering and Naval Architecture of the University of Zagreb.

BRODOGRADNJA learns from experienced scientists and experts and promotes young researchers and engineers.
<table>
<thead>
<tr>
<th>Title</th>
<th>Pages</th>
<th>Type</th>
<th>Authors</th>
</tr>
</thead>
<tbody>
<tr>
<td>LIFE CYCLE COST ANALYSIS FOR THE YAW DAMPING SYSTEM OF A WARSHIP</td>
<td>1-9</td>
<td>Professional</td>
<td>Uğur Buğra Çelebi, Levent Bilgili, Bilkutay Yılmaz</td>
</tr>
<tr>
<td>FROM A FINANCIAL VIEWPOINT</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>FATIGUE ASSESSMENT OF CRACK GROWTH BASED ON FAILURE ASSESSMENT</td>
<td>11-24</td>
<td>Original</td>
<td>Shilun Zhao, Chaohe Chen, Tianhui Fan</td>
</tr>
<tr>
<td>DIAGRAMS FOR A SEMI-SUBMERSIBLE PLATFORM</td>
<td></td>
<td>scientific</td>
<td></td>
</tr>
<tr>
<td>NUMERICAL STUDY ON THE PREDICTION OF THE BOW FLARE SLAMMING PRESSURE</td>
<td>25-42</td>
<td>Original</td>
<td>Daewon Seo, Kwang-Leol Jeong</td>
</tr>
<tr>
<td>FOR THE CONTAINER SHIP IN REGULAR WAVE</td>
<td></td>
<td>scientific</td>
<td></td>
</tr>
<tr>
<td>NUMERICAL SIMULATION OF THE CAVITATING FLOW AROUND MARINE CO-</td>
<td>43-57</td>
<td>Original</td>
<td>Boucetta Djahida, Imine Omar</td>
</tr>
<tr>
<td>ROTATING TANDEM PROPELLERS</td>
<td></td>
<td>scientific</td>
<td></td>
</tr>
<tr>
<td>EXERGY ANALYSIS OF THE MAIN PROPULSION STEAM TURBINE FROM MARINE</td>
<td>59-77</td>
<td>Original</td>
<td>Vedran Mrzljak, Igor Poljak, Jasna Prpić-Oršić</td>
</tr>
<tr>
<td>PROPULSION PLANT</td>
<td></td>
<td>scientific</td>
<td></td>
</tr>
<tr>
<td>AN ALTERNATIVE SOLUTION FOR INTERACTION OF FLEXURAL GRAVITY WAVES</td>
<td>79-91</td>
<td>Original</td>
<td>Šime Malenica</td>
</tr>
<tr>
<td>WITH BOTTOM MOUNTED CIRCULAR CYLINDER</td>
<td></td>
<td>scientific</td>
<td></td>
</tr>
<tr>
<td>A FUZZY EVALUATION MODEL OF CHOOSING A MIDDLE MANAGER FOR AN</td>
<td>93-107</td>
<td>Professional</td>
<td>Ji-Feng Ding, Jung-Fong Kuo, Wen-Hui Tai</td>
</tr>
<tr>
<td>INTERNATIONAL SHIPPING SERVICE PROVIDER</td>
<td></td>
<td>paper</td>
<td></td>
</tr>
<tr>
<td>NUMERICAL SIMULATION OF ICE MILLING LOADS ON PROPELLER BLADE WITH</td>
<td>109-128</td>
<td>Original</td>
<td>Feng Wang, Zao-Jian Zou, Zhou Li, Yang Wang, Hao Yu, Haihua Zhang</td>
</tr>
<tr>
<td>COHESIVE ELEMENT METHOD</td>
<td></td>
<td>scientific</td>
<td></td>
</tr>
<tr>
<td>REEFER VESSEL VERSUS CONTAINER SHIP</td>
<td>129-141</td>
<td>Professional</td>
<td>Predrag Ćudina, Ana Bezić</td>
</tr>
</tbody>
</table>
NUMERICAL AND EXPERIMENTAL CALCULATION OF ROLL AMPLITUDE EFFECT ON ROLL DAMPING (str.1-15)
Burak Yıldız, Bekir Şener, Ahmet Yurtseven, Toru Katayama
Original scientific paper

EXPERIMENTAL STUDY OF WELDING RESIDUAL STRESS OF HIGH-STRENGTH SHIPBUILDING STEEL (str.17-32)
Yongjin Guo, Luyun Chen, Hongdong Wang, Hong Yi
Professional paper

WATER ENTRY HYDROELASTICITY ANALYSIS OF LATTICE SANDWICH PANEL WITH IMPERFECTION: SIMULATION AND ENGINEERING MODEL (str.33-59)
Wang Hao, Cheng Yuan-Sheng, Pei Da-Ming, Hao Wei-Wei, Gan Lin
Professional paper

NUMERICAL SIMULATIONS OF ADDED RESISTANCE IN REGULAR HEAD WAVES ON A CONTAINER SHIP (str.61-86)
Young-Gill Lee, Cheolho Kim, Jeong-Ho Park, Hyeongjun Kim, Insu Lee, Bongyong Jin
Original scientific paper

RETROFITTING ANALYSIS OF TANKER SHIP HULL STRUCTURE SUBJECTED TO CORROSION (str.87-109)
Davide Chichi, Yordan Garbatov
Original scientific paper

DESIGN AND MODEL TEST OF STRUCTURAL MONITORING AND ASSESSMENT SYSTEM FOR TRIMARAN (str.111-134)
Haoyun Tang, Huilong Ren, Qi Zhong
Original scientific paper

ANOTHER BLOW ON THE TORN DOWN WALL-THE INCLINING EXPERIMENT (str.135-153)
Selahattin Ozsayan, Metin Taylan
Review article

PISTON RING MATERIAL IN A TWO-STROKE ENGINE WHICH SUSTAINS WEAR DUE TO CATALYST FINES (str.155-169)
Miroslav Vukičević, Nikola Račić, Špiro Ivošević
Original scientific paper

A LID APPROACH FOR PREDICTING WAVE INDUCED MOTIONS OF TRIMARAN IN REGULAR WAVES (str.171-185)
Zongyu Jiang, Yun Gao, Jie Liu
Professional paper
<table>
<thead>
<tr>
<th>Title</th>
<th>Pages</th>
<th>Authors</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>DETERMINATION OF AN OPTIMAL FLEET FOR A CNG TRANSPORTATION SCENARIO IN THE MEDITERRANEAN SEA</td>
<td>str.1-23</td>
<td>Francesco Mauro, Luca Braidotti, Giorgio Trincas</td>
<td>Original scientific paper</td>
</tr>
<tr>
<td>EFFECT OF BREAKING WAVE SHAPE ON IMPACT LOAD ON A MONOPILE STRUCTURE</td>
<td>str.25-42</td>
<td>Duje Veić, Wojciech Sulisz, Rohan Soman</td>
<td>Original scientific paper</td>
</tr>
<tr>
<td>SIMPLIFIED AND ADVANCED APPROACHES FOR EVACUATION ANALYSIS OF PASSENGER SHIPS IN THE EARLY STAGE OF DESIGN</td>
<td>str.43-59</td>
<td>Carlo Nasso, Serena Bertagna, Francesco Mauro, Alberto Marinò, Vittorio Bucci</td>
<td>Professional paper</td>
</tr>
<tr>
<td>ENVIRONMENTAL AND COST-EFFECTIVENESS COMPARISON OF DUAL FUEL PROPULSION OPTIONS FOR EMISSIONS REDUCTION ONBOARD LNG CARRIERS</td>
<td>str.61-77</td>
<td>Nader R. Ammar</td>
<td>Review article</td>
</tr>
<tr>
<td>EVALUATION MODEL OF MARINE POLLUTION BY WASTEWATER FROM CRUISE SHIPS</td>
<td>str.79-92</td>
<td>Tina Perić, Vice Mihanović, Nikola Račić</td>
<td>Original scientific paper</td>
</tr>
<tr>
<td>STUDY OF CONTINUOUS ICEBREAKING PROCESS WITH COHESIVE ELEMENT METHOD</td>
<td>str.93-114</td>
<td>Feng Wang, Li Zhou, Zao-Jian Zou, Ming Song, Yang Wang, Yi Liu</td>
<td>Original scientific paper</td>
</tr>
<tr>
<td>VEHICLE SECURING SAFETY ASSESSMENTS OF A KOREAN COASTAL CAR FERRY ACCORDING TO ACCELERATION PREDICTION APPROACHES</td>
<td>str.115-131</td>
<td>Joonmo Choung, Se-Min Jeong</td>
<td>Professional paper</td>
</tr>
<tr>
<td>VALIDATION OF OPTIMALLY DESIGNED STATOR-PROPELLER SYSTEM BY EFD AND CFD</td>
<td>str.133-151</td>
<td>Yong Jin Shin, Moon Chan Kim, Jin Gu Kang, Hyeon Ung Kim, I Rok Shin</td>
<td>Original scientific paper</td>
</tr>
<tr>
<td>A DECISION-SUPPORT TOOL FOR DEMOLITION SALE OF A VESSEL</td>
<td>str.153-173</td>
<td>Basak Akdemir, Ahmet Beskese</td>
<td>Review article</td>
</tr>
</tbody>
</table>
PSO-BASED PID CONTROLLER DESIGN FOR SHIP COURSE-KEEPING AUTOPILOT (str.1-15)

Tatijana Dlabač, Martin Ćalasan, Maja Krčum, Nikola Marvučić
Original scientific paper

DETERMINATION OF LINEAR AND NONLINEAR ROLL DAMPING COEFFICIENTS OF A SHIP SECTION USING CFD (str.17-33)

Soon-Seok Song, Sang-Hyun Kim, Kwang-Jun Paik
Original scientific paper

CONTRIBUTION TO THE SEAKEEPING ANALYSIS OF MULTIHULL WARSHIPS (str.35-50)

Rodrigo Perez, Jose M. Riola
Review article

THE TURBULENT BOUNDARY LAYER AND FRICTIONAL DRAG CHARACTERISTICS OF NEW GENERATION MARINE FOULING CONTROL COATINGS (str.51-65)

Burcu Erbas
Original scientific paper
<table>
<thead>
<tr>
<th>Title</th>
<th>Authors</th>
<th>Type</th>
<th>Pages</th>
</tr>
</thead>
<tbody>
<tr>
<td>LIFE CYCLE COST ANALYSIS FOR THE YAW DAMPING SYSTEM OF A WARSHIP FROM A FINANCIAL VIEWPOINT</td>
<td>Uğur Buğra Çelebi, Levent Bilgili, Bilkutay Yılmaz</td>
<td>Professional paper</td>
<td>str.1-9</td>
</tr>
<tr>
<td>FATIGUE ASSESSMENT OF CRACK GROWTH BASED ON FAILURE ASSESSMENT DIAGRAMS FOR A SEMI-SUBMERSIBLE PLATFORM</td>
<td>Shilun Zhao, Chaohe Chen, Tianhui Fan</td>
<td>Original scientific paper</td>
<td>str.11-24</td>
</tr>
<tr>
<td>NUMERICAL STUDY ON THE PREDICTION OF THE BOW FLARE SLAMMING PRESSURE FOR THE CONTAINERSHIP IN REGULAR WAVE</td>
<td>Daewon Seo, Kwang-Leol Jeong</td>
<td>Original scientific paper</td>
<td>str.25-42</td>
</tr>
<tr>
<td>NUMERICAL SIMULATION OF THE CAVITATING FLOW AROUND MARINE CO-ROTATING TANDEM PROPELLERS</td>
<td>Boucetta Djahida, Imine Omar</td>
<td>Original scientific paper</td>
<td>str.43-57</td>
</tr>
<tr>
<td>EXERGY ANALYSIS OF THE MAIN PROPULSION STEAM TURBINE FROM MARINE PROPULSION PLANT</td>
<td>Vedran Mrzljak, Igor Poljak, Jasna Prpić-Oršić</td>
<td>Original scientific paper</td>
<td>str.59-77</td>
</tr>
<tr>
<td>AN ALTERNATIVE SOLUTION FOR INTERACTION OF FLEXURAL GRAVITY WAVES WITH BOTTOM MOUNTED CIRCULAR CYLINDER</td>
<td>Šime Malenica</td>
<td>Original scientific paper</td>
<td>str.79-91</td>
</tr>
<tr>
<td>A FUZZY EVALUATION MODEL OF CHOOSING A MIDDLE MANAGER FOR AN INTERNATIONAL SHIPPING SERVICE PROVIDER</td>
<td>Ji-Feng Ding, Jung-Fong Kuo, Wen-Hui Tai</td>
<td>Professional paper</td>
<td>str.93-107</td>
</tr>
<tr>
<td>NUMERICAL SIMULATION OF ICE MILLING LOADS ON PROPELLER BLADE WITH COHESIVE ELEMENT METHOD</td>
<td>Feng Wang, Zao-Jian Zou, Zhou Li, Yang Wang, Hao Yu, Haihua Zhang</td>
<td>Original scientific paper</td>
<td>str.109-128</td>
</tr>
<tr>
<td>REEFER VESSEL VERSUS CONTAINER SHIP</td>
<td>Predrag Ćudina, Ana Bezić</td>
<td>Professional paper</td>
<td>str.129-141</td>
</tr>
</tbody>
</table>
LIFE CYCLE COST ANALYSIS FOR THE YAW DAMPING SYSTEM OF A WARSHIP FROM A FINANCIAL VIEWPOINT

Summary

Life cycle assessment (LCA) is implemented to various processes with a strict exception for naval platforms and warships. Thus, the recent studies for life cycle analysis of military ships are limited. LCA is a holistic method based on developing an optimization between environmental performance, cost-benefit balance and usability. LCC is a supporting subsection of LCA implemented to the analysis in order to calculate the probable costs during whole phases of a product. In this study, LCA and LCC are implemented to a naval platform and a calculation, which is based on escalation, is realized to compare the maintenance costs and initial cost for all life cycle of the platform. A generic warship is selected and yaw damping system of the ship was chosen for implementation of LCA and LCC. It is accepted that the systems and devices are used 1500 hours during cruising and 7260 hours during hoteling. First, the initial cost of yaw damping system is calculated, then, costs of long term maintenance with and without escalation are considered. In the final part, the profit gained by recycling of yaw damping system is calculated. The results has shown that an extra cost of 38 % and 19 % for operation and maintenance and total cost, respectively, must be added according to escalation calculations. While initial cost and operation and maintenance costs have 51 % and 49 % of total costs without escalation, the share of initial cost decreases to 43 % and the share of operation and maintenance increases to 57 % with the utilization of escalation.

Key words: life cycle assessment; life cycle cost; naval platforms; recycle; escalation

1. Introduction

Navies have great importance for both protecting the coastline and prevent illegal activities across territorial waters. Naval platforms are exceptions for almost all international rules and regulations, which particularly involve the impacts of ships on the human health and environment. Besides, because navies are uncompromisable defence systems, cost analysis is generally ignored to some extent. Thus, implementing approaches such as Life Cycle Assessment (LCA) and Life Cycle Cost (LCC) to naval platforms is not usual. On the other hand, these holistic methods are instrumental and may provide significant benefits. Planning the ship’s all life cycle and using ship life cycle assessment (SLCA), which is a unique
combination of LCA, and LCC not only ensure excellence in war tasks but also provide a remarkable reduction of emissions and wastes. Conducting SLCA may result in a longer service period and minimize the running costs during all phases.

[1] presented a general view for main topics of a life cycle of a product. He touched the types of LCA and its main sections briefly and specified the holistic approach of LCA, generally. In this study, modern ship design approach for green shipping is also explained and energy consumption and environmental impacts of ship LCA is mentioned. [2] indicated that ports and shipping activities are one of the most influential players for developing and being competitive in Europe. The authors also stated that the ever-growing shipping sector causes significant environmental impact. They recognized the importance of a life cycle assessment study in order to identify and measure the environmental impacts of ship-related wastes. It is derived from this study that carcinogens and heavy metals have the greatest impact on environmental issues. It is also indicated that recycling waste oil and fuels may reduce the harmful effects on the environment. [3] compared the two types of materials used in superstructure by the help of LCA methods and investigated the environmental impacts. By using different LCA analyze methods and software; he indicated that the environmental impacts of the composite superstructure are far less than steel superstructure. [4] indicated that LCA is used in order to determine the environmental impacts of different industries. [5] explained that LCA is a method used for determining the consumption and harmful effects of a product. The authors also stated that LCA consists of all phases of a product from the cradle to grave. [6] also presented the effects of LCA on reducing the emissions and discharges during operation and highlighted LCA method in terms of waste management. [7] indicated that growing global race in production industry forces system generators and providers to estimate and optimize the LCC of the whole system process in terms of performance, safety, reliability and maintainability. In this context, the authors stated that system providers should consider the whole process in order to ensure customer requests from the concept design phase to sale phase. [8] expressed that many production companies adopted LCA to evaluate the environmental performances clearly. However, it is also indicated that due to the complexity of the ship’s systems and manufacturing processes it is almost impossible to use LCA properly for ships. The authors investigated the manufacturing methods for different types of ships and developed database software to utilize LCA for ships. [9] investigated the role of the material efficiency to reduce CO\textsubscript{2} emissions during ship manufacture in a life cycle perspective. They resulted that designing and manufacturing the 100 % of the hull with reusable materials reduce CO\textsubscript{2} emissions by 29 %. [10] investigated the implications of a new-build hybrid power system for Ro-Ro ships from a sustainability and life cycle perspective. The authors used a bottom-up integrated system approach to optimize operational profile and considered the manufacturing processes, mass breakdown and end of life management plans. They concluded that because the impacts of hybrid systems are significant, a proper control system must be implemented. [11] studied on ship air emissions from a life cycle perspective. The authors presented a mathematical framework to provide a holistic assessment of airborne emissions for ships. [12] utilized LCA to investigate the steel in ship recycling industry in Bangladesh. They focused on the evaluation of energy use and emissions during different phases of LCA of the steel. The authors resulted that rerolling and cutting activities are the primary sources of emissions and they offered using protective gear to reduce the emissions.

Besides shipping, LCA and LCC have a great area of utilization in various subjects. [13] used LCA to assess the environmental impacts of freshwater thermal pollution. [14] studied the implementation of LCA in Environmental Impact Assessment (EIA) procedures. For their study, the authors investigated two wastewater treatment plant. Four EIA steps, which could theoretically benefit from LCA implementation, are identified: (a) the
environmental comparison of alternatives, (b) the identification of key impacts, (c) the impact assessment, (d) the impact of mitigation measures. LCA is implemented to each step for specific goals. [15] compared different methods to quantify greenhouse gas (GHG) emissions of cropping systems using LCA. The authors aimed to compare several methods for estimating CO$_2$ and N$_2$O emissions and to estimate the relative contribution of soil GHG emissions to the overall Global Warming Potential (GWP). [16] prepared a critical review on Building Information Modeling (BIM) based LCA method application to buildings. The authors reviewed recent studies focused on BIM-based LCA and carried out a methodological analysis for their integration, focusing on the contribution of BIM to simplify data input and optimize output data and results during LCA application to buildings. [17] focused on different modeling options in terms of database choices, system boundary definitions and replacement scenarios of building materials during LCA of the buildings. [18] studied on LCA data quality and the authors concluded that LCA data quality assessment approach is not adequate for enterprise-scientific data because it focuses uniquely on industry average data. They offered some suggestions for allowing consistent data quality assessment that would ensure the usefulness of LCA information. [19] proposed a method aimed to support LCA for evaluation of environmental benefits achievable by light-weight design solutions for the automotive sector. The study mainly focused on developing fuel reduction value coefficient based on weight induced fuel consumption. [20] investigated a shipping container home in terms of LCA and Life Cycle Environmental Impacts (LCEI). The authors used LCA to evaluate six LCEI category indicators: Cumulative Energy Demand (CED), water use, solid waste, global warming potential (GWP), acidification potential, eutrophication potential. [21] implemented LCC to compare the costs of different electrical energy storage systems. [22] studied probabilistic LCC analysis for renewable and non-renewable power plants. The authors developed two probabilistic methods to assess the costs of power plants. [23] utilized LCC to analyze green-building implementation using timber applications. LCC provided the most suitable timbers for different applications in terms of green-building. [24] evaluated the effectiveness of separating layers in railroad track structure using LCC. The authors combined LCC with reliability analysis. [25] integrated LCA and LCC and applied the combined method to the design process of a hybrid train. [26] combines LCA and LCC in Eco-Care-Matrix in order to evaluate the performance of a modernized manufacturing system for glass containers.

[27] presented a novel experimental measurement method to predict wave induced motions and load responses in real sea waves. This method consists of tests, which have been realized with large-scale models under natural environmental conditions. The study focused on the investigation of the ship hull against pitch, roll, bow acceleration and vertical bending moment (VBM) motions. The authors concluded that the difference between the results of experimental and numerical methods of short-term predictions in combined average sailing conditions is less than 8%. In [28] the authors adopted a buoy wave height meter to measure and analyse the coastal wave environment. The seakeeping performance experiments conducted for the same tonnage of round bilge vessel and the deep-V hybrid monohull of large-scale vessel model under the coastal wave conditions. The results show that the difference between the motion characteristics of large-scale vessel models in the coastal wave environment and small-scale vessel models in tank is significant. [29] proposed a computer-based system on automatic elimination of ship design parameters for seakeeping performance. According to the system, first, the weakest parameter is identified and then, these parameters are eliminated for the best seakeeping performance.

Price escalation method is used by [30] for non-linear water tariffs for domestic uses in Spain. The authors analyse the determinants of the price escalation of water supply tariffs in
Spain. They resulted that tariff escalation is influenced by environmental factors and decision makers’ strategic choices. [31] prepared a review on the impact of tariff escalation on the environment. The author mainly focused on the impact of escalating tariffs on the allocation of production and processing between exporters and importers, comparison of the environmental impacts of primary production and processing, the impact of increased income from removing tariff escalation on environmental protection expenditures, the environmental impact of decreased transport to freight.

The methods and models of LCA are presented in the technical report of North Atlantic Treaty Organization (NATO) Research and Technology Organization. Cost prediction methods and models are exemplified in the report and a comprehensive guide is developed for LCC applications of multi-national military projects [32]. NATO also identified the life cycle costs based on generic cost allocation and presented detailed information on these costs. Thus, NATO, by considering the effects of scheduling and efficiency, takes an active role in cost application methods [33]. In another report, NATO explained the subcomponents of warships and costs, in detail. The report presents the costs during the phases of the job definition of ships, pre-feasibility study, project definition, design and development, manufacturing and inventory [34]. US Department of Defence indicated that due to the limited labour force and sources of navies, life cycle cost is an important management tool for efficient source allocation. The main purposes of the document are to reveal the total cost estimation and annual expenses of the ships during design, operation, repair and maintenance processes and to reduce the costs by using LCA methods [35].

In this study, a generic warship operational usage scenario is demonstrated. It is estimated that systems and devices are used 1,500 hours during cruising and 7,260 hours during hotelling. Vital and supporting systems and devices are run 7/24.

2. Materials and Methods

LCA realizes the evaluations with successive and independent processes perspective and it is used for estimating the total environmental impacts caused by all phases of life cycle including the processes not considered in the traditional analysis [36]. LCA can be identified by the help of 6 RE philosophy: Re-think (detailed analyses for the product and its function), re-duce (minimizing the raw material and energy consumption), re-place (using less harmful materials instead of more harmful ones), re-cycle (recyclable materials are chosen), re-use (the product is produced as reusable), re-pair (the product is produced as appropriate for repair) [37]. According to [38], a traditional LCA process can be separated into four phases, which are interrelated with each other. In the first phase, the goal and scope of the study and analysis must be identified and determined. Then, the inventory analysis of the system or the product must be created. The third phase includes the calculations for impact assessment of different stages in the life cycle. Interpretation phase is related to other phases and provides the relationships between them.

Cost escalation, which is used in this study to calculate the real costs of a ship’s system during its life cycle, refers to the increase in the amount of money required to sustain a project over and above the original budgeted amount. Cost escalation is also identified as a result of problems such as delay in land acquisition, unexpected problems in the supply of raw materials, illegal encroachment during project implementation. It is also stated that delays between the planning stage and actual implementation of projects are significant problems resulting in cost escalation [39].

The formula for total system cost is presented in Equation 1. The equation includes various types of expenses which are explained below.

\[ C = C_2 + C_t + C_o \] (1)
where;
- \( C \): Total system cost
- \( C_R \): Research and development cost
- \( C_I \): Investment cost
- \( C_O \): Operation and maintenance cost

3. Results and Discussion

There are four levels for maintenance and repair processes of a ship. Level 1 (Consumer Level) involves planned maintenances which can be operated by crew; Level 2 (On-Site Level) involves planned maintenances which must be realized by the help of repair mechanics; Level 3 (Shipyard Level) consists of informing shipyard personnel in order to perform construction and assembly processes uneventfully and certification process in accordance with international standards for construction and repair; Level 4 (Manufacturer Level) consists of planned maintenances, malfunction localization, repairs and tests/settings which cannot be run by the navy.

In this study, a generic operation concept of a ship is considered, the maintenance periods of different parts of the ship, the levels and processes of these maintenances and the activities and phases of a warship during its life-cycle are investigated.

3.1 Scenario-1 Yaw Damping System

In this sub-section, the required maintenances, maintenance periods and durations of a yaw damping system are explained. Yaw damping is a system consists of active fin equipment which is driven by hydraulic drives. Pitch, heave and roll are three dangerous types of movements, which have adverse effects on load, passengers and ship capabilities, which occurs in rough sea conditions. These movements restrict manoeuvring and shooting ability of warships. Yaw fins move in opposite direction to yaw motion in order to help the ship keep still on water surface.

Two LCA scenarios are considered in this study for consumables and replacement parts of yaw damping system. In consumables scenario, lubricating oil, which is used in hydraulic systems and main bearings, and high-pressure hoses are considered as the most important consumables. While lubricating oil is changed annually, high-pressure hoses are changed quadrennially. Thus, during a life cycle of a ship, which is accepted as 35 years, roughly, lubricating oil and high-pressure hoses are changed 35 and 9 times, respectively. Recommended recycling methods are reprocessing in appropriate recycling facilities.

Scenario for replaced parts includes fin plates, electric motor bearings and valves. While bearings and valves are changed quadrennially, fin plates are changed during half-life modernization. Recycling recommendation for all types of parts is reprocessing in appropriate recycling facilities. Various types of equipment such as oil, sealing components, vibration dampers and hoses replaced with new ones during maintenances. Recycling this equipment may provide side income and minimize the deleterious impacts on the environment.

Hourly wage is assumed as 31 € in man-hour calculations. An escalation calculation, based on the data obtained from the European Central Bank, is realized in order to present the net values of cost for an estimated 35 years life-span of a warship.

Long-term maintenance costs for different levels are estimated as 13,478, 638,591, 46,132, 11,377 and 709,580 € for support, calibration, measure and test tool, Level 1, Level 2 and Level 3, respectively and annually. That means the maintenance of a ship causes a cost 2,128,734 € during approximately 35 years life-span. On the other hand, when escalation is implemented to the processes, the maintenance costs for different levels are calculated as 23,683, 1,020,257, 56,194, 15,567 and 1,116,702 € for support, calibration, measure and test
tool, Level 1, Level 2 and Level 3, respectively and annually. The total maintenance cost is estimated as 3,350,106 € during 35 years life-span.

Figure 1 presents the distribution of systems on which yaw cost analysis is implemented.

![Ship Distribution Chart]

**3.2 Profit Gained by Recycling of Yaw Damping System**

A sample maintenance scenario for yaw damping system is presented and some suggestions for recycling/reusing of consumables and replaced parts are made. The prices of scrap and wastes are predicated on pricing policy of Europe. Table 1 presents the gained profit by implementing recycling processes for yaw damping system.

<table>
<thead>
<tr>
<th>Consumables /Parts</th>
<th>Amount</th>
<th>Unit Price</th>
<th>Period</th>
<th>Total Amount</th>
<th>Gained Profit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil</td>
<td>~420 l</td>
<td>0.86 €/l</td>
<td>Annually</td>
<td>14,700 l</td>
<td>12,642 €</td>
</tr>
<tr>
<td>Steel Sheet</td>
<td>~4 t</td>
<td>210 €/t</td>
<td>~17 years</td>
<td>~8 t</td>
<td>1,680 €</td>
</tr>
<tr>
<td>Hose (Plastic)</td>
<td>~0.4 t</td>
<td>1,645 €/kg</td>
<td>Quadrennially</td>
<td>3.6 t</td>
<td>5,922 €</td>
</tr>
<tr>
<td>Valves (Stainless Steel)</td>
<td>~0.3 t</td>
<td>835 €/t</td>
<td>Quadrennially</td>
<td>2.7 t</td>
<td>2,255 €</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>22,499 €</td>
</tr>
</tbody>
</table>

When escalation is implemented on profit gained by recycling process, the net profits for different consumables and parts are presented as follows: 14,316.89, 2,007.32, 6,973.76 and 2,586.38 € for oil, steel sheet, hose and valves, respectively during 35 years life-span of the ship. Thus, the total net profit is calculated as 25,704.35 € for all consumables and parts.

**4. Conclusion**

Analyzing the costs for the system from a holistic viewpoint shows that the initial investment cost, which consists of research and development cost, is approximately equal to the operation and maintenance costs occurred during all life cycle. Considering the real value of costs in accordance with the data of the European Central Bank, it is also estimated that operation and maintenance costs may be much more than the initial investment cost. While initial investment cost has 51% of total costs, operation and maintenance costs have 49%, without using escalation, it can be seen that these ratios change when escalation method is implemented. In this case, while the initial investment cost decreases to 43%, operation and maintenance cost increases to 57%. Thus, it can be deduced that an extra cost, which is as much as the initial investment cost at least, may occur.
The total cost of all systems is calculated as 9,411,780.04 €, whose 2.41 %, 48.6 % and 48.98 % are expenditures for research and development, investment and operation and maintenance, respectively. When escalation calculation is implemented on the costs, the total cost of all systems is estimated as 11,161,197.32 €, in which research and development, investment and operation and maintenance processes are responsible for 2.03 %, 40.98 % and 56.97 %, respectively. It is also observed that escalation calculation only has effects on operation and maintenance process. It can be seen that there is approximately 1,750,000 € difference between two cases, with and without escalation calculation. This amount brings an extra cost by 38 % and 19 % for operation and maintenance and total cost, respectively. Thus, it is accepted that considering possible extra costs in cost analysis calculations is much more beneficial and realistic.

The results of cost analysis show that operation and maintenance cost is almost equal to initial investment cost. Analyzing the costs with escalation formulas show that the realistic values may be higher than initial investment cost. An extra cost by 38 % and 19 % for operation and maintenance and total cost, respectively, must be added according to escalation calculations. While initial cost and operation and maintenance costs have 51 % and 49 % of total costs without escalation. The share of initial cost decreases to 43 % and the share of operation and maintenance increases to 57 % when escalation is applied. These changes in amounts show that a realistic life cycle cost analysis from the cradle to the grave perspective can provide a better budget prediction. Moreover, recycling and reusing are extremely remarkable solutions for reducing waste energy and sources.

REFERENCES


Submitted: 19.09.2017. Uğur Buğra Çelebi, ugu.celebi@gmail.com, Yildiz Technical University, Yildiz Campus, Barbaros Bulvari, 34349, Beşiktaş/Istanbul
Accepted: 21.11.2018. Levent Bilgili, leventbilgili1661@gmail.com, Bandirma Onyedi Eylul University, Yeni Mahalle Şehit Astsubay Mustafa Soner Varlık Caddesi, No: 77, 10200, Bandırma/Balıkesir
Bilkutay Yılmaz, Undersecretariat for Defense Industries, Devlet Mahallesi Süleyman Emin Caddesi No:6-7, 06420, Çankaya/Ankara
FATIGUE ASSESSMENT OF CRACK GROWTH BASED ON FAILURE ASSESSMENT DIAGRAMS FOR A SEMI-SUBMERSIBLE PLATFORM

UDC 629.563.21:629.5.015.4
Original scientific paper

Summary

This paper deals with the assessment of fatigue crack propagation on the connection between column and brace for a semi-submersible. The analysis of global and local structural responses under different sea states are performed to acquire the transfer functions of stresses. Based on an existing crack the Failure Assessment Diagrams (FAD) are applied as criterion of acceptance for the safety of crack and structure during the crack growth calculation cycle. The crack growth rate considering threshold stress intensity factor and stress ratio is used. During safety assessment the stress response from ultimate sea state is outlined. A comparison of fatigue crack growth using ultimate stress and normal stress data with different crack growth rate is presented. The results show the reliability of fatigue assessment using FAD as a measurement of acceptability of crack propagation.

Key words: Crack propagation; Semi-submersible; FAD; Fracture mechanics; Structural safety; Offshore platform

1. Introduction

Currently the S-N curve is still effective in engineering field for the fatigue life estimation subjected to the initial state of crack failure. For ship structure that is exposed to fatigue damage reliability-based method [1] is also used in the inspection planning or other aspects corresponding to marine structures. Even aging effects on marine structure integrity [2] are also considered in recent study. In general, the fatigue life is deemed on the occurrence of crack or other types of severe structural defect. Fracture mechanics is introduced in the fatigue assessment and becoming more popular in the aspect of crack propagation which is specifically acknowledged as the second stage of the entire fatigue process. Due to the complexity of the interaction between aggressive marine environment and fatigue wave load, methods for fatigue crack assessment varies from many situations. Nevertheless, all the practices are based on Paris Law [3] and Palmgren-Miner’s rule of cumulative damage ratio. Throughout recent years several types of crack growth rate have been developed for the crack growth calculation. McEvily [4] and his co-workers developed the modified constitutive
relation which considered the initial defect and effects of load sequence. Huang and Moan [5][6][7] presented the unique crack growth rate model which determined the material constants for fatigue life prediction of structures subjected to various amplitude loading history. Cui [8] applied this unique crack growth model into the crack growth calculation for several key parts of a deep water semi-submersible.

As a long-term interaction between offshore structure and marine environment fatigue assessment depends on various factors such as wave loading history, corrosion, wind and other environmental attacks. A basic safety requirement during the crack growth process should not be ignored and thus the Failure Assessment Diagrams (FAD) [9] is introduced in each load cycle during the cracking process. In the field of subsea pipeline, the FAD have been a useful tool for assessing the acceptability of flawed pipelines and also in the fatigue assessment for tubular joints [11] even for non-metallic materials safety assessment [12].

However only constant amplitude loading is used in the normal FAD process and irregular random wave loads are dominant in fatigue assessment. This paper provides a method for using irregular wave load with FAD and performs fatigue assessment of an existing crack. After the global and local structural response analysis, the crack growth calculation will consider the ultimate stress for safety assessment in FAD process. To use random loads, the fatigue wave load history is applied by combining the stress transfer function and wave scatter diagram. It is necessary to split the fatigue wave load into separate cycle for creating single load case for every step of crack growth. Then a comparison is presented for discussion of using ultimate stress response and stress generated by wave loading history in the crack growth calculation. The results show the feasibility and reliability of the method in fatigue crack growth calculation and assessment.

2. Analysis Approach

2.1 Separate Fatigue load cycles

For better understanding a comparison of S-N curve and FAD is listed below in Table 1.

<table>
<thead>
<tr>
<th>Description</th>
<th>S-N curve</th>
<th>FAD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Considering crack growth</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Stage of fatigue development</td>
<td>Until occurrence of crack</td>
<td>Until failure of crack</td>
</tr>
<tr>
<td>Criterion of structure failure</td>
<td>Crack(s) found</td>
<td>FAD outreached</td>
</tr>
<tr>
<td>Fatigue load</td>
<td>Long term load distribution</td>
<td>Regular constant amplitude load</td>
</tr>
<tr>
<td>Considering sequence of loads</td>
<td>No</td>
<td>No</td>
</tr>
<tr>
<td>Considering stress components</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Using crack growth rate</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Crack safety evaluation</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Considering random wave load</td>
<td>Yes</td>
<td>No</td>
</tr>
</tbody>
</table>

From Table 1 the advantages of introducing FAD into offshore structure fatigue assessment are clearly demonstrated. The only problem is implementing irregular wave load into Separate assessment cycle. According to BS 7910 for the situation of using variable amplitude loading, the stress spectrum should be represented as a distribution of stress ranges versus numbers of occurrences [9]. Based on the P-M rules of linear damage accumulation, the wave scatter diagram can be introduced for dividing the consecutive long-term load distribution into Separate cycles. In this paper wave scatter diagrams in respect to 8
Fatigue Assessment of Crack Growth Based on Shilun Zhao, Chaohe Chen
Failure Assessment Diagrams for Semi-Submersible Tianhui Fan, Yong Jiang, Yijun Shen

directional sea states are provided by CNOOC Energy Technology & Services Limited. A diagram of North direction to the semisubmersible is listed below in Table 2.

<table>
<thead>
<tr>
<th>$H_s$ (m)</th>
<th>&lt;2</th>
<th>2 ~ 3</th>
<th>3 ~ 4</th>
<th>4 ~ 6</th>
<th>6 ~ 8</th>
<th>8 ~ 10</th>
<th>&gt;10</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-</td>
<td>-</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>-</td>
<td>-</td>
<td>0</td>
</tr>
<tr>
<td>0 ~ 0.5</td>
<td>-</td>
<td>-</td>
<td>0.05</td>
<td>0.17</td>
<td>0.01</td>
<td>-</td>
<td>-</td>
<td>0.22</td>
</tr>
<tr>
<td>0.5 ~ 1</td>
<td>-</td>
<td>-</td>
<td>0</td>
<td>0.57</td>
<td>0.05</td>
<td>0</td>
<td>-</td>
<td>0.61</td>
</tr>
<tr>
<td>1 ~ 1.5</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.19</td>
<td>0.04</td>
<td>0.01</td>
<td>-</td>
<td>0.23</td>
</tr>
<tr>
<td>1.5 ~ 2</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.03</td>
<td>0.11</td>
<td>0.02</td>
<td>-</td>
<td>0.16</td>
</tr>
<tr>
<td>2 ~ 2.5</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.01</td>
<td>0.1</td>
<td>0.01</td>
<td>-</td>
<td>0.12</td>
</tr>
<tr>
<td>2.5 ~ 3</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.23</td>
<td>0.05</td>
<td>0</td>
<td>0.28</td>
</tr>
<tr>
<td>3 ~ 3.5</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.01</td>
<td>0.11</td>
<td>0.01</td>
<td>0.14</td>
</tr>
<tr>
<td>3.5 ~ 4</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.01</td>
<td>0.03</td>
<td>0</td>
<td>0.04</td>
</tr>
<tr>
<td>4 ~ 4.5</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0</td>
<td>0</td>
<td>-</td>
<td>0</td>
</tr>
<tr>
<td>4.5 ~ 5</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0</td>
</tr>
<tr>
<td>&gt;5</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Total</td>
<td>0</td>
<td>0</td>
<td>0.05</td>
<td>0.96</td>
<td>0.56</td>
<td>0.23</td>
<td>0.01</td>
<td>1.8</td>
</tr>
</tbody>
</table>

In Table 2 $H_s$ and $T_{xz}$ are presenting significant wave height and average cross zero period respectively. For a specific significant wave height $H_i$ the fatigue stress range $\Delta \sigma_i$ can be described as

$$\Delta \sigma_i = H_i \cdot \Delta \sigma_{unit}$$

where $\Delta \sigma_{unit}$ is stress range under unit wave height by picking maximum stress $\sigma_{mx}$ and minimum stress $\sigma_{mn}$ from stress transfer functions at different phase positions under the same period.

Supposed the annual cycle number of $\Delta \sigma_i$ under $H_i$ is $n_{xz}$, then

$$n_{xz} = 365 \times 24 \times 3600 \times \frac{p_i}{T_{xz}}$$

where $T_{xz}$ is the cross zero period of $H_i$ and $p_i$ is the probabilistic occurrence rate of $H_i$ and $T_{xz}$.

Based on above separation process of irregular wave loads the fatigue assessment of crack growth can be represented as a series of $\Delta \sigma_i$ versus $n_{xz}$. This is a method for assessing the acceptability of crack growth in each fatigue cycle using FAD under irregular amplitude wave load. The complete fatigue assessment procedure is illustrated in Figure 1.
Crack growth cycle starts from the 1st wave direction which is followed by picking up the periods. Under the specific period all the wave heights in the wave scatter data will be applied with the stress range in the calculation of SIF. The crack growth cycle number under a specific $\Delta \sigma_i$ aligns with $n_{yi}$. Namely the propagation process proceeds with different $n_{yi}$ under corresponding $\Delta \sigma_i$. A new crack growth generation under a new $\Delta \sigma_{i+1}$ starts with the end of previous number of $n_{yi}$. A year is counted when all the wave directions and periods are used in the crack propagation process.

The overall procedure relies on FEM analysis of the semisubmersible. The structural stress responses are obtained from the FEM analysis under various sea states. The stress transfer functions are extracted from previous step under unit wave amplitude. For the weld geometry of the cracked component, the hot spot stress $\sigma_{hx}$ is calculated via extrapolation from stresses $\sigma_{0.5t}$ and $\sigma_{1.5t}$ at readout points 0.5t and 1.5t [12],

$$\sigma_{hx} = 0.5 \times (3 \cdot \sigma_{0.5t} - \sigma_{1.5t})$$ (3)

When the stress ranges are all prepared then stress intensity factor (SIF) $K_i$ can be calculated for the fatigue assessment with other parameters determined by the weld profile and the crack, such as initial crack size in depth $a_0$, crack size in width $c_0$, angle $\theta$ from $c_0$ to $a_0$, etc. The fatigue assessment processes with the crack growth rate under a single load cycle to obtain an increment $\Delta \alpha$ on the crack growth size up to the critical crack size $a_c$. Before generating a new crack size $a_i = a_{i-1} + \Delta \sigma$ the FAD is used to analyze the acceptability of the $a_i$. For conservatism the stress responses under potential critical wave load used in the FAD assessment to evaluate the crack growth status while the wave scatter data is used in calculating the crack propagation increments. This procedure continues until the assessment point of $a_i$ outreaches the assessment line which presents the acceptable limit. At this stage the crack is deemed as failure and the fatigue life of the welded structure is set due.
sequence of 8 diagrams applied in crack growth cycles is in clockwise from North to North West. A reversed sequence of diagrams is also performed in the crack growth process using Paris law for comparison. Since it is based on the Palmgren-Miner’s rule of a linear crack growth accumulation, the effect of loading sequence of stress range within each diagram is ignored.

2.2 Failure Assessment Diagrams (FAD)

These diagrams are one of the main engineering tools for assessing the fracture-plastic collapse in the cracked components. Simultaneously the fracture and plastic collapse are assessed by normalized parameters fracture ratio $K_r$ and load ratio $L_r$ defined as following,

$$K_r = \frac{K_I}{K_{max}}$$

(4)

$$L_r = \frac{\sigma_{ref}}{\sigma_Y}$$

(5)

The fracture ratio of $K_r$ to $K_{max}$, corresponding to stress intensity factor and material fracture toughness respectively, evaluates the current cracked component against fracture. The load ratio of $\sigma_{ref}$ to $\sigma_Y$, representing the reference stress and yield strength, is to evaluate the component against plastic collapse. The reference stress $\sigma_{ref}$ varies from different types of crack. Nevertheless, it is defined by primary membrane and bending stress components $P_m$, $P_b$, also by crack depth $a$, crack width $c$, plate thickness $B$ and width in plane of flaw $W$. For the case of surface flaw under normal bending strain considered in this paper, $\sigma_{ref}$ is defined as following,

$$\sigma_{ref} = \frac{P_b + [P_b^2 + 9P_m^2(1-\alpha^n)^2]^{0.5}}{3(1-\alpha^n)^2}$$

(6)

where

$$\alpha^n = \frac{2a/B}{1+(B/c)} \text{ for } W \geq 2(c+B)$$

(7)

$$\alpha^n = (a/B)(2c/W) \text{ for } W < 2(c+B)$$

(8)

Corresponding to various types of crack which is denoted as flaw in the BS 7910:2013 [9], the specifications of $\sigma_{ref}$ can be referred in Annex M and P of the document.

Generally, the failure assessment line is defined as following,

$$K_r = f(L_r)$$

(9)

It will be modified due to different situations such as requirement of stress-strain data or ductile tearing. According to BS 7910 Option 1 approach is selected because of lacking uniaxial tensile stress-strain data and not considering ductile crack growth. For these two situations Option 2 and Option 3 approaches are introduced in BS 7910. The assessment line of Option 1 is defined as below,

$$\begin{align*}
    f(L_r) = (1 + 0.5L_r^2)^{-0.5} [0.3 + 0.7 \exp(-\mu L_r^d)] & \quad \text{for } L_r \leq 1 \\
    f(L_r) = f(1) \cdot L_r^{(1-1)/(2N)} & \quad \text{for } 1 < L_r \leq L_{r,max} \\
    f(L_r) = 0 & \quad \text{for } L_r > L_{r,max}
\end{align*}$$

(10)
where $\mu, N$ are defined as

$$\mu = \min\left(0.001 \frac{E}{\sigma_Y}, 0.6\right)$$

(11)

$$N = 0.3 \left(1 - \frac{\sigma_Y}{\sigma_U}\right)$$

(12)

In the equation $\sigma_Y$ represents tensile strength of material and $E$ is elastic modulus.

**Figure 2 Option 1 Assessment Line**

Figure 2 illustrates the Option 1 assessment line and path of assessment points on the same coordinate and their relationships on defining the critical point of failure. The crack growth path is not the real time growing direction of the crack on welded joint but a set of the assessment points. $L_{r,max}$ is the permitted value of $L_r$ and is set at a point as

$$L_{r,max} = \frac{\sigma_Y + \sigma_U}{2\sigma_Y}$$

(13)

The purpose of $L_{r,max}$ is to prevent plastic collapse and assessment line is cut off where the $L_{r,max}$ is reached.

### 2.3 Fatigue Crack Growth Law

Paris law is the typical crack growth rate expression from experimental practice. Considering a threshold value of $\Delta K_0$, the stress intensity factor corresponding to the hot spot stress range of cracked component, $da/dN$ is assumed to be zero if the value of $\Delta K$ is below the value of $\Delta K_0$. Based on Paris law a modified $da/dN$ is used in the calculation of crack propagation.

$$\frac{da}{dN} = C[(\Delta K)^m - (\Delta K_{th0})^m]$$

(14)

$$\Delta K_P = M_R \Delta K$$

(15)

$$M_R = \begin{cases} (1 - R)^{-\beta} & (R < 0) \\ (1 - R)^{-\beta} & (0 \leq R < 0.5) \\ (1.05 - 1.4R + 0.6R^2)^{-\beta} & (0.5 \leq R < 1) \end{cases}$$

(16)
This is the unique crack growth rate developed by Huang [5] for the fatigue life prediction of marine steel structures. Since crack growth rates of welded joint under different applied loading ratios behaves independently, the purpose of bringing up this modified crack growth rate is to establish a concise model for crack growth data under different stress ratios \( R \) to the curve corresponding to \( R = 0 \). \( \Delta K_e \) and \( \Delta K_{th,0} \) are stress intensity factor and threshold value on the level of \( R = 0 \). \( M_{R} \) is the modifying factor for stress ratio \( R \). According to Huang [5,6,7], most of the crack growth rate data gathers around the curve on the level of \( R = 0 \) after transferring the \( da/dN - \Delta K \) data into \( da/dN - \Delta K_e \) data.

In equation \( \beta \) and \( \beta_1 \) are parameters depending on material property and environment. Material constants \( C, m \) are these corresponding to \( R = 0 \). For offshore structures Huang [5] recommends

\[
\begin{align*}
\beta &= 0.22 + \frac{0.65}{1+0.0835 \Delta K^2} \\
\beta_1 &= 0.84
\end{align*}
\]  

(17)  

(18)

Huang [5] also recommends the mean value crack growth rate under insufficient data for material if the unique crack growth rate is applied

\[
\frac{da}{dN} = 8.32 \times 10^{-9}(\Delta K_e^{2.08} - 7.2^{2.08})
\]  

(19)

Besides Huang’s recommendation the crack growth rate based on Paris law

\[
\frac{da}{dN} = C(\Delta K)^m
\]  

(20)

is assumed to be a reference compared to the mean value crack growth rate. Constants \( C, m \) that depend on material and the applied conditions differ in two situations. Both values of \( C \) and \( m \) for Paris law are defined in BS 7910:2013 [9].

For the simplified fatigue crack growth rate \( C = 2.3 \times 10^{-12} \) and \( m = 3 \) under the marine environment at temperatures up to 20°C, with or without cathodic protection, both values above are recommended for steels (excluding austenitic stainless steels). Due to situation of overtaken design fatigue life and the cathodic protection has lost effect, the fatigue crack growth threshold for welded joint is set to \( \Delta K_0 = 0 \, N/mm^{3/2} \).

2.4 Stress Intensity Factor for Fatigue Assessment

The stress intensity factor \( \Delta K \) is determined by following equation

\[
\Delta K = Y(\Delta \sigma)\sqrt{a}
\]  

(18)

where \( Y \) is the stress intensity correction factor. It contains the primary stress and secondary stress components specifically defined in BS 7910:2013 and for the fatigue assessment only primary stress components offer contribution on the crack growth. Nevertheless, for the fracture assessment both stress components are necessarily considered [9].
3. Analysis Approach

3.1 Finite Element Modelling (FEM)

The finite element model of the semi-submersible is established via ANSYS suite, as seen in Figure 3. The principle dimensions of semisubmersible are given in Table 3. Basically, two level of finite element (FE) models are used in the fatigue assessment [13]. Level one consists of the semi-submersible global FE model and is used to obtain global stress and strain on the connection of aft column and brace, where the crack is discovered. In level two the connection is modeled with internal vertical and horizontal stiffeners at each plate and frame. As seen in Figure 4 a refined mesh with boundary condition defined is created on the interconnected plates between brace and outer shell of column.

![Fig.3 Finite Element Model](image1)

![Fig.4 Local Structure Model](image2)

The elements used in the FEM analysis are BEAM188, SHELL181 and MASS21 regarding to line body, surface body and mass unit respectively. The local connection is modeled by slicing the edges from global structure and the boundary conditions are obtained by interpolating the nodes on the cutting edges. According to previous study relating to crack propagation solid elements are applied in the finite element analysis on crack tip positioning and crack grow path predicting [14][15][16]. However, the stress response is the only focused parameter in the crack propagation process with built in FAD assessment. Hence the crack tip opening position and the crack grow path are not considered in the element simulation. Considering the integrity and accuracy of load transfer, SHELL 181 is used in the global structure as well as local connection and the mesh is refined for adjacent are around the initial crack.

For the global FE model mesh size is defined in accordance with the spacing of frames. The mesh size of element for connection FE model is set to 1~1.5t, where t is the plate thickness. Boundary conditions are defined at the main deck plane by constraining 3 different nodes according to DNV GL rules [17].

<table>
<thead>
<tr>
<th>Length</th>
<th>Width</th>
<th>Depth</th>
<th>Displacement</th>
<th>Water Depth</th>
</tr>
</thead>
<tbody>
<tr>
<td>104.5 m</td>
<td>71.5 m</td>
<td>42.0 m</td>
<td>44060 ton</td>
<td>1500 m</td>
</tr>
<tr>
<td>Wetted Area</td>
<td>Draft</td>
<td></td>
<td>Moment of Inertia (x10^{10} kg*m^2)</td>
<td></td>
</tr>
<tr>
<td>11933 m²</td>
<td>21 m</td>
<td></td>
<td>Ixx</td>
<td>Iyy</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>4.450</td>
<td>5.465</td>
</tr>
</tbody>
</table>
Fatigue Assessment of Crack Growth Based on Shilun Zhao, Chaohe Chen
Failure Assessment Diagrams for Semi-Submersible Tianhui Fan, Yong Jiang, Yijun Shen

8 different sea states headings aligned with 8 directional wave scatter diagrams are considered during the hydrodynamic pressure analysis from 0° to 360° with interval of 45° and 18 periods from 3s to 30s with interval of 1.5s. According to DNV GL codes [18] each load case corresponding to one period and one wave heading should be calculated at two-time instances, namely corresponding to a wave crest amidships and a wave zero crossing at the same point. Thus, two phase positions of 0° and 270° are applied in each period and a total number of load case is $8 \times 18 \times 2 = 288$ while the number of stress range is $8 \times 18 = 144$. These pressures applied on the subsea part of structure under different sea state are transferred by AQWA WAVE as Figure 5 shows. During the FE analysis of global structure several characteristic hydrodynamic responses are considered for potential strength failure, such as split force between pontoons $F_x$, torsion moment about a transverse horizontal axis $M_T$, longitudinal shear force between the pontoons $F_L$, longitudinal acceleration of deck mass $a_L$, transverse acceleration of deck mass $a_T$, vertical acceleration of deck mass $a_V$. Definitions for these kinds of responses are given in DNV GL codes [18] and these types of responses correlate to various potential critical load cases of the structure. A design wave approach [18] is applied for the semi-submersible platform to define design wave loads and to evaluate the stress of characteristic responses. In table 4 the periods and wave heights of corresponding responses are listed and the first principle stress of local joint (Node No. 39700) where the crack is located are also listed in the last column. Crest position describes positions of the wave zero-crossing point and the wave crest regarding to midship of semi-submersible platform. Boundary conditions of local connection are obtained via global FE model strain responses. Figure 6 shows Von Mises equivalent stress distribution of local connection under ultimate sea state of 45° and ultimate stress $\sigma_{ult} = 47MPa$ is obtained. It is derived from load case of $F_L$ with period = 8.98 s and wave height = 17.9 m and the wave zero-crossing amidships.

Table 4 Stress under Characteristic Response

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Characteristic Response</th>
<th>Period (s)</th>
<th>Direction (degree)</th>
<th>Wave Height (m)</th>
<th>Crest Position (degree)</th>
<th>1st Principle Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>01</td>
<td>$F_x$</td>
<td>8.98</td>
<td>90</td>
<td>14.528</td>
<td>0</td>
<td>21.62</td>
</tr>
<tr>
<td>02</td>
<td>$F_x$</td>
<td>270</td>
<td>18.84</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>03</td>
<td>$M_T$</td>
<td>7.85</td>
<td>120</td>
<td>10.486</td>
<td>0</td>
<td>16.46</td>
</tr>
<tr>
<td>04</td>
<td>$M_T$</td>
<td>270</td>
<td>22.04</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>05</td>
<td>$F_L$</td>
<td>8.98</td>
<td>45</td>
<td>17.916</td>
<td>0</td>
<td>46.85</td>
</tr>
<tr>
<td>06</td>
<td>$F_L$</td>
<td>270</td>
<td>20.22</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>07</td>
<td>$a_L$</td>
<td>6.98</td>
<td>0</td>
<td>12.014</td>
<td>0</td>
<td>20.31</td>
</tr>
<tr>
<td>08</td>
<td>$a_L$</td>
<td>270</td>
<td>14.65</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>09</td>
<td>$a_T$</td>
<td>6.28</td>
<td>90</td>
<td>11.352</td>
<td>0</td>
<td>19.68</td>
</tr>
<tr>
<td>10</td>
<td>$a_T$</td>
<td>270</td>
<td>29.07</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>$a_V$</td>
<td>10.47</td>
<td>0</td>
<td>22.476</td>
<td>0</td>
<td>31.24</td>
</tr>
<tr>
<td>12</td>
<td>$a_V$</td>
<td>270</td>
<td>24.76</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>$F_L+F_x$</td>
<td>8.98</td>
<td>60</td>
<td>17.342</td>
<td>0</td>
<td>32.03</td>
</tr>
<tr>
<td>14</td>
<td>$M_T+F_x$</td>
<td>270</td>
<td>20.14</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>$M_T+F_x$</td>
<td>11.683</td>
<td>16.52</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>$M_T+F_x$</td>
<td>270</td>
<td>22.03</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
3.2 Define Crack Properties

As crack growth process is very sensitive to input parameters, the initial crack dimensions are carefully selected in previous study. For example, N S. Ermolaeva [20] decides the depth and width of crack according to BS 7910:2005 for analyzing partial penetration weld in a steeling casting. Other studies [8][13][20][21] also choose the initial dimensions of crack based on current codes in a conservative way when they are lacking available data of crack properties. It is also recommended to use actual crack data detected by available devices [9] as Wolfgang Fricke [22] and G. Terán [23] perform physical fatigue test on welded steel structures.

The type of material used is DH36 and the yield strength $\sigma_y$ and tensile strength $\sigma_t$ are 355 MPa and 490 MPa respectively. Crack located in the plate between left aft column and brace is detected upon subsea inspection by ACFM device and is parallel to the direction of weld, considered as toe crack for conservatism. According to the definition of flaw types from BS 7910:2013, this crack is defined as surface planar flaw due to its orientation and location in the connection. The initial crack depth $a_0 = 1.5 \text{mm}$, initial crack width $c_0 = 2.3 \text{mm}$. The adjacent plate thickness of aft column is $t = 40 \text{mm}$.

3.3 Fatigue Assessment Results and Discussions

Crack growth calculation proceeds after the FEM analysis by splitting fatigue load distribution into Separate cycles as described in chapter 2.1. Using 2 types of crack growth rate and using stress range under ultimate sea state as unique constant fatigue load for the propagation calculation, also using the Paris Law under the reversed load sequence of wave scatter diagrams. the results are listed below in Table 3. The paths of assessment points under 2 types of growth rate are recorded in Figure 7. The crack growth in depth $a$ with the increment of load cycles is presented in Figure 8.

From Table 5 the results from four different calculations are compared. The critical crack size including depth and width are similar. The deviations between all the critical crack sizes are less than 9%. The biggest deviation is between $a_{f1}$ and $a_{f2}$, which is 8.24%. The largest depth value obtained under Paris law while the largest width value acquired under ultimate sea state with Paris law. The shortest remaining life happened under ultimate sea state with Paris law which only gives a result of 1 year and the longest can be seen under Paris law giving 14 years.
Fatigue Assessment of Crack Growth Based on Failure Assessment Diagrams for Semi-Submersible

Shilun Zhao, Chaohe Chen
Tianhui Fan, Yong Jiang, Yijun Shen

Table 5 Fatigue Assessment Results

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Fracture Ratio (K_{in})</td>
<td>0.3427</td>
<td>0.3427</td>
<td>0.3427</td>
<td>0.3427</td>
</tr>
<tr>
<td>Initial Load Ratio (L_{r0})</td>
<td>0.2797</td>
<td>0.2797</td>
<td>0.2797</td>
<td>0.2797</td>
</tr>
<tr>
<td>Critical Fracture Ratio (K_{rf})</td>
<td>0.9677</td>
<td>0.9604</td>
<td>0.9738</td>
<td>0.9738</td>
</tr>
<tr>
<td>Critical Load Ratio (L_{rf})</td>
<td>0.4079</td>
<td>0.4199</td>
<td>0.4249</td>
<td>0.4249</td>
</tr>
<tr>
<td>Critical Crack Depth (\alpha_f)</td>
<td>13.94 mm</td>
<td>15.09 mm</td>
<td>15.05 mm</td>
<td>14.83 mm</td>
</tr>
<tr>
<td>Critical Crack Width (c_f)</td>
<td>24.01 mm</td>
<td>23.35 mm</td>
<td>23.29 mm</td>
<td>23.06 mm</td>
</tr>
<tr>
<td>Number of Cycles</td>
<td>10346</td>
<td>74625</td>
<td>74636</td>
<td>57503</td>
</tr>
<tr>
<td>Remaining Life of Crack</td>
<td>1 year</td>
<td>14 years</td>
<td>14 years</td>
<td>10 years</td>
</tr>
</tbody>
</table>

Fig. 7 Paths of Assessment Points

Fig. 8 Crack Size versus Numbers of Load Cycle
This fact can be explained by the stress range used within the crack growth cycles. Ultimate stress range contributes more in each fatigue cycle with bigger values of $\Delta K$ than the stress ranges from wave scatter diagram do. This is also reflected from Figure 7 that the numbers of assessment points are less from Figure 7 (a) than from Figure 7 (b)&(c). The numbers of assessment points represent the effective load cycles from different load case. Thus, with bigger value of $\Delta K$ within a crack growth cycle the crack growth size on depth $a_i$ or width $c_i$ is bigger, which means fatigue crack propagates faster. The structure will reach failure with less fatigue load cycles. This can contribute to shorter remaining life.

According to Table 5 and Figure 7, using unique crack growth rate to calculate and assess fatigue crack gives results on an intermediate level other than those using Paris law under ultimate sea state and pure Paris law. It shows in Figure 8 that the crack growth process generates less 17000 load cycles approximately under unique crack growth rate than under Paris law. In terms of critical crack sizes, they are relatively close when it comes to two crack growth rates with 1.7% and 1.3% deviation in depth and width respectively. This is aligned with the theory study of Huang [5,6,7] and practice of Cui [8]. Huang pointed that mean value unique crack growth rate was obtained by transferring crack growth rate under any stress ratio to that under zero stress ratio. Comparing to the database of curves of $da/dN$ versus $\Delta K$ the curve under zero stress ratio is in the mediate level among the curves under other values of stress ratio (from $R = 0.05\sim0.8$). Cui performed fatigue crack growth with consecutive stress spectrum using Huang’s unique crack growth rate and obtained shorter remaining fatigue life than using traditional Paris law. This fact is also approved in this paper by using splitting consecutive stress ranges into separate fatigue load cycle and adding in the FAD as a standard for evaluating the safety and acceptability of each fatigue crack growth in the load cycles.

By comparing the results from crack propagation under normal sequence and reversed sequence, as shown in Table 5, Figure 7(b) and Figure 8, it can be concluded that random loading sequence from wave scatter diagrams merely has an effect in the separate crack growth cycles. As it is mentioned above this separate propagation process is based on linear damage accumulation, for a specific diagram the range of $H_a$ and $T_z$ with the combination of load cases are set. Thus, the results from 2 type of sequences are almost equal and the effect of loading sequence of diagrams can be ignored.

Furthermore, the largest critical crack size (in depth or in width), $a_i = 15.09\,mm$ and $c_i = 24.01\,mm$ is still far from penetrating the plate thickness, $t = 40\,mm$, of the local connection between column and brace. Above all, it proves that the method applied in this paper can provide a reasonable and reliable result on calculating and assessing structural fatigue crack.

According to the Shao [10], who performed physical tests on the crack in a T shape tubular joint and applied FAD on assessing the safety of welded tubular joint containing fatigue cracks, all types of failure assessment diagrams have sufficient safety redundancy and using FAD in assessing welded tubular joints is safe and relatively conservative. Thus, it can be concluded that using Option 1 assessment line is safe on assessing fatigue crack growth. Besides, based on above works the results obtained in this paper could be reliable preference for engineering practice.
4. Conclusions

In this paper a new method for calculating and assessing fatigue crack growth in column and brace connection of semisubmersible is introduced. By splitting consecutive stress spectrum or stress ranges into the distribution of stress ranges versus their occurrences using scatter diagram, the Option 1 assessment line is added into the separate crack growth cycles as a tool for assessing the propagation safety of fatigue crack. Using unique crack growth rate on predicting critical crack size and remaining life is more reliable than using traditional Paris law. Option 1 assessment line is optional on choosing FAD according to previous study since all kinds of current FAD are relatively conservative.

This method is based on P-M rules of linear accumulative damage, which is widely acknowledged suitable in engineering practices. By comparing crack growth process under normal and reversed sequences of loading using wave scatter diagrams, it is approved that the effect of loading sequence of diagrams can be ignored. Through the assessment of FAD it can be assured that each crack growth size \(a_i\) or \(c_i\) is considered as acceptable in fatigue load cycles. Due to lack of physical tests data the accuracy of the calculation is still under investigation. However, by comparing to existent study the practice in this paper still provides reasonable results on the critical values of fatigue crack and is proved to be aligned with previous work. The method presented in this paper extends the use of FAD to the randomly loaded offshore structure by applying wave scatter diagrams and separating crack growth cycles. According to Table 1 it considers the stress components which include the primary and secondary stress components in the stress intensity factor and via applying wave scatter diagrams and by splitting crack growth process into separate cycles, the accuracy of crack propagation calculation is improved and more reliable with the safety evaluation of FAD. Therefore, it is concluded the method presented in this paper is practical in fatigue assessment of semisubmersible and reliable on providing critical values of fatigue crack.

Acknowledgment

This paper was financially supported by National Natural Science Foundation of China (Grant No. 51709118); China Postdoctoral Science Foundation (Grant No. 2017M612669); Science and Technology Program of Guangzhou (Grant No. 201804010482); the Fundamental Research Funds for the Central Universities (Grant No.2017BQ089); the Funds for Marine Economic Development of Guangdong, China (Grant no. GDME-2018B003); by State Key Laboratory of Ocean Engineering (Shanghai Jiao Tong University)(Grant No. 1708); and by State Key Laboratory of Coastal and Offshore Engineering (Dalian University of Technology)( Grant No. LP1805).

REFERENCES


NUMERICAL STUDY ON THE PREDICTION OF THE BOW FLARE SLAMMING PRESSURE FOR THE CONTAINER SHIP IN REGULAR WAVE

UDC 629.544:629.5.017.2:629.5.024.31
Original scientific paper

Summary

A container ship has large flare on the bow flare region to accommodate more container recently. This flare region experiences the impact pressure due to slamming phenomenon under rough sea conditions. The impact pressure is transferred to the hull structure and the impact pressure causes structural damage. A strength assessment of the bow region is evaluated based on the empirical formulas according to various classifications. In this study, the numerical simulations were performed to compare with empirical method, and predict the bow flare slamming pressure on the container ship. It is found that the bow flare slamming pressure generated up to 625kPa near 0.975 station and the bow flare slamming pressure obtained from the formulas of the CSR and ABS tend to be similar to the CFD results.

Key words: Slamming load; bow flare slamming; impact pressure; container ship; CFD; KCS.

1. Introduction

Ships operated under rough sea conditions encounter repetitive load by relative motion against the waves from stem to stern. In the process of entering into water, the ship structure is temporarily subjected to the impact pressure. The impact pressures are categorized into bow flare slamming, bottom slamming, stern slamming, and green water loading depending on the position where the impact occurs, and various studies have been carried out through calculations and model experiments. In general, bow flare slamming occurs often in large container ships with a large flare angle, while stern slamming frequently occurs on LNGCs since they have a relatively wide stern, and bottom slamming occurs in tankers because they have blunt stems. Repeatedly applied impact pressure causes structural damage and affects the stability of the vessel.

Hence it is important to estimate the impact load of the slamming phenomenon. Von Karman [1] and Wagner [2] have tried to find an asymptotic solution to the slamming pressure
on a 2D circular cylinder and wedge shapes long ago. Since then, studies based on the potential flow have been conducted to predict the slamming impact pressure by Dobrovol'skaya [3], Zhao and Faltinsen [4], and Muzaferija et al. [5]. However, there is a difficulty in simulating the breaking wave using the potential code when the free surface hits the hull structure. Since then, many studies have tried to estimate the slamming impact pressure including breaking wave by model experiments and Computational Fluid Dynamics (CFD). Wang et al. [6] measured the slamming pressure through a model experiment on drop test of a wedge and tried to investigate the relationship between slamming pressure and dead rise angle. Various studies have been conducted recently, and Hong et al. [7] have examined the model experiment of the drop test on the 2D wedge section and the 2D bow section in particular. Hong et al. [8] also conducted a model experiment to measure bow flare slamming on container ships in regular and irregular waves.

Wang and Soares [9, 10] confirmed that numerical results on slamming pressure agree well with those of drop tests of the wedge through a model experiment. Lee and Lew [11], Kim et al. [12] and Rahaman et al.[13] conducted the calculation using CFD on an LNG carrier and container ship. However, it is still difficult to accurately estimate the impact pressure through numerical simulations, because the slamming pressure occurs suddenly and disappears in a matter of seconds. In particular, it is necessary to keep the time interval sufficiently small in CFD, and the total time of calculation increases significantly.

A significant motion of the vessel and intense free water surface have to be calculated accurately to obtain reliable slamming load. Although studies, which predicted the slamming pressure were conducted often under irregular wave conditions in full scale [14, 15], this is deemed impractical to be applied to the design phase considering the calculation time for simulation. Therefore, simplifications of irregular waves to regular waves under head sea conditions, and simplified interpretations using incident waves have become common [16].

The subject vessel of this study is the the KRISO container ship (KCS) with various experimental data. Recently, experimental results of added resistance in regular waves were published at a CFD workshop held in Japan in 2015, and they were widely used for verification of numerical simulation. Unfortunately, experiments for measuring slamming pressure have not been performed yet, and the results of this numerical study are expected to be the basis of comparable data in terms of prediction of slamming load.

The slamming pressure from the numerical simulation was directly compared with the rule formula proposed by various classification society. These results are expected to be the database for the revision of slamming load formulas in the future.

2. Numerical setup and calculation conditions

The target vessel is the KCS container with a length of 230m. The simulation can be conducted on the full scale, but present numerical simulations were performed on a model scale considering number of grid, calculation time and accuracy. The longitudinal centre of gravity (LCG) is 111.6m based on AP and the vertical centre of gravity (VCG) of gravity is 14.322m above the keel (Table 1).

A grid generation and numerical simulation were carried out using a commercial code STAR-CCM+ Version 11.06. The trimmed mesh method of STAR-CCM+ has the advantage because the mesh size can be set either relatively small for complex flow ranges or large in cases of simple flow ranges through the configuration control of the mesh density in accordance with each flow characteristic used.
Table 1 Principal parameters for the KCS

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Ship</th>
<th>Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scale ratio</td>
<td>1/31.599</td>
<td>7.2786</td>
</tr>
<tr>
<td>LBP [m]</td>
<td>230</td>
<td>7.2786</td>
</tr>
<tr>
<td>B [m]</td>
<td>32.2</td>
<td>1.019</td>
</tr>
<tr>
<td>T [m]</td>
<td>10.8</td>
<td>0.3418</td>
</tr>
<tr>
<td>Δ [m³]</td>
<td>52030</td>
<td>1.649</td>
</tr>
<tr>
<td>LCG</td>
<td>111.6</td>
<td>3.532</td>
</tr>
<tr>
<td>VCG [m], from keel</td>
<td>14.322</td>
<td>0.453</td>
</tr>
<tr>
<td>Vₙ [m/s]</td>
<td>2.572(5kts)</td>
<td>0.458</td>
</tr>
<tr>
<td>Kₓₓ/B</td>
<td>0.4</td>
<td></td>
</tr>
<tr>
<td>Kᵧᵧ/LBP, Kzz/LBP</td>
<td>0.25</td>
<td></td>
</tr>
</tbody>
</table>

The calculation of the spatial gradient of the physical property in a polyhedral grid made according to a trimmed mesh uses the least square method for second order accuracy. Moreover, to simulate the boundary layer flow around the ship surface more accurately, we used the prism layer technique to grid layers with 4.4 million grids from the hull surface by the half-width model ship as shown in Fig. 1. The density of the grid was selected based on the results of 2 dimensional numerical validation.

The generation of the incident wave is an important factor in estimating the slamming pressure accurately. Abdussamie et al. [17, 18] studied the grid dependency on the wave generation and found that more than 30 grids for wave height and 80 grids for wave length were needed to generate the waves well numerically. Therefore, 50 grids for wave height and around 100 grids for the wave length were used in the present numerical simulations.

The numerical calculation used Stokes fifth order wave theory in consideration of the wave height and period. When the motion of the ship is increased, the reflected wave is generated in the direction of FP. As a result, the closer the inlet region is to the ship, the more the reflected waves and the incident waves overlap. It is possible to solve the problem by generating the grid by locating the inlet region far from the ship, but the number of grids is increased excessively.

Kim et al. [19, 20, 21] have solved the problem of reflected waves without increasing the number of grids using Euler Overlay method (EOM). In this study, the Euler Overlay Method (EOM) was applied to simulate waves accurately without numerical damping in the 0.5L of inlet region. In addition, a damping zone region was also applied in the region of 0.5L from the outlet to reduce the influence of the reflected wave due to generated wave. The velocity on the inlet boundary is determined by adding the ship advancing speed to theoretical wave velocity. In this study, the ship advancing speed is 0.458m/s in model scale. In the regions applied EOM, the difference of the water velocity and target velocity is added to the source term of Navier-Stokes equations. In the inlet EOM region, the target velocity is sum of ship advancing velocity and wave velocity. In the outlet EOM region, the target velocity is the ship advancing velocity. The k-ω turbulent model was used and it is one of the most widely used turbulent models for external aero and hydrodynamics. Turbulent specific dissipation rate and turbulent kinetic energy are equal to $1.0 \times 10^{-4}$(1/s) and $1.0 \times 10^{-3}$ (J/kg), respectively.
A symmetric condition was applied to the centre plane, top, bottom and the side of the calculation domain while the wall condition was applied to the surface of the ship. The Volume of Fluid (VOF) method was used to satisfy the free surface boundary conditions reflect the free surface.

A dynamic fluid body interaction (DFBI) scheme was applied to simulate the motion of the ship by the wave. Except for the heave and pitch motions, all other motions were stationary fixed. For more accuracy, an overset grid system was also used to simulate a large ship motion by waves. An overset mesh allows large and complex motions and moving parts to be easily set up and simulated. An overset mesh typically containing a body of interest such as a ship is superimposed on a background mesh containing the surrounding geometry, and data is interpolated between each other the two. The overset mesh system have been often used using by researchers for slamming phenomena or ship motion due to wave [22].

In hash environment, the operators usually reduce the ship speed to decrease the motions and wave loads. The standard velocity for slamming load prediction is suggested the significantly low speed as 0.25% of design speed when significant wave height is more than 12m in ABS rule [23]. In addition, International Maritime Organization (IMO) is considering
enforcing the minimum power propulsion power to maintain a ship speed of at least 5 knots for ships operating under rough sea conditions [24]. In the present research, the ship advancing speed was set to 5 knots the minimum speed keeping a ship maneuverable.

An equivalent design wave method was used to select the target wave height and the target wave period was considered as relative motion and velocity. Six degrees of freedom motion of the KCS container ship used KR3D owned by KRS, the motion analysis program based on potential theory, owned by the KRS. Based on the calculated motion of the ship, the response amplitude operators (RAOs) were derived for relative motion and velocity of the FP and incident wave (Fig. 1 Fig. 2).

The relative motion and relative velocity were the highest generated at wave direction of 120deg, 150deg and 180deg in Fig. 2. Since it is not known which wave direction is generated the maximum slamming pressure, all conditions of wave direction should be considered for nonlinear problems such as slamming.

In the case of a ship with a bulbous, it is known that the greatest slamming pressure is occurred at the head sea condition (180°) due to the bulbous shapes. It has the effect of increasing the angle of attack of flow on hull surface [25]. Hence the wave heading that had the greatest impact on the slamming at the bow flare was selected as incident wave of 180°. However, it is necessary to accurately predict the maximum slamming pressure in various wave directions in future.

To determine the wave height of the incident wave, North Atlantic Standard Wave Data (IACS Rec. No. 34) and PM Spectrum were used to calculate the long term with the probability level of $10^{-8}$ and then divide that by the maximum value of relative motion. Numerical calculation was carried out under the condition of 12.57s (wave period), 20.7m (wave height) as shown in Table 2.
### Table 2 Test conditions for slamming simulation

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship speed ((V_s))</td>
<td>5knots</td>
</tr>
<tr>
<td>Wave height ((H))</td>
<td>20.7m</td>
</tr>
<tr>
<td>Wave period ((T))</td>
<td>12.57sec</td>
</tr>
<tr>
<td>Wave heading</td>
<td>180deg</td>
</tr>
</tbody>
</table>

### 3. Validation of the numerical setup

Impact pressure prediction through the CFD analysis is generally known to be the most influenced by the time interval \((dt)\) and the grid resolution [25]. The time interval is known to be the most dominant factor affecting the impact pressure prediction through the CFD analysis in general. In this study, prior to estimating the slamming pressure on the 3D vessel, the validity of the numerical calculation was investigated in order to improve the accuracy of the impact pressure estimation on the free fall of a wedge-shaped body with reliable experimental results.

#### 3.1 Target object

A wedge-shaped body was selected to examine the validity of numerical setup. The length and breadth of the wedge-shaped body are 800mm and 600mm as shown in Fig. 3. The deadrise angle of wedge is 30 degrees and the wedge-shaped body was dropped at a point of 0.5m from free surface. For the measuring point and further detailed information, refer to Kim et al. [27].

![Fig. 3 Experiment on the 2D wedge [27]](image)

#### 3.2 Influence of the grid size

A numerical simulation was carried out to investigate the effect of the grid size on the prediction of the impact pressure as shown in Fig. 4. The grid size is defined as \(B/Y_{cell}\). \(B/Y_{cell}\) of the coarse mesh, medium mesh and fine mesh are 60, 120 and 240, respectively, as shown in Table 3. \(B\) is the breadth of the wedge-shaped body, and \(Y_{cell}\) is the minimum grid size in the direction of the breadth. The time interval of the numerical simulation is set equal to \(5 \times 10^{-5}\) (2 0kHz), which is the same value as the measured velocity of the sensor used in the model test.
Numerical Study on the Prediction of Bow Flare Slamming Load on Container Ship in Regular Wave  

Dae-Won Seo, Kwang-Leol Jeong

**Fig. 4** Grid system for various numbers of cells

![Grid system for various numbers of cells](image)

**Table 3** Test conditions of the various grid systems

<table>
<thead>
<tr>
<th></th>
<th>$\Delta x$, $\Delta y$ [m]</th>
<th>B/Y cell</th>
<th>dt [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coarse</td>
<td>0.01</td>
<td>60</td>
<td></td>
</tr>
<tr>
<td>Medium</td>
<td>0.005</td>
<td>120</td>
<td>$5\times10^{-5}$</td>
</tr>
<tr>
<td>Fine</td>
<td>0.0025</td>
<td>240</td>
<td></td>
</tr>
</tbody>
</table>

The influence of the grid size on the estimation of the impact pressure is shown in **Fig. 5**. As shown in the figure, the results of the numerical simulation show that the maximum pressure and characteristics are similar to the experimental results. In all cases of the numerical calculations, pressure oscillations were generated as compared to the experiment, but this was because experimental results were processed through a filter. Above the medium grid size, the peak pressure value was found to match that of the experimental values within a range of approximately 3%. However, the point of occurrence of the peak pressure and the absolute value in case of coarse mesh were found to be significantly different from those of the experimental results. Based on the results of this analysis, the grid size was constructed to be greater than 120 of B/Y cell when calculating the slamming impact pressure of a three-dimensional vessel in this study.

**Fig. 5** Comparison of pressure history on various grid systems

![Comparison of pressure history on various grid systems](image)
3.3 Influence of the time interval

It is known that the estimation of the impact pressure acting on the hull for an extremely short period of time is greatly affected by the time interval in numerical calculation. Therefore, numerical simulations for four cases were additionally carried out to examine the influence of the time interval on the size and aspect of the impact pressure. From the study on the free fall experiment of a wedge-shaped body by Kim et al. [27], the Courant number is 0.006 when the time interval is the smallest with approximately $1\times10^{-5}\text{s}$. As such, we performed the calculation under four conditions of the Courant number $=0.006$, which divided the rise time by 100, Courant number $=0.03$ which divided the time step by 20, Courant number $=0.06$, which divided the time interval by 10, and Courant number $=0.6$, which was the time interval, as shown in Table 4.

<table>
<thead>
<tr>
<th>dt [s]</th>
<th>Courant No.</th>
<th>$\Delta x, \Delta y$ [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$1\times10^{-5}$</td>
<td>0.006</td>
<td>0.005</td>
</tr>
<tr>
<td>$5\times10^{-5}$</td>
<td>0.03</td>
<td></td>
</tr>
<tr>
<td>$1\times10^{-4}$</td>
<td>0.06</td>
<td></td>
</tr>
<tr>
<td>$1\times10^{-3}$</td>
<td>0.6</td>
<td></td>
</tr>
</tbody>
</table>

**Fig. 6** shows the estimated calculated results of the impact pressure measured over time in the experiment and from the numerical simulation. As seen in the figure, the predicted maximum pressure value was considerably lower than the experimental results when the Courant number $=0.06$. However, the impact pressure pattern tends to be similar to the results of the model test in all cases except for this case. In particular, when the Courant number was lower than 0.03, the maximum impact pressure from the numerical results was in good agreement with the experimental results quantitatively. When the Courant number was 0.06, the maximum peak pressure was about 30kPa and 10% lower compared to that of the model test, but rise time and the duration of pressure from the numerical results were shown to be almost the same as the experiment. From the perspective of the efficiency of the numerical calculation, it was decided that the results of the numerical calculation were also meaningful even when the Courant number was 0.06. As we determined that the case when the Courant number $=0.06$ was also significant for the efficiency of the numerical analysis, we used 0.06 for the Courant number to calculate the slamming of the ship.

**Fig. 6** Comparison of the pressure history on various time intervals
4. RESULTS OF THE NUMERICAL CALCULATIONS

4.1 Incident wave and ship motion

In ship operation under the rough sea conditions, the impact pressure is occurred on the hull because of the relative motion with incident waves. Hence, it is necessary to implement accurate incident waves for numerical simulation of slamming phenomena. Fig. 7 compares the wave elevation measured by the present calculation that obtained by a theoretical method at a 0.5L distance away from the FP, in the direction of the entrance of the vessel towards the stern. As shown in this figure, it can be confirmed that the incident wave to be generated is properly implemented in numerical simulation. This is because the waveform compensation using the EOM method and the generated grid system has the sufficient grid density per wave length and height.

In this study, motions of the vessel caused by 180° incident waves were performed in heave and pitch motion as free condition in head sea, as shown in Fig. 8. For the stability of the numerical solution, the motion of the ship was virtually forced for about 2 seconds. The pitch angle and heave motion of the ship change from −6.3 degrees to 7.6 degrees and from −0.103m to 0.046m with regular waves, respectively.

![Fig. 7 Comparison of the wave height histories generated by the CFD and theory method (Vs=5knots, H=20.7m, and T=12.57s)](image)

![Fig. 8 Time histories of the heave and pitch motion on the KCS ship (Vs=5knots, H=20.7m, and T=12.57s)](image)

4.2 Bow flare slamming pressure

As shown in Fig. 9, it was divided into 5 sections from FP to 0.85st, and divided into 5 sections based on the draft to measure the impact pressure. The bow impact pressure was measured at a total of 25 measuring points.
Fig. 9 Pressure measuring points of numerical simulation

Fig. 10 shows the slamming pressure acting on the 25 points as described above. A high impact pressures larger then 10kPa was mainly observed near the free surface as shown in Fig. 10 (a, b, c). In the upper region, the impact pressures were not high as shown in Fig. 10 (d) and 10(e). The highest peak pressure was measured at FP 12 point around 14.75 seconds.

(a) Probe number of 1 – 5
(b) Probe number of 6 - 10
(c) Probe number of 11 – 15
(d) Probe number of 16 - 20
(e) Probe number of 21 - 25

Fig. 10 Pressure time history at specific points
Fig. 11 shows the time history of the pressure at the FP8 point where the maximum peak pressure is observed. As shown in the figure, the impact pressure temporarily increases and then disappears in region A. In region B, the static pressure is constantly maintained after the impact pressure has passed. The total duration time of the impact pressure is expressed as the sum of the rise time ($t_{\text{rise}}$) and drop time ($t_{\text{drop}}$), which is about 0.023 seconds (0.13s in the full scale). The duration time of the B region due to the static pressure is approximately 0.93 seconds (5.23s in full scale). The maximum peak pressure acting on FP8 point is approximately 19.78 kPa and is approximately 625kPa in the full scale.

Fig. 11 Pressure time history at FP8

Fig. 12 shows the free surface and hull surface pressures at around 14.75 seconds where the maximum pressure occurs. As the ship descends below the free surface, the impact pressure on the bow region begins to increase around 14.75s (Fig. 12(b)), and then disappears (Fig. 12(c)). Then only the static pressure remains as the bow descends below the free surface by pitch motion (Fig. 12(d)).

Fig. 12 Pressure distributions and free surface

Fig. 13(a) shows pressure distributions on the hull surface at 14.75 seconds when the maximum pressure occurs and Fig.13(b) shows the maximum peak pressure on the probe
points. As the highest impact pressure was generated near the FP 0.975 station as shown in the figure, a structural assessment of this region is necessary.

(a) Pressure distributions at 14.75s (b) Maximum peak pressure on the probe points

**Fig. 13** Pressure distributions on the hull surface

5. **CLASSIFICATION RULES**

5.1 Slamming pressure formula for each classification

As shown in Table 5, each classification has proposed a formula of the slamming pressure acting on the hull. All formulas of the classification adequately take into account the water-entry impact pressure due to vertical relative motion and the breaking wave impact pressure due to the velocity component in the forward direction. In this study, the slamming pressure by IACS, ABS, BV, DNV, LR, and NK were compared with and reviewed the present calculation results.

<table>
<thead>
<tr>
<th>Class</th>
<th>Rule</th>
<th>Bow flare</th>
<th>Bottom</th>
<th>Stern</th>
</tr>
</thead>
<tbody>
<tr>
<td>IACS</td>
<td>Bulk, Tanker</td>
<td>O</td>
<td>O</td>
<td>-</td>
</tr>
<tr>
<td>ABS</td>
<td>Container</td>
<td>O</td>
<td>O</td>
<td>-</td>
</tr>
<tr>
<td>BV</td>
<td>Container</td>
<td>O</td>
<td>O</td>
<td>O</td>
</tr>
<tr>
<td>DNV</td>
<td>Ships</td>
<td>O</td>
<td>O</td>
<td>O</td>
</tr>
<tr>
<td>LR</td>
<td>Ships</td>
<td>O</td>
<td>O</td>
<td>O</td>
</tr>
<tr>
<td>NK</td>
<td>Steel ship</td>
<td>O</td>
<td>O</td>
<td>-</td>
</tr>
</tbody>
</table>

The common structural rule (CSR) in the international classification society defines the bow slamming pressure equation as shown in Eq (1). The CSR is a common rule developed for cargo ships and tankers, but it was also applied to container ships for relative comparison and analy
The main feature for the slamming pressure prediction is the longitudinal bow flare impact pressure distribution factor \( f_{FB} \). \( f_{FB} \) is from 0.55 to 1.0 depending on the considered longitudinal position. \( V_{lm} \) represents impact velocity, which is a function of LBP, Vs, and the waterline angle. \( \gamma_{wl} \) means the local bow impact angle derived from flare angle and water line angle.

\[
P_{FB} = 1.025 f_{FB} C_{FB} V_{lm}^2 \sin \gamma_{wl}
\]

where,

- \( f_{FB} \): longitudinal bow flare impact pressure distribution factor
- \( V_{lm} \): impact speed
  \[
  = 0.514 V_{ref} \sin \alpha_{wl} + \sqrt{L}
  \]
- \( C_{FB} \): coefficient on considering point
- \( \gamma_{wl} \): local bow impact angle
  \[
  = \tan^{-1}(\tan \beta_{pl}/\cos \alpha_{wl})
  \]
- \( \alpha_{wl} \): flare angle at the calculation point
- \( \beta_{pl} \): water line angle at the calculation point

The rule of the ABS classification provides an empirical formula of the bow flare slamming pressure due to wave, and also provides the bow flare slamming pressure as shown in Eq. (2) [29]. \( M_{vi} \) is a function of LBP, Vs, and \( C_{b} \); \( K_{ij} \) is a function of the waterline angle and flare angle. Unlike other classification rules, the breadth \( B_i \) at the considered point is reflected, and this uniquely requires the breadth of the bow shape.

\[
P_{ij} = P_{oij} \text{ or } P_{bij}
\]

\[
P_{oij} = k_1 (9M_{Ri} - h_{ij}^2)^{1/2}
\]

\[
P_{bij} = k_2 k_3 \left[ C_2 + K_{ij} M_{vi} \left( 1 + E_{ni} \right) \right]
\]

Where,

- \( k_1 = 9.807 \)
- \( k_2 = 1.025 \)
- \( k_3 = 1 \) for \( h_{ij} \leq h_b^* \)
  \[
  = 1 + (h_{ij}/h_b^*-1)^2 \text{ for } h_b^* < h_{ij} < 2h_b^*
  \]
  \[
  = 2 \text{ for } h_{ij} \geq 2h_b^*
  \]
- \( C_2 = 39.2 m \)
- \( M_{vi} = B_i M_{Ri} \) \( (M_{Ri} = c_1 A_i (VL/C_b)^{1/2} \)
- \( K_{ij} = f_{ij} [(r_j/(bb_{ij} + 0.5h_{ij}))^{1/2}[l_{ij}/r_j][1.09 + 0.029V - 0.47C_b]^2
  \]
- \( f_{ij} = [90/\beta_{ij}' - 1]^2[\tan^2(\beta_{ij}')]9.86\cos \gamma
  \]
- \( E_{ni} = \log n_{ij}\)
- \( n_{ij} = 5730(M_{vi}/M_{ni})^{1/2} G_{ij} \geq 1.0 \)
The rule of the BV classification provides an empirical formula of the bow flare slamming pressure as shown in Eq (3). The feature of this formula is that the characteristics of a structural part ($C_s$) are considered when calculating bow flare slamming pressure, but the length, direction, and location have no relation to the slamming pressure [30]. The coefficient ($C_z$) that reflects the upward position is applied to consider relative motion, but the vessels of this study have all constant values.

$$P_{fl} = C_s C_z (0.22 + 0.15 \tan \alpha) \times (0.4V \sin \beta + \sqrt{L})^2$$  (3)

Where,

- $C_s = 1.8$ for plating and ordinary stiffeners
- $= 1.0$ for primary supporting members
- $C_z = H - 0.5(z - T_{LC})$ for $z \geq 2H + T_{LC} - 11$
- $= 5.5$ for $z < 2H + T_{LC} - 11$
- $H$: Wave parameter
- $\alpha$: flare angle at the calculation point
- $\beta$: water line angle at the calculation point

Equation 4 is the empirical formula of the bow flare slamming pressure in the DNV classification and is similar to the rule of BV. $C_f$ is the factor that reflects the roll and pitch angle of the ship, and design load of rule should be referred to [31].

$$P_{sl} = C (2.2 + C_f) \times (0.4V \sin \beta + 0.6\sqrt{L})^2$$  (4)

where,

- $C = 0.18(C_w - 0.5h_o)$, maximum 1.0
- $C_w$: wave coefficient
- $h_o$: vertical distance(m) from the waterline at draught
- $C_f = 1.5\tan(\alpha + \gamma)$, maximum 4.0
- $\gamma = 0.4(\rho \cos \beta - \theta \sin \beta)$
- $\alpha$: flare angle at the calculation point
- $\beta$: water line angle at the calculation point

In the LR classification, the empirical formula of the bow flare slamming pressure is provided as shown in Eq. (5). The formula is a function of the coefficient considering the bow shape and the relative vertical velocity [32].

$$P_{bf} = 0.5 \left( K_{bf} V_{bf}^2 + K_{rv} H_{rv} V_{rv}^2 \right)$$  (5)
Numerical Study on the Prediction of Bow Flare Slamming Load on Container Ship in Regular Wave

Dae-Won Seo, Kwang-Leol Jeong

Where,

\( K_{bf} \): hull form shape coefficient for wave impacts

\[
K_{bf} = \begin{cases} 
\frac{\pi}{\tan \psi} & \text{for } \psi \geq 10 \\
28(1-\tan(2\psi)) & \text{for } \psi < 10
\end{cases}
\]

\( \psi \): effective deadrise angle

\( V_{bf} \): threshold velocity for wave impact

\[
V_{bf} = \frac{\sqrt{10}}{\cos \alpha_p}
\]

\( K_{rv} \): hull form shape coefficient for impact due to forward speed

\[
K_{rv} = \begin{cases} 
\frac{\pi}{\tan(90-\alpha_p)} & \text{for } \alpha_p \leq 80 \\
28(1-\tan(2(90-\alpha_p))) & \text{for } \alpha_p > 80
\end{cases}
\]

\( H_{rv} \): relative wave heading coefficient

The empirical formula of the bow slamming pressure in the NK classification is defined as shown in Eq. (6) [33].

\[
P = \frac{1}{2} \rho C_e K_p \left( \frac{v_n}{\cos \beta_0} \right)^2
\]

(6)

Where,

\( \beta_0 \): relative impact angle between wave surface and a point under consideration on ship’s surface

\[
\beta_0 = \Phi - \Phi_b
\]

\( \Phi = \tan^{-1}(1/\tan \beta_k \cos \gamma) \)

\( C_e = \begin{cases} 
\beta_0/40 + 0.25 \text{ for } \beta_0 \leq 30^\circ \\
1.0 \text{ for } \beta_0 > 30^\circ
\end{cases} \)

\( v_n \): Maximum relative velocity

5.2 Comparison of the bow flare slamming pressures

The empirical formula in various classifications provide the quasi static pressure without impact pressure. Hence, it is unreasonable to compare the maximum impact pressure calculated in CFD with the quasi static pressure calculated using the empirical formula. Though it is decided insignificant to make the direct comparison of the absolute value of the quasi-static pressure proposed by a classification with the maximum value of the dynamic pressure derived from numerical calculations, the maximum value of the dynamic pressure from the numerical calculations and the slamming pressure are shown in Fig. 14, to observe qualitative characteristics in investigating the tendency of the slamming pressure on the hull.
Numerical Study on the Prediction of Bow Flare Slamming Load on Container Ship in Regular Wave

The numerical results show that the bow flare slamming pressure increases from 0.9 St to FP and the maximum impact pressure is generated at 0.975St. The tendency of the pressure to increase towards the forward direction was similar to the results of ABS and CSR. Except for other results of the ABS and CSR rule, it is found that the slamming pressure is rather decreased in the direction of the FP, which demonstrated a relatively different tendency from the results of the numerical calculations.

6. CONCLUSION

Prior to a numerical simulation on a container ship, a numerical calculation was conducted to examine the validities of the numerical code on a reliable model test. A numerical simulation was performed to predict the bow slamming pressure on the container ship under extreme wave conditions (H=20.47m, T=12.57s) and these results were compared to the empirical formulas of different classifications.

1) In order to obtain relatively quantitative numerical calculation results, the grid density (B/Y\_cell) should be 120 or greater, and the Courant number should be 0.06 or smaller.

2) The position that generated the maximum impact pressure is around 0.975St., and the maximum impact pressure is measured momentarily at FP8 near the bow region. The maximum peak pressure acting on the FP8 point is approximately 19.78kPa and is approximately 625kPa in the full scale. The duration time of the impact pressure is 0.13 seconds in the full scale. The static pressure continues to act as the bow descends below free surface by pitch motion.

3) Though it is unreasonable to directly compare the absolute value of the quasi-static pressure proposed by a classification with the maximum value of the dynamic pressure derived from numerical calculations, from the point of view of qualitative tendency, the results of the ABS, and CSR classification showed that the impact pressure was significantly increased in the FP direction as the results of the numerical simulation.

In this study, the numerical simulation was performed under one wave condition (head sea) in order to predict the bow flare slamming pressure of the KCS ship. Since slamming
phenomenon has a high nonlinearity, it is not known under which conditions the maximum slamming pressure will be generated. Hence it is necessary to carry out additional calculation in various wave conditions and ship speed in future.

REFERENCES

Numerical Study on the Prediction of Bow Flare Slamming Load on Container Ship in Regular Wave


[28] IACS: Common Structural Rules for Bulk Carriers and Oil Tankers, Part 1, Chapter 4, Section 5, 2014.


[31] DNV-GL: Rule for Classification of Ships- Hull structural design Ships with length 100 meters and above Pt.3 Ch. 1, Sec. 7, 2016.


Submitted: 10.06.2018. Daewon Seo, dwseo@krs.co.kr
Ship & Offshore Technology Center, Korean Register, Republic of Korea
Accepted: 03.01.2019. Kwang-Leol Jeong, kl.jeong@nextfoam.co.kr
Research center, NEXTfoam Co., Ltd., Republic of Korea
NUMERICAL SIMULATION OF THE CAVITATING FLOW AROUND MARINE CO-ROTATING TANDEM PROPELLERS

UDC 629.5.016.8:629.5.03:532.528
Original scientific paper

Summary

In the present paper, a numerical simulation has been carried out to determine the hydrodynamic characteristics in cavitating viscous flow of the conventional INSEAN E779A propeller in single and tandem configuration by using Singhal et al. cavitation model implemented in FLUENT Software. Firstly, calculations have been carried out on single E779A propeller in non-cavitating and cavitating flows. The computed performances have shown good agreement with experimental data. Next, the numerical approach has been applied in loaded conditions to the case of tandem propeller configurations with respectively 0.2 and 0.6 axial displacement. Results reveal that cavitation is qualitatively well predicted and the cavitation area is rather more pronounced on the fore propeller. The use of tandem co-rotating propeller in loaded conditions is highlighted.

Key words: Numerical simulation; marine tandem co-rotating propellers; cavitating viscous flow; Singhal et al. cavitation model; open water performances

1. Introduction

Cavitation has significant impacts on marine propellers. Indeed, it causes the erosion of the blade therefore destroys the propeller. Furthermore, it can affect the efficiency and the thrust, considerably limiting the operation of propulsion system. The costs of the systems involved in the ship propulsion that might be damaged by this phenomenon are enormous. It is very difficult to avoid completely these effects but reducing it as much as possible. To do so, more attention should be given for the design phase of the propeller.

Understanding and analyzing the cavitation phenomenon has been proved to be very important for the proper design of marine propellers. The appropriate technique is coupling of experimental and numerical studies [1]. Several authors have studied the general aspect of the cavitation flows [2-5]. Some others have interested on specific aspects of cavitation flows such as: bubble formation and dynamics, erosion, acoustics associated with noise due to implosion and bubble collapse, rotating cavitation, cavitation Vortices ... etc.
F. A. Pereira et al. [6], experimentally, studied a propeller operating in non-uniform wake. The authors establish quantitative correlations between the near-field pressure and the cavitation model that occurs on the propeller blades. Y. Chang et al. [7] carried out an experimental study on the flow around a marine propeller in a cavitation tunnel. Authors in [8] investigated the cavitation structure of the INSEAN E779A propeller in a uniform flow by using experimental velocimetry (PIV). In the work of R. Arazgaldi et al. [9] cavitation is also experimentally and numerically studied around two different types of marine propellers.

The flow around the marine propeller is known to be unsteady, highly turbulent and occasionally cavitating. The choice of propeller type for a particular propulsion application may be the result of several considerations, i.e., maximum efficiency, noise reduction, manoeuvrability, installation cost and minimization of cavitation risk.

Stefano Gaggero et al. [10] optimize the propeller design by using the multi-objective numerical approach. Authors improve the propeller efficiency and reduce the cavitation extension for high-speed craft propeller.

Computational Fluid Dynamic (CFD) might be the appropriate tools to investigate the performance of tandem co-rotating propellers and thus identifying the optimum conditions for their applications in naval propulsion. D. Boucetta and O. Imine [11] performed a numerical simulation that explores the hydrodynamic behaviour of co-rotating tandems using the RANS approach.

Qin Sun et al. [12] investigate the hydrodynamic characteristics of tandem propellers and demonstrate their range of application to ships. A simplified practical design approach has been proposed which, together with the experiments, has been helpful in assessing the importance of some propeller design parameters. Open water design charts have been produced by testing two model tandem propellers with an axial relative position less than 0.3. The obtained results confirm that propulsive power, bollard pull and vibration levels were better than those of conventional propeller.

Authors in [13] studied the design and performance of tandem propeller device which has good cavitation characteristics. The experiments show that it is possible to design efficient tandem propellers with large number of blades and difference pitch ratio between the aft and the fore propellers greater than 0.2.

Since experimental and numerical analysis, devoted to this type of thrusters, especially those dealing with the cavitation problem, are very scarce and less detailed, a comprehensive study based on a numerical approach is proposed in the following in order to identify the optimal use of co-rotating tandem marine propellers even in the presence of cavitation.

The aim of this work is to investigate numerically the steady cavitating flow around co-rotating tandem propeller by using the RANS method. As the first step, the numerical approach proposed has been validated by applying it on the case of single propeller in both non-cavitating and cavitating conditions. After that, tandem propeller is tested on the cavitating and non-cavitating flow and comparison of results was made with the case of single propellers.
2. Geometric modelling

Three-dimensional flow modelling using CFD has been carried out to investigate the appearance of the cavitating flow as function on the operating conditions. Two propeller devices based on the conventional INSEAN E779A, single propeller and tandem co-rotating configuration, have been considered. The main geometrical parameters of these propellers are summarized in Table 1. According to the study mentioned in reference [11], the propellers used in the tandem geometries have idem diameters and the relative axial displacement between the aft and the forward propeller is equal to 0.2 and 0.6. The angular displacement between the tandem propellers was kept 0°. Figure 1 shows the geometric shape of the INSEAN E779A propeller in single and tandem configurations generated using Gambit.

Table 1 Key design data of the tested propellers

<table>
<thead>
<tr>
<th>Model name</th>
<th>INSEAN E779A single propeller</th>
<th>Tandem co-rotating propeller</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blades number</td>
<td>4</td>
<td>(4+4)</td>
</tr>
<tr>
<td>Diameter (m)</td>
<td>0.22727</td>
<td>0.22727</td>
</tr>
<tr>
<td>Pitch ratio at 0.7R</td>
<td>1.1</td>
<td>(P_{Fore}/D)=(P_{Aft}/D)- 0.2</td>
</tr>
<tr>
<td>Expanded area ratio</td>
<td>0.689</td>
<td>(0.689+0.689)</td>
</tr>
<tr>
<td>Skew</td>
<td>4°48’</td>
<td>4°48’</td>
</tr>
<tr>
<td>Rake</td>
<td>4°35’</td>
<td>4°35’</td>
</tr>
</tbody>
</table>

Single INSEAN E779A propeller  Co-rotating tandem propeller

Fig. 1 3D view of tested propellers blades

3. Mesh generation

Taking into account the periodicity condition, the computational domain was created for only one blade. The proportions of the domain were chosen according to the studies cited in [14]. Due to the complexity of marine propeller geometry, unstructured tetrahedral mesh has been adopted using the Gambit pre-processor. A mesh refinement zone is defined near the propeller blade. The mesh was generated in such a way that cell sizes near the blade wall were small and increased progressively towards outer boundary. Finally, calculation domain meshed by tetrahedral meshes has been obtained, as shown in Figure 2. In the case of cavitation, the mesh is designed so as not to over-fine the wall (boundary layers) in order to
avoid too excessive calculation time and agree well with the chosen turbulence model (K-\(\varepsilon\) Standard) as it is reported in [8]. Particular attention was also paid to the control of element skewness to ensure a good quality of the produced mesh. The number of cells thus generated is 903000 equivalent to a number of nodes of 182000.

![Fig. 2 Boundary condition and mesh over the computational domain](image)

**4. Physical conditions**

Numerical tests are carried out for the case of propeller in open water conditions. A moving reference frame is assigned to the fluid where the rotational speed of propeller is constant. The inlet condition is represented by the axial velocity, while pressure outlet boundary has been adopted as outlet condition. For the cavitating cases, the static outlet pressure is determined by cavitation number.

In this study, calculations are performed using CFD code Fluent and adopting K-\(\varepsilon\) Standard as turbulence model. The SIMPLE algorithm has been adopted and the schemes were all in the second order. Propellers were tested in cavitating conditions by adopting the Singhal cavitation model implemented in code Fluent. Both non-cavitating and cavitating calculations were performed at a rotational speed of \(n=36\) rps on single and tandem co-rotating propeller configurations in a uniform flow.

The first calculation in cavitating flow was carried out on a model of the INSEAN E779A propeller for an advance parameter \(J=0.71\) and for three cavitation numbers (\(\sigma=1.763\), \(\sigma=2.270\) and \(\sigma=2.775\)). The test conditions were reproduced according to the experimental test carried out in the INSEAN cavitation tunnel [15]. It should be noted that for a given \(J\) leading to a non-cavitating flow, the cavitation can be caused by a decrease in pressure at the outlet while still maintaining the same \(J\).

In the second step, two tandem configurations having respectively a relative axial displacement equal to 0.2 and 0.6 were tested on the cavitating flow for a cavitation number equal 1.763 and an advance parameter \(J=0.71\). These conditions were chosen for a comparison with twin-screw propulsion configuration at loaded conditions.
5. Results and discussions

For the study of co-rotating tandem cavitation, the INSEAN E779A propeller was chosen as a reference due to the availability of experimental data in the literature [16]. In this case, the INSEAN E779A propeller, in single configuration, was first tested and validated in the non-cavitating case.

As a first step, three tetrahedral meshes, different by the number of cells, were generated to test the sensitivity of the solution to the mesh size as resume Table 2. Computation was made at J=0.71 for the open water performance prediction of the conventional E779A propeller. Table 3 compares the computed thrust and torque coefficients on the three grids with the experimental values [17]. In non-cavitating conditions, small discrepancy is observed especially for $K_Q$ between the medium and the fine grid. This tendency seems to be common in most of the RANS CFD simulation for marine propellers [8-16-18].

Table 2 Details of grid size

<table>
<thead>
<tr>
<th>Grid</th>
<th>Type of element</th>
<th>Number of cells</th>
<th>Number of nodes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coarse</td>
<td>Tetrahedral</td>
<td>303721</td>
<td>60236</td>
</tr>
<tr>
<td>Medium</td>
<td>Tetrahedral</td>
<td>641755</td>
<td>122484</td>
</tr>
<tr>
<td>Fine</td>
<td>Tetrahedral</td>
<td>1294365</td>
<td>241077</td>
</tr>
</tbody>
</table>

Table 3 Comparison of predicted and the experimental values of $K_T$ and $K_Q$ for the tested grids

<table>
<thead>
<tr>
<th>Grid</th>
<th>J</th>
<th>$K_T$</th>
<th>$10K_Q$</th>
<th>$K_T$</th>
<th>$10K_Q$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coarse</td>
<td>0.71</td>
<td>0.232</td>
<td>0.443</td>
<td>0.238</td>
<td>0.429</td>
</tr>
<tr>
<td>Medium</td>
<td>0.71</td>
<td>0.233</td>
<td>0.442</td>
<td>0.238</td>
<td>0.429</td>
</tr>
<tr>
<td>Fine</td>
<td>0.71</td>
<td>0.231</td>
<td>0.441</td>
<td>0.238</td>
<td>0.429</td>
</tr>
</tbody>
</table>

For the next calculations, single and tandem co-rotating propeller, the rotational speed of propellers is kept constant and equal to $n=36$ rps. The inlet velocity is changing in such a way to obtain the advance coefficient between J=0.3 and J=1.1.

The results of this simulation are illustrated in Figure 3 and Figure 4. As shown in Figure 3 good agreement between the experimental and numerical data for the tested J values is obtained for the E779A propeller. However, the error slightly increases for the low values of J, 10% for the $K_T$ coefficient and 6% for the $K_Q$ coefficient.

Figure 4 illustrates the performance of the (L/D=0.6) tandem compared to doubled INSEAN E779A propeller. This representation makes it possible to highlight the advantages of using the tandem compared to the double conventional propeller in a ship with twin-screw propulsion.

From the result exam, it is clearly observed that from J=1 the tandem provides a higher thrust than that of two separate propellers accompanied by a slight increase in torque. Moreover, the efficiency of the two technological solutions become equal at J=1.05 after which the tandem maintains its supremacy in thrust and torque, about 16% increase.
In this part, the INSEAN E779A propeller was tested for three more or less pronounced cavitation cases corresponding to rotational speed \( n = 36 \text{ rps} \). The calculations have been performed for three cavitation number \( J = 0.71 \). Results of open water tests were represented in terms of thrust, torque and efficiency coefficients and compared to the available experimental data as it is summarized in Table 4. Based on these results, the simulation of the cavitating flow using the numerical approach proposed in this study is globally satisfactory despite a slight difference being observed for all the coefficients. This fact has also been observed by other authors [15-16] who have concluded that numerical cavitation models reproduce the hydrodynamic coefficients more or less correctly and that an improvement of these models is necessary for the refinement of the results.
Table 4 Comparison of predicted and the experimental values of $K_T$, $K_Q$ and $\eta_0$ in cavitating flow

<table>
<thead>
<tr>
<th>$\sigma$</th>
<th>J</th>
<th>$K_T$</th>
<th>10$K_Q$</th>
<th>$\eta_0$</th>
<th>$K_T$</th>
<th>10$K_Q$</th>
<th>$\eta_0$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.763</td>
<td>0.71</td>
<td>0.250</td>
<td>0.448</td>
<td>0.619</td>
<td>0.255</td>
<td>0.460</td>
<td>0.626</td>
</tr>
<tr>
<td>2.270</td>
<td>0.71</td>
<td>0.250</td>
<td>0.449</td>
<td>0.602</td>
<td>0.255</td>
<td>0.461</td>
<td>0.625</td>
</tr>
<tr>
<td>2.775</td>
<td>0.71</td>
<td>0.251</td>
<td>0.450</td>
<td>0.604</td>
<td>0.256</td>
<td>0.464</td>
<td>0.623</td>
</tr>
</tbody>
</table>

Figure 5 illustrates a comparison of the cavitating area on the propeller blades obtained numerically with the experimental visualizations for the three cavitation numbers [14]. It should be noted that the surface of cavitation area increases with the decrease in the cavitation number which is obvious and it is confirmed experimentally and numerically. Furthermore, the cavitation seems to be very developed at the blade tip and the calculation results are in good agreement with the experimental visualization. However, the cavitation results obtained experimentally shows a slightly different pattern of the flow compared to the numerical results. Indeed, it is observed in Figure 5 that cavitation follows the tip vortex trajectory. Unfortunately, the numerical predictions do not reproduce it faithfully. It should be noted that this fact is observed in all previous numerical studies devoted to cavitation on the same propeller [16-17-19]. The reasons given seem to be related to the quality of the mesh in these regions [19] and to the theoretical model of cavitation adopted for the simulation.
Fig. 5 Comparison of numerical and experimental visualizations of cavitation development on the E779A propeller
The tandem co-rotating propeller was also studied in the case of cavitating flow. In this part, a pair of the conventional E779A propeller fitted in tandem co-rotating configuration is tested. Tandem propeller with identical diameters are spaced axially by a distance \( L/D = 0.6 \). The cavitation simulations are performed for a cavitation number of 1.763 at the same rotational speed and advance parameter. These operational conditions were chosen for a comparison with single propeller behavior in similar loaded conditions. Table 5 shows the detail of the tandem hydrodynamic coefficients, obtained numerically, with and without cavitation. It is obvious that cavitation causes a decrease of marine propellers performances. This fact has been confirmed through the analysis of these results although the observed decrease of the \( K_T \) and \( K_Q \) coefficients is minimal. In addition, the unequal contributions of thrust and torque of the tandem aft and fore propellers, observed in the case of non-cavitating flow, are maintained even in the presence of cavitation.

<table>
<thead>
<tr>
<th></th>
<th>Non-cavitation</th>
<th>Cavitation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( K_T )</td>
<td>( 10K_Q )</td>
</tr>
<tr>
<td>Advance parameter</td>
<td>( J = 0.71 )</td>
<td></td>
</tr>
<tr>
<td>Fore propeller</td>
<td>0.240</td>
<td>0.436</td>
</tr>
<tr>
<td>Aft propeller</td>
<td>0.143</td>
<td>0.338</td>
</tr>
<tr>
<td>Tandem</td>
<td>0.383</td>
<td>0.774</td>
</tr>
</tbody>
</table>

This behavior of the tandem propellers is well elucidated by the pressure contours obtained on the blades in the cavitating and non-cavitating case. Indeed, Figure 6 shows the pressure contours on the faces of the two tandem propellers for \( J = 0.71 \). In the absence of cavitation, the minimum pressure coefficient \( C_{P_{\text{min}}} \) recorded on the upper surface of the aft propeller blade, equal to -1.84, assumes that cavitation would be more intense on this face. Nevertheless, the depression surface is rather extended on the back side of the fore propeller which induces a greater cavitation area on this face. This is obviously well confirmed by the pressure contours on the blades in the cavitating case. Moreover, the examination of these contours also reveals the appearance of a limited zone of strong depression in the lower portion of the aft propeller which could be the seat of a cavitation. All these suggestions or predictions are well attested by the visualization of the vapor area on the tandem blades shown in Figure 7.

The development of cavitation on the tandem blades for the advance parameter \( J = 0.71 \) is shown in Figure 8. Comparing the tandem performances to that of single propeller (Figure 5), maximum cavitation area and vapor fraction are noticed on the single propeller. Results reveal that for the tandem operating case the propellers develop very little cavitation in the back side of the aft propeller localized in the bottom leading edge. This result is justified by the existence of a small suction pressure zone as it is shown in Figure 6.
Boucetta Djahida, Imine Omar

NUMERICAL SIMULATION OF THE CAVITATING FLOW AROUND MARINE CO-ROTATING TANDEM PROPELLERS

Tandem Fore propeller (L/D=0.6), Non-cavitating flow

Tandem Aft propeller (L/D=0.6), Non-cavitating flow

Tandem Fore propeller (L/D=0.6), Cavitating flow

Tandem Aft propeller (L/D=0.6), Cavitating flow

Fig. 6 Distribution of pressure coefficient on the tandem blades in cavitating and non-cavitating flow for J=0.71
Numerical simulation of the cavitating flow around marine co-rotating tandem propellers

Boucetta Djahida, Imine Omar

Testes of cavitation are also applied for another tandem configuration with an axial displacement ratio equal to L/D=0.2, usually used for ship equipped with tandem co-rotating device [12-20]. Simulations are conducted in the same operating conditions in order to compare the obtained results at least qualitatively to those reported in reference [12-13]. In this case, calculations are carried out only for one critical cavitation number (σ = 1.763) and for an advance parameter J=0.71. Table 6 resumes open water performances of the tandem L/D=0.2 in cavitating and non cavitating conditions. It is observed that the fall of the hydrodynamic efficient keeps the same trend as the previous cases. This observation has already been reported in [13-14] which indicates that the extent of cavitation is more pronounced on the fore propeller. In order to diminish the cavitation effect in tandem propellers, authors in [14] suggest that the reduction of the fore propeller diameter could improve the tandem performances in cavitating flow. Otherwise, to achieve the same objective, it is recommended in reference [13-21] to reduce the fore propeller pitch and increase the aft one. These two affirmations suppose that pitch and diameter ratios adjustment promise the optimal tandem design.
Table 6 Computational estimation of the hydrodynamic characteristics of tandem L/D=0.2 propeller

| Advance parameter | Non-cavitation | Cavitation | | | |
|-------------------|----------------|------------|----------------|----------------|
| | $K_T$ | $10K_Q$ | $\eta_0$ | $K_T$ | $10K_Q$ | $\eta_0$ |
| Fore propeller | 0.147 | 0.341 | 0.486 | 0.161 | 0.372 | 0.490 |
| Aft propeller | 0.195 | 0.361 | 0.611 | 0.191 | 0.359 | 0.609 |
| Tandem | 0.342 | 0.702 | 0.550 | 0.352 | 0.727 | 0.549 |

Fig. 8 Development of cavities on the L/D=0.2 tandem blades for $J=0.71$

In order to bring the advantage of replacing conventional propeller by tandem solution even in cavitating conditions, tandem geometries (L/D=0.6 and L/D=0.2) are tested at theirs operational conditions where the tandem propeller give the double single propeller thrust coefficient ($K_T \text{ Tandem} = 2K_T \text{ Single}$). Using $K_T$ curve (Figure 4) and assuming that the single propeller is tested for $J=0.71$, equal thrust condition is realized for tandem propellers (L/D=0.6) in cavitating and non-cavitating cases at the corresponding advance coefficient $J=0.5$. By the same way, the advance coefficient for tandem propellers (L/D=0.2) is determined equal to $J=0.4$. All simulations are performed for the same rotational speed, $n=36$ rps, and for the same cavitation number $\sigma=1.763$. The hydrodynamic coefficients results are presented in table 7 where is observed, in non-cavitating case, a decrease of tandem's advance parameter compared to twin-screw propeller solution. Also, to conserve the same thrust by maintaining rotational speed and propeller diameter constants, the advance velocity diminishes in the case of tandem configuration between 35 and 40%. While the absorbed power remains slightly lower for tandem configuration, about 80%. In cavitating case, a small decrease of $K_T$, about 6%, is recorded for tandem propellers whilst the $K_Q$ increase slightly up to 3% compared to the twin-screw configuration.

The cavitation areas on tandem propellers are presented in Figure 9. As it is clearly seen, the fore propeller exhibits more developed cavitation area than the aft propeller due probably to the difference of $K_T$ of these two propellers. However, the fall of $K_T$ coefficient is approximatively the same on the tandem propellers. Furthermore, the cavitation on the fore
propeller covers up to 30% of the propeller area whilst is only about 15% on the twin propeller configuration.

Table 7 Hydrodynamic characteristics of tandem propellers in cavitating and non-cavitating conditions at the same thrust coefficient

<table>
<thead>
<tr>
<th></th>
<th>Non-cavitation</th>
<th>Cavitation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$K_T$</td>
<td>$10K_Q$</td>
</tr>
<tr>
<td><strong>L/D=0.6, J=0.5</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fore propeller</td>
<td>0.315</td>
<td>0.544</td>
</tr>
<tr>
<td>Aft propeller</td>
<td>0.151</td>
<td>0.348</td>
</tr>
<tr>
<td>Tandem</td>
<td>0.466</td>
<td>0.892</td>
</tr>
<tr>
<td><strong>L/D=0.2, J=0.4</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fore propeller</td>
<td>0.316</td>
<td>0.538</td>
</tr>
<tr>
<td>Aft propeller</td>
<td>0.150</td>
<td>0.352</td>
</tr>
<tr>
<td>Tandem</td>
<td>0.466</td>
<td>0.890</td>
</tr>
</tbody>
</table>

**Fig. 9** Contours of volume fraction on the tandem blades for $\sigma=1.763$
6. Conclusion

In this work numerical simulations have been carried out to study the conventional INSEAN E779A propeller cavitation in single and tandem configuration. Numerical simulation has been successfully developed for INSEAN Propeller E779A in non-cavitating and cavitating flows where the computed axial thrust and torque are in good agreement with experimental data. Also, the cavitation model used in this study responds faithfully to the change of cavitation number. However, this model fails to reproduce suitably all the cavitation types which can possibly appear.

For the tandem co-rotating configuration, results show that the propellers behavior in cavitating flow is qualitatively well predicted and the cavitation area is more pronounced on the fore propeller comparing to the previous experimental studies. Despite the slightly decrease in the tandem efficient for the case of cavitation, the tandem co-rotating propeller confirms its utility in high loaded conditions.

7. Acknowledgments

Authors express their sincere thanks to the INSEAN, and in particular Dr. Francesco Salvatore, for providing the geometry and the experimental measurements of the E779A propeller.

Nomenclature

- **D**: Propeller diameter
- **Z**: Blade number
- **P/D**: Propeller pitch ratio
- **L/D**: Relative axial displacement
- **C_{Pmin}**: Pressure coefficient
- **σ**: Cavitation number
- **n**: Number of propeller revolutions
- **J**: Advance coefficient
- **K_T**: Thrust coefficient
- **K_Q**: Torque coefficient
- **η_0**: Propeller efficiency in open water

REFERENCES


Submitted: 29.11.2017. Boucetta Djahida, djahida.boucetta@yahoo.fr
Imine Omar, imine_omar@yahoo.fr
Accepted: 08.01.2019. Aeronautics and propulsive systems laboratory Department of marine engineering, Faculty of mechanical engineering University of Sciences and Technology of Oran, USTO-MB, Oran, Algeria
EXERGY ANALYSIS OF THE MAIN PROPULSION STEAM TURBINE FROM MARINE PROPULSION PLANT

UDC 629.5.016:629.5.03
Original scientific paper

Summary

The paper presents exergy analysis of main propulsion steam turbine from LNG carrier steam propulsion plant. Measurement data required for turbine exergy analysis were obtained during the LNG carrier exploitation at three different turbine loads. Turbine cumulative exergy destruction and exergy efficiency are directly proportional - they increase during the increase in propulsion propeller speed (steam turbine load). Cumulative exergy destruction and exergy efficiency amounts 2041 kW and 66.01 % at the lowest (41.78 rpm), up to the 5923 kW and 80.72 % at the highest (83.00 rpm) propulsion propeller speed. Increase in propulsion propeller speed resulted with an increase in analyzed turbine developed power from 3964 kW at 41.78 rpm to 24805 kW at 83.00 rpm. Analyzed turbine lost power at the highest propulsion propeller speed is the highest and amounts 3339 kW. Steam content at the main propulsion turbine outlet decreases during the increase in propulsion propeller speed. Exergy flow streams can vary considerably, even for a small difference in propulsion propeller speed. Steam turbine in land-based power plant (high power steam turbine) or in marine steam plant (low power steam turbine) is not the component which exergy destruction or exergy efficiency is significantly influenced by the ambient temperature change. A detail analysis of main propulsion steam turbine from the marine steam power plant at several loads is hard to find in the scientific and professional literature.

Key words: marine steam turbine; exergy analysis; propulsion; marine steam plant

1. Introduction

In ship propulsion nowadays, diesel engines in general (mostly slow speed diesel engines) have a leading role [1,2], due to several significant advantages. The wide presence of diesel engines in ship propulsion enabled the development of different numerical models for investigation of their operating parameters [3] and for optimization of their processes [4].

Steam propulsion, in general, is only slightly present on ships, but it is still the dominant type of propulsion for LNG (Liquefied Natural Gas) carriers [5] due to the specificity of their operation and the transported cargo. Any steam system is usually very complex because it is assembled from a large number of components [6]. The majority of marine steam propulsion
plant components are the same as components in conventional land-based steam power plants, but their operation principle is much more dynamic. Usually, the marine steam propulsion plant consists of two steam generators [7] due to safety operation and two parallel operating turbo-generators [8] to ensure electricity supply at any time. Propulsion propeller (or more of them) drive is ensured with main propulsion turbine [9]. Steam after expansion in turbo-generators and main propulsion turbine goes to the main condenser [10] on liquefaction.

On water return channel to steam generators there are several devices which provide water heating. The first of such devices is evaporator (fresh water generator) [11], the steam marine plant component which is not required in land-based steam plants. After evaporator is usually located sealing steam condenser [12] and two or more feed water heaters [13,14]. Between feed water heaters is located deaerator [15,16] with its dual function - feed water heating and removal of gaseous components from feed water to reduce corrosion. On water return channel are also mounted hot well [17] for collecting all the condensate from the system and desuperheater. Desuperheater is a heat exchanger which is used for steam cooling and preparation for the purpose of heating the cargo and all auxiliary systems [18]. General marine steam propulsion plant scheme of one conventional LNG carrier can be found in [8].

At this moment, new systems for LNG carrier propulsion, which are at least partially based on steam turbines, are under the development [19]. One of the main goals of such propulsion systems is to reduce greenhouse gas emissions at the lowest possible level [20,21,22]. For such complex marine propulsion systems is necessary to provide the economic and profitability analysis [23] as well as operational risk assessment [24] in order to minimize possible harmful consequences.

This paper presents a complete exergy analysis and exergy flow analysis of main propulsion steam turbine from LNG carrier steam propulsion plant. Analyzed turbine has two cylinders (high pressure and low pressure cylinder) and three steam subtractions. Measurement data required for main propulsion turbine exergy analysis were obtained during the LNG carrier exploitation at three different turbine loads (low load -41.78 rpm, middle load -74.59 rpm and high load -83.00 rpm).

The main propulsion steam turbine is analyzed during the harbour leaving until reaching the cruising speed. Therefore, low turbine load (41.78 rpm) represents the beginning of ship acceleration; middle load (74.59 rpm) is a turbine load during ship acceleration and high turbine load at 83.00 rpm is turbine load at the ship cruising speed. Turbine high load (load at the ship cruising speed) usually amounts around 85 % of turbine maximum (full) load because on such turbine load specific fuel consumption is the lowest. In this particular case, the main turbine developed power at high load (83.00 rpm) amounts 84.3 % of maximum turbine power.

Energy and exergy analysis are widely used methods for the investigation of entire steam power plants or its components [8,16]. Equally, such analyses can be used for the investigation of entire ship energy systems – examples can be found in [25] for chemical tanker or in [26] for a cruise ship. Those methods are black-box methods because such analyses do not require knowledge of any component inner structure - the relevant are only energy and exergy inputs and outputs of the component to obtain its efficiency and losses.

Any steam turbine energy analysis gives as a result comparison of how much real (polytropic) steam expansion process deviates from the ideal (isentropic) steam expansion process throughout the turbine. The comparison of these two steam expansion process allows calculation of turbine real and ideal power, after which is calculated turbine energy losses and energy efficiencies. So, the baseline of any steam turbine energy analysis is comparison of real and ideal steam expansion processes, as presented for marine turbo-generators in [8]. Energy analysis does not take into account the ambient conditions in which turbine operate.
Exergy analysis of the main propulsion steam turbine from marine propulsion plant

Vedran Mrzljak, Igor Poljak, Jasna Prpić-Oršić

Exergy analysis of the main propulsion steam turbine at different loads will present a change in turbine exergy efficiencies and losses during the load increase - to evaluate the current turbine operation and identify possible problems. Analysis of exergy flows and steam mass flows throughout the main propulsion turbine at different loads can be used as a baseline for turbine (and entire power plant) optimization. Steam content calculation at the main turbine outlet is commonly used for better protection of turbine blades and for increasing a period between maintenance. Exergy analysis, unlike energy analysis, takes into account the conditions of the ambient in which turbine operates what allows analysis of the main propulsion steam turbine exergy efficiencies and losses at different ambient temperatures. At the end - calculated turbine exergy destruction divided by turbine developed power at any load (specific turbine exergy destruction) can be used for direct comparison of main marine propulsion steam turbine with any other steam turbine.

2. Steam turbine exergy analysis

2.1 General equations for exergy analysis

Mass balance for a standard volume in steady state disregarding potential and kinetic energy can be defined according to [27,28] with an equation:

\[ \sum \dot{m}_{IN} = \sum \dot{m}_{OUT} \quad (1) \]

Exergy analysis of any steam plant component is based on the second law of thermodynamics [8,29]. The exergy balance equation for a standard volume (control volume) in steady state is [30,31]:

\[ \dot{X}_{\text{heat}} - P = \sum \dot{m}_{OUT} \cdot e_{OUT} - \sum \dot{m}_{IN} \cdot e_{IN} + \dot{E}_{\text{ex,D}} \quad (2) \]

where the net exergy transfer by heat (\( \dot{X}_{\text{heat}} \)) at the temperature \( T \) is defined as [32,33]:

\[ \dot{X}_{\text{heat}} = \sum (1 - \frac{T_0}{T}) \cdot \dot{Q} \quad (3) \]

Specific exergy was defined according to [34,35]:

\[ e = (h - h_0) - T_0 \cdot (s - s_0) \quad (4) \]

The total exergy of any fluid stream (exergy power) is defined according to [36,37]:

\[ \dot{E}_{\text{ex}} = \dot{m} \cdot e = \dot{m} \cdot [(h - h_0) - T_0 \cdot (s - s_0)] \quad (5) \]

Exergy efficiency is also called second law efficiency or effectiveness [29,38]. In general, it can be defined by using an equation:

\[ \eta_{\text{ex}} = \frac{\text{Exergy output}}{\text{Exergy input}} \quad (6) \]

2.2 Main steam propulsion turbine exergy analysis

Analyzed main steam propulsion turbine is mounted in the steam propulsion power plant of commercial LNG carrier. Main characteristics and specifications of the LNG carrier, on which the main propulsion turbine is mounted, are presented in Table 1:
Table 1 Main characteristics and specifications of the LNG carrier

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Gross tonnage</td>
<td>100450 tons</td>
</tr>
<tr>
<td>Dead weight tonnage</td>
<td>84812 tons</td>
</tr>
<tr>
<td>Overall length</td>
<td>288 m</td>
</tr>
<tr>
<td>Max breadth</td>
<td>44 m</td>
</tr>
<tr>
<td>Design draft</td>
<td>9.3 m</td>
</tr>
<tr>
<td>Steam generators</td>
<td>2 x Mitsubishi MB-4E-KS</td>
</tr>
<tr>
<td>Main propulsion turbine</td>
<td>Mitsubishi MS40-2 (max. power 29420 kW)</td>
</tr>
<tr>
<td>Turbo-generators</td>
<td>2 x Shinko RGA 92-2 (max. power 3850 kW each)</td>
</tr>
</tbody>
</table>

Main propulsion turbine consists of two cylinders: high pressure (HP) and low pressure (LP) cylinders, Fig. 1. The HP cylinder consists of eight stages (one Curtis and seven Rateau stages - all for ahead drive) while the LP cylinder consists of ten stages (eight Rateau stages for ahead drive and two Curtis stages for astern drive). At the main turbine, Fig. 1, can be seen three steam subtractions – first from HP cylinder, second between HP and LP cylinder and third from the LP cylinder. Steam subtraction mass flows to each steam plant component were regulated by using electronically driven valves from the main engine room control electronic system [39].

At Fig. 1 can also be seen steam stream flow marks, necessary for the main turbine exergy analysis (marked with numbers from 1 to 7). At each steam subtraction point (points 2, 4 and 6) was defined to which steam plant component (or more of them) the steam was leading during subtraction. Steam mass flow taken from the turbine during subtractions is dependable on the current steam propulsion system load, what will be further described during the measurement results presentation.

Both steam turbine cylinders were connected with the same shaft and then connected to the propulsion propeller through the gearbox which reduces propeller speed. The highest propulsion propeller speed (and thus the highest steam system load) observed in this analysis was 83.00 rpm. Propulsion propeller speed is directly proportional to steam system load.

Fig. 1 Main propulsion turbine scheme and stream flow marks

Main propulsion turbine exergy analysis will be defined by using steam stream flow marks from Fig. 1. The real (polytropic) steam expansion for the entire main propulsion turbine (both cylinders) in the h-s diagram, according to stream flow marks on the Fig. 1, is presented in Fig. 2. In the Fig. 2 was marked steam mass flows between all main turbine flow
points as well as steam mass flows subtracted from the turbine. Main turbine steam expansion end (point 7, Fig. 2) is under the saturation line, because after expansion on the main turbine, steam was led directly into the steam condenser. As turbine load increases, steam content at the LP turbine outlet (point 7) decreases.

The entire exergy analysis of the main propulsion turbine was based on the real (polytropic) steam expansion [40]. Cumulative steam mass flow lost on the turbine labyrinth seals [41] in this analysis was neglected.

According to flow stream points and steam expansion presented in Fig. 1 and Fig. 2, mass flow balance, energy (power) and exergy balance equations for the main propulsion turbine analysis are:

- Mass flow balance:
  \[ \dot{m}_1 = \dot{m}_2 + \dot{m}_4 + \dot{m}_6 + \dot{m}_7 \]  

(7)

- Exergy flow of a stream (exergy power of a stream):
  \[ \dot{E}_{ex,n} = \dot{m}_n \cdot \varepsilon_n = \dot{m}_n \cdot (h_n - h_0) - T_0 \cdot (s_n - s_0) \]  

(8)

where index n marks steam stream flows (according to Fig. 1, n = 1...7). The ambient state in the LNG carrier engine room during the measurements was:

- pressure: \( p_0 = 0.1 \text{ MPa} = 1 \text{ bar} \),
- temperature: \( T_0 = 25 ^\circ \text{C} = 298.15 \text{ K} \).

- Main turbine developed power:
  \[ P_{\text{main turbine}} = \dot{m}_1 \cdot (h_1 - h_2) + (\dot{m}_1 - \dot{m}_2) \cdot (h_2 - h_3) + \\
  + (\dot{m}_1 - \dot{m}_4) \cdot (h_5 - h_6) + (\dot{m}_1 - \dot{m}_2 - \dot{m}_4 - \dot{m}_6) \cdot (h_6 - h_7) \]  

(9)
- Main turbine lost power:

\[
P_{\text{main turbine},\text{PL}} = m_2 \cdot (h_2 - h_3) + m_2 \cdot (h_3 - h_6) + m_2 \cdot (h_6 - h_7) + \]

\[
+ m_4 \cdot (h_5 - h_6) + m_4 \cdot (h_6 - h_7) + m_6 \cdot (h_6 - h_7) \tag{10}
\]

- Cumulative exergy power input:

\[
\dot{E}_{\text{ex},\text{IN}} = \dot{E}_{\text{ex},1} = \dot{m}_1 \cdot e_1 = \dot{m}_1 \cdot [(h_l - h_0) - T_0 \cdot (s_l - s_0)] \tag{11}
\]

- Cumulative exergy power output:

\[
\dot{E}_{\text{ex},\text{OUT}} = \dot{E}_{\text{ex},1} + \dot{E}_{\text{ex},2} + \dot{E}_{\text{ex,6}} + \dot{E}_{\text{ex,7}} + P_{\text{main turbine}} = m_2 \cdot e_2 + m_4 \cdot e_4 + \\
+ m_6 \cdot e_6 + m_7 \cdot e_7 + P_{\text{main turbine}} \tag{12}
\]

- Cumulative exergy power loss (cumulative exergy destruction):

\[
\dot{E}_{\text{ex,D}} = \dot{E}_{\text{ex,IN}} - \dot{E}_{\text{ex,OUT}} = \dot{E}_{\text{ex,1}} - \dot{E}_{\text{ex,2}} - \dot{E}_{\text{ex,4}} - \dot{E}_{\text{ex,6}} - \dot{E}_{\text{ex,7}} - P_{\text{main turbine}} \tag{13}
\]

\[
= \dot{m}_1 \cdot e_1 - \dot{m}_2 \cdot e_2 - \dot{m}_4 \cdot e_4 - \dot{m}_6 \cdot e_6 - \dot{m}_7 \cdot e_7 - P_{\text{main turbine}}
\]

- Exergy efficiency:

\[
\eta_{\text{ex}} = \frac{P_{\text{main turbine}}}{\dot{E}_{\text{ex},1} - \dot{E}_{\text{ex,2}} - \dot{E}_{\text{ex,4}} - \dot{E}_{\text{ex,6}} - \dot{E}_{\text{ex,7}}} = \frac{P_{\text{main turbine}}}{m_1 \cdot e_1 - m_2 \cdot e_2 - m_4 \cdot e_4 - m_6 \cdot e_6 - m_7 \cdot e_7} \tag{14}
\]

Steam specific enthalpies and specific entropies were calculated from measured steam pressures and temperatures of each flow stream by using NIST REFPROP software [42].

2.3 Entire marine steam power plant exergy efficiency and destruction

Total exergy destruction of the entire steam power plant (any steam power plant), according to [27] and [31], is the sum of exergy destructions of all components and can be calculated by using an equation:

\[
\dot{E}_{\text{ex,D,plant}} = \sum \dot{E}_{\text{ex,D(all components)}} \tag{15}
\]

while the entire steam power plant exergy efficiency can be calculated as:

\[
\eta_{\text{ex,plant}} = \frac{P_{\text{produced power}}}{\dot{E}_{\text{ex,fuel}}} = \frac{P_{\text{main turbine}}}{m_{\text{fuel}} \cdot \dot{e}_{\text{fuel}}} \tag{16}
\]

In land-based steam power plants produced power is main steam turbine power. If the land-based steam power plant has auxiliary steam turbines, their power is usually much lower than the power produced by main turbine, so the power of auxiliary steam turbines in such steam power plants can be neglected in the equation (16).

For the marine steam power plant at LNG carrier in which analyzed main propulsion turbine operates, exergy destruction of the entire plant can be calculated as presented in equation (15). The equation for exergy efficiency calculation of the entire marine steam power plant at LNG carrier should be modified when compared to equation (16) because of two reasons.

The first reason is that in the marine steam power plant at analyzed LNG carrier along with main propulsion steam turbine operates three auxiliary steam turbines (two turbo-
generators and steam turbine for the main feed water pump drive). Cumulative power produced by auxiliary steam turbines in some marine steam plant operating regimes can notably influenced total produced power. The second reason of the equation (16) correction for the analyzed marine steam power plant at LNG carrier is that both marine steam generators simultaneously use two fuels - heavy fuel oil (HFO) and LNG. Therefore, the equation of exergy efficiency calculation for the entire marine steam power plant at the analyzed LNG carrier, in any load, will be:

$$\eta_{ex,\text{plant}} = \frac{P_{\text{main turbine}} + P_{\text{turbo-generator1}} + P_{\text{turbo-generator2}} + P_{\text{pump}}}{m_{\text{HFO}} \cdot \varepsilon_{\text{HFO}} + m_{\text{LNG}} \cdot \varepsilon_{\text{LNG}}}$$ (17)

For equation (17) it should be noted that power produced by the main steam turbine is the most dominant one during the majority of marine steam power plant operation. HFO and LNG mass flows in equation (17) are cumulative mass flows for both steam generators, while specific exergies of both fuels can be calculated from fuel mass fractions.

Analyzed main propulsion steam turbine from LNG carrier doesn’t have steam re-heating. Therefore, for such entire marine steam propulsion plant can be expected exergy efficiencies between 12 % and 15 % at low main turbine load, around 20 % at middle main turbine load and between 25 % and 30 % at high main turbine load. For comparison, similar marine steam propulsion plant from LNG carrier where main turbine posses steam re-heating has plant exergy efficiency of around 34 % at high main turbine load [16].

3. Main propulsion turbine measurement results and measuring equipment

Measurement results of required steam operating parameters (pressure, temperature and mass flow) for main propulsion steam turbine, according to stream flows – Fig. 1, are presented in relation to the propulsion propeller speed, Table 2 and Table 3. Propulsion propeller speed is directly proportional to main propulsion turbine load, higher propulsion propeller speed denotes a higher steam turbine load. In Table 2 are presented measurements for HP turbine cylinder, while in Table 3 are presented measurements for LP turbine cylinder.

Table 2 Main propulsion turbine measurement results – HP cylinder

<table>
<thead>
<tr>
<th>Stream flow mark (Fig. 1)</th>
<th>Propulsion propeller speed (rpm)</th>
<th>Steam mass flow at the HP turbine entrance (kg/h)</th>
<th>Steam temperature at the HP turbine entrance (°C)</th>
<th>Steam pressure at the HP turbine entrance (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>41.78</td>
<td>16605</td>
<td>488.0</td>
<td>6.190</td>
</tr>
<tr>
<td></td>
<td>74.59</td>
<td>65012</td>
<td>513.5</td>
<td>6.020</td>
</tr>
<tr>
<td></td>
<td>83.00</td>
<td>96474</td>
<td>500.0</td>
<td>5.899</td>
</tr>
<tr>
<td>2</td>
<td>41.78</td>
<td>0</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>74.59</td>
<td>0</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>83.00</td>
<td>3268</td>
<td>350.0</td>
<td>1.565</td>
</tr>
<tr>
<td>3</td>
<td>41.78</td>
<td>16605</td>
<td>243.0</td>
<td>0.151</td>
</tr>
<tr>
<td></td>
<td>74.59</td>
<td>65012</td>
<td>256.0</td>
<td>0.467</td>
</tr>
<tr>
<td></td>
<td>83.00</td>
<td>93206</td>
<td>256.0</td>
<td>0.593</td>
</tr>
</tbody>
</table>
Table 3 Main propulsion turbine measurement results – LP cylinder

<table>
<thead>
<tr>
<th>Stream flow mark (Fig. 1)</th>
<th>Propulsion propeller speed (rpm)</th>
<th>Subtraction steam mass flow between HP and LP turbine (kg/h)</th>
<th>Subtraction steam temperature between HP and LP turbine (°C)</th>
<th>Subtraction steam pressure between HP and LP turbine (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>41.78</td>
<td>0</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>74.59</td>
<td>4690</td>
<td>256.0</td>
<td>0.467</td>
</tr>
<tr>
<td></td>
<td>83.00</td>
<td>13609</td>
<td>256.0</td>
<td>0.593</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Stream flow mark (Fig. 1)</th>
<th>Propulsion propeller speed (rpm)</th>
<th>Steam mass flow at the LP turbine entrance (kg/h)</th>
<th>Steam temperature at the LP turbine entrance (°C)</th>
<th>Steam pressure at the LP turbine entrance (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>41.78</td>
<td>16605</td>
<td>243.0</td>
<td>0.151</td>
</tr>
<tr>
<td></td>
<td>74.59</td>
<td>60322</td>
<td>256.0</td>
<td>0.467</td>
</tr>
<tr>
<td></td>
<td>83.00</td>
<td>79597</td>
<td>256.0</td>
<td>0.593</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Stream flow mark (Fig. 1)</th>
<th>Propulsion propeller speed (rpm)</th>
<th>Steam mass flow of LP turbine subtraction (kg/h)</th>
<th>Steam temperature of LP turbine subtraction (°C)</th>
<th>Steam pressure of LP turbine subtraction (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>41.78</td>
<td>0</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>74.59</td>
<td>2032</td>
<td>156.0</td>
<td>0.097</td>
</tr>
<tr>
<td></td>
<td>83.00</td>
<td>3355</td>
<td>153.0</td>
<td>0.121</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Stream flow mark (Fig. 1)</th>
<th>Propulsion propeller speed (rpm)</th>
<th>Steam mass flow at the LP turbine outlet (kg/h)</th>
<th>Steam temperature at the LP turbine outlet (°C)</th>
<th>Steam pressure at the LP turbine outlet (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7</td>
<td>41.78</td>
<td>16605</td>
<td>32.50</td>
<td>0.00489</td>
</tr>
<tr>
<td></td>
<td>74.59</td>
<td>58290</td>
<td>29.47</td>
<td>0.00412</td>
</tr>
<tr>
<td></td>
<td>83.00</td>
<td>76242</td>
<td>34.92</td>
<td>0.00561</td>
</tr>
</tbody>
</table>

Measurement results were obtained from the existing measuring equipment mounted on both main propulsion turbine cylinders and on all steam subtraction streams. List of used measuring equipment was presented in Table 4.

Table 4 Main propulsion turbine measuring equipment

<table>
<thead>
<tr>
<th>Stream flow mark (Fig. 1)</th>
<th>Steam mass flow (differential pressure transmitters [43])</th>
<th>Steam pressure (pressure transmitters [44])</th>
<th>Steam temperature (immersion probes [45])</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Yamatake JTD960A</td>
<td>Yamatake JTG960A</td>
<td>Greisinger GTF 601-Pt100</td>
</tr>
<tr>
<td>2</td>
<td>Yamatake JTD960A</td>
<td>Yamatake JTG940A</td>
<td>Greisinger GTF 601-Pt100</td>
</tr>
<tr>
<td>3</td>
<td>Yamatake JTD930A</td>
<td>Yamatake JTG940A</td>
<td>Greisinger GTF 401-Pt100</td>
</tr>
<tr>
<td>4</td>
<td>Yamatake JTD930A</td>
<td>Yamatake JTG940A</td>
<td>Greisinger GTF 401-Pt100</td>
</tr>
<tr>
<td>5</td>
<td>Yamatake JTD930A</td>
<td>Yamatake JTG940A</td>
<td>Greisinger GTF 401-Pt100</td>
</tr>
<tr>
<td>6</td>
<td>Yamatake JTD920A</td>
<td>Yamatake JTG940A</td>
<td>Greisinger GTF 401-Pt100</td>
</tr>
<tr>
<td>7</td>
<td>Yamatake JTD910A</td>
<td>Yamatake JTG940A</td>
<td>Greisinger GTF 401-Pt100</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Propulsion propeller speed</th>
<th>Kyma Shaft Power Meter (KPM-PFS) [46]</th>
</tr>
</thead>
</table>
4. Main propulsion turbine exergy analysis results with the discussion

Cumulative exergy power input and output of the main propulsion turbine at different propulsion propeller speeds, calculated by using equations (11) and (12) are presented in Fig. 3. Main turbine cumulative exergy power input and output increases during the increase in propulsion propeller speed (increase in steam system load). The difference between main turbine cumulative exergy power input and output also increases during the increase in propulsion propeller speed, what defines main turbine exergy power losses (exergy destruction).

At the lowest observed propulsion propeller speed of 41.78 rpm, main turbine cumulative exergy power input amounts 6277 kW, while cumulative exergy power output amounts 4236 kW. At propulsion propeller speed of 74.59 rpm cumulative main turbine exergy power input and output amounts 25205 kW and 20098 kW, while at 83.00 rpm cumulative exergy power input and output amounts 36824 kW and 30901 kW. Presented exergy power inputs and outputs in Fig. 3 for every propulsion propeller speed are valid for the ambient state in the LNG carrier engine room during measurements.

Main propulsion turbine cumulative exergy destruction and exergy efficiency are directly proportional - both increases during the increase in propulsion propeller speed, Fig. 4. According to equation (13), increase in main turbine cumulative exergy destruction during the increase in turbine load is caused by a fact that cumulative exergy power input increases faster than cumulative exergy power output. Main turbine exergy efficiency also increases during the increase in turbine load because turbine developed power increases faster than the difference in exergy flows between turbine inlet and outlet, equation (14).

In the observed main turbine load range, cumulative exergy destruction and exergy efficiency amounts 2041 kW and 66.01 % at propulsion propeller speed of 41.78 rpm after which increases to 5107 kW and 78.12 % at 74.59 rpm. At the highest observed turbine load (at the highest observed propulsion propeller speed) cumulative turbine exergy destruction and efficiency are the highest and amounts 5923 kW and 80.72 %.

When the analyzed marine propulsion steam turbine is compared with high power steam turbines from land-based thermal power plants such as turbines in [47] and [48], it can be concluded that analyzed marine turbine has higher exergy destruction in regards to developed power, while its exergy efficiency is lower. Compared to high power steam turbines, marine propulsion steam turbine must be much flexible in operation, especially during maneuvering.
in ports, so its lower exergy efficiency and higher exergy destruction in regards to developed power are expected.

![Fig. 4](image)

**Fig. 4** Change in main turbine cumulative exergy destruction and exergy efficiency at different propulsion propeller speeds - according to measurement state

Main propulsion turbine developed power is calculated by using measured steam mass flows and steam specific enthalpies obtained from measured steam pressures and temperatures, equation (9). Increase in propulsion propeller speed resulted in an increase in analyzed turbine developed power, Fig. 5. At the lowest observed propulsion propeller speed of 41.78 rpm, main turbine developed power amounts 3964 kW after which increases to 18232 kW at 74.59 rpm and at the highest observed propulsion propeller speed of 83.00 rpm turbine developed power amounts 24805 kW. Turbine developed power is an essential element for proper defining cumulative turbine exergy power output, exergy destruction and exergy efficiency, equations (12), (13) and (14). An increase in turbine developed power during the increase in propulsion propeller speed is expected, because measurements of required operating parameters for exergy analysis were performed while LNG carrier leaving the port (41.78 rpm) until the ship reaches its usual navigation speed at open sea (83.00 rpm).

Turbine lost power, defined by an equation (10), is an additional main turbine power which can be developed if the steam was not subtracted from the turbine. If the steam subtractions were closed, additional steam mass flow will expand through the both turbine cylinders (HP and LP) between the same steam specific enthalpies (obtained from measured steam pressure and temperature) and thus more power will be produced. Main turbine steam subtractions are calculated in detail, to ensure continuous steam supply from the main turbine to all necessary steam consumers in the propulsion plant. If the steam subtractions were not in operation, especially at high turbine loads, it is questionable whether the power plant could operate at all.

Therefore, turbine lost power represents the additional power which can be developed for propulsion propeller drive if necessary. Analyzed main propulsion turbine operates in a manner that on lower loads (41.78 rpm) steam subtractions were closed (Table 2 and Table 3) and turbine lost power is equal to 0 kW, Fig. 5. At lower steam plant loads, steam consumers get the necessary amount of steam directly from steam generators.

Increase in steam system and consequentially main turbine load resulted with steam subtractions opening. Steam mass flow subtracted from the main propulsion turbine increases proportionally with an increase in turbine load. As a result, turbine lost power at propulsion propeller speed of 74.59 rpm amounts 987 kW, while at the highest propulsion propeller speed of 83.00 rpm turbine lost power is the highest and amounts 3339 kW because at the
83.00 rpm all steam subtractions are opened and the steam mass flow subtracted from the main turbine is the highest.

![Fig. 5 Main turbine developed and lost power at different propulsion propeller speeds](image)

After expansion in main propulsion turbine cylinders, at the turbine outlet, steam was led directly to the main steam condenser (point 7, Fig. 1). To be able to liquefy that steam in the condenser, its operating parameters must be under the saturation line, in the area of saturated steam. It is interesting to observe the change of steam content at the main turbine outlet, according to measured data from Table 3.

At the lowest propulsion propeller speed of 41.78 rpm, steam content at the main turbine outlet is high and amounts 98.83 %, Fig. 6. Increase in main turbine load resulted with a decrease in steam content at the turbine outlet. At the main turbine outlet, steam content of 93.23 % was observed at propulsion propeller speed of 74.59 rpm, while at the highest load (83.00 rpm) steam content is the lowest and amounts 92.10 %.

The majority of LNG carrier operation during the whole exploitation period can be expected at the highest steam system load (and thus at the highest main turbine load). As presented in Fig. 6, at the highest loads the amount of water droplets in the steam after turbine will be the highest. High amount of water droplets will lead to increased erosion on the turbine rotor blades, especially on the last stages of LP turbine cylinder. Therefore, the rotor blades at the last few stages of LP turbine are covered with special protective linings made of hard materials (usually stellite) for wear protection in order to prolong their replacement period [49,50].

![Fig. 6 Change in steam content at LP turbine outlet during the increase in propulsion propeller speed](image)
Exergy flow streams at each main propulsion turbine stream point, according to Fig. 1, for each observed propulsion propeller speed are presented in Fig. 7. Exergy flow stream at each turbine stream point is calculated by using equation (8) and measured operating data (engine room ambient state). The best explanation of Fig. 7 is to analyze exergy flow streams at each propulsion propeller speed.

At the lowest propulsion propeller speed of 41.78 rpm, exergy flow stream at the turbine inlet (point 1, Fig. 1) amounts 6277 kW. At this propulsion propeller speed all turbine subtractions are closed and exergy flow streams are equal to zero (point 2, 4 and 6, Fig. 1). At the HP turbine cylinder outlet (point 3) exergy flow stream amounts 2922 kW and the same exergy flow stream enters in the LP turbine cylinder, Fig. 7. At the LP cylinder outlet (at the main turbine outlet), exergy flow stream at 41.78 rpm amounts only 272 kW.

The exergy flow stream at the turbine inlet (point 1) at 74.59 rpm is significantly higher in comparison with the lower propulsion propeller speed and amounts 25205 kW, Fig. 7. Steam subtraction at the HP turbine cylinder (point 2) is closed also at 74.59 rpm. At the HP turbine cylinder outlet (point 3), exergy flow stream amounts 14338 kW. At the LP turbine cylinder inlet (point 5), exergy flow stream amounts 13304 kW what is lower than an exergy flow stream at the HP turbine cylinder outlet (point 3) for exergy flow stream subtracted between HP and LP turbine cylinders which amounts 1034 kW (point 4) at this propulsion propeller speed. At the LP turbine cylinder inlet (point 5), exergy flow stream amounts 288 kW and finally, at the LP turbine cylinder outlet (point 7) exergy flow stream is equal to 543 kW. If compared propulsion propeller speeds of 41.78 rpm and 74.59 rpm, exergy flow streams are significantly higher at higher propulsion propeller speed for each turbine stream point (with an exception of HP turbine cylinder subtraction which is closed in both cases).

The highest exergy flow streams at each turbine stream point can be observed at the highest propulsion propeller speed of 83.00 rpm, Fig. 7, although the difference in propulsion propeller speeds of 74.59 rpm and 83.00 rpm is not significant. At the turbine inlet (point 1) exergy flow stream amounts 36824 kW at 83.00 rpm. Only at the highest observed propulsion propeller speed, steam subtraction at the HP turbine cylinder (point 2) is open and exergy flow stream at this subtraction amounts 944 kW. At the LP turbine cylinder inlet (point 5), exergy flow stream amounts 18228 kW, what is lower than an exergy flow stream at the HP cylinder outlet (point 3) which amounts 21344 kW for exergy flow stream subtracted between HP and LP turbine cylinders (point 4). The last subtraction at the LP turbine cylinder (point 6) takes an exergy flow stream equal to 502 kW, while at the turbine outlet (at the LP cylinder outlet, point 7) exergy flow stream amounts 1534 kW for a propulsion propeller speed of 83.00 rpm.

Presented change leads to a conclusion that exergy flow streams can vary considerably, even for a small difference in propulsion propeller speed. Likewise, exergy flow streams are directly proportional to analyzed turbine load, they increases at each stream point during the increase in turbine load.
Exergy destruction and exergy efficiency (as well as exergy flow streams) of any steam plant component depend on the ambient pressure and temperature. Real change in the ambient pressure is small and the influence of the ambient pressure change on exergy efficiency or exergy destruction of any steam plant component can be neglected [51]. The ambient temperature change, in practically expected ranges, can have a significant influence on some steam plant components exergy destruction and exergy efficiency.

Some researchers as Ahmadi and Toghraie [28], Ameri et al. [38], Aljundi [52] and Kopac and Hilalci [53] investigate the influence of the ambient temperature on several steam plant components from land-based steam power plants. In each analysis one of the observed components was high power steam turbine and it was investigated the ambient temperature influence on high power steam turbine exergy efficiency and exergy power losses (exergy destruction). All researchers concluded that the ambient temperature change has low impact on high power steam turbines exergy destruction and exergy efficiency. The ambient temperature increase causes an increase in steam turbine exergy destruction and a decrease in steam turbine exergy efficiency. Usually, an increase in the ambient temperature of 10 °C causes a decrease in high power steam turbine exergy efficiency for about 1 % or less.

It can be expected that the influence of the ambient temperature on exergy destruction and exergy efficiency of marine main propulsion steam turbine will also have low impact. Ambient temperature of marine propulsion steam plants usually has a greater influence on exergy destruction and exergy efficiency of plant components when compared to the land-based plants, depending greatly on the geographic area in which ship operates. Likewise, marine steam propulsion plants are more frequently affected by different ambient temperatures than land-based steam plants.

In this analysis, the ambient temperature was varied from 10 °C to 40 °C in the steps of 10 °C, while the ambient pressure remains constant as at the measurement state (1 bar). Selected ambient temperature range can be expected in the wider operating range of the analyzed LNG carrier.

Main propulsion turbine cumulative exergy destruction change during the ambient temperature variation in all three observed propulsion propeller speeds is presented in Fig. 8. As for high power steam turbines from land-based steam power plants, for the analyzed
Vedran Mrzljak, Igor Poljak, Jasna Prpić-Oršić

Exergy analysis of the main propulsion steam turbine from marine propulsion plant

Marine propulsion turbine cumulative exergy destruction increases during the increase in the ambient temperature, at each observed turbine load.

At propulsion propeller speed of 41.78 rpm, turbine cumulative exergy destruction is the lowest and amounts 1938 kW at the ambient temperature of 10 °C, while at the highest observed ambient temperature of 40 °C turbine cumulative exergy destruction is the highest and amounts 2143 kW. An increase in the ambient temperature for 10 °C causes an average increase in turbine cumulative exergy destruction of 68 kW at this propulsion propeller speed.

At 74.59 rpm, turbine cumulative exergy destruction increases from 4851 kW at the ambient temperature of 10 °C to 5363 kW at the ambient temperature of 40 °C. An increase in the ambient temperature for 10 °C causes an average increase in turbine cumulative exergy destruction of 171 kW at this propulsion propeller speed.

Finally, at the highest main propulsion turbine load (83.00 rpm), exergy destruction increases from 5627 kW at the ambient temperature of 10 °C to 6220 kW at the ambient temperature of 40 °C. An increase in the ambient temperature for 10 °C causes an average increase in turbine cumulative exergy destruction of 198 kW at this propulsion propeller speed.

For the analyzed marine propulsion steam turbine is also valid a conclusion that the average increase in cumulative exergy destruction during the ambient temperature increase is bigger and bigger as turbine load increases. Ambient temperature change does not have a significant impact on the analyzed turbine cumulative exergy destruction.

![Fig. 8 Change in main turbine cumulative exergy destruction during the ambient temperature variation at different propulsion propeller speeds](image)

Analyzed main propulsion turbine exergy efficiency change during the change in the ambient temperature is presented in Fig. 9 for each observed propulsion propeller speed. Marine turbine exergy efficiency decreases during the increase in the ambient temperature for each propulsion propeller speed, so the change is the same as for high power steam turbines from land-based steam power plants.

The highest main propulsion turbine exergy efficiency can be observed at the lowest ambient temperature of 10 °C at each turbine load. At the lowest observed ambient temperature of 10 °C analyzed turbine exergy efficiency amounts 67.16 % at 41.78 rpm, 78.99 % at 74.59 rpm and 81.51 % at the highest propulsion propeller speed of 83.00 rpm. At the highest observed ambient temperature of 40 °C marine propulsion turbine exergy efficiency is the lowest for each turbine load and amounts 64.91 % at 41.78 rpm, 77.27 % at
74.59 rpm and 79.95 % at 83.00 rpm. An increase in the ambient temperature for 10 °C causes average decrease in marine propulsion turbine exergy efficiency of 0.75 % at 41.78 rpm, 0.57 % at 74.59 rpm and 0.52 % at 83.00 rpm.

For the analyzed marine steam turbine is valid a conclusion that the average decrease in exergy efficiency during the ambient temperature increase is as lower as turbine load increases. The ambient temperature change does not have significant impact also on the marine propulsion turbine exergy efficiency.

![Fig. 9 Change in main turbine exergy efficiency during the ambient temperature variation at different propulsion propeller speeds](image)

5. Conclusion

This paper presents an exergy analysis of main propulsion steam turbine from LNG carrier steam propulsion plant at three different turbine loads (low load - 41.78 rpm, middle load - 74.59 rpm and high load - 83.00 rpm). Analyzed turbine cumulative exergy destruction and exergy efficiency are directly proportional. The lowest turbine cumulative exergy destruction and exergy efficiency amounts 2041 kW and 66.01 % at propulsion propeller speed of 41.78 rpm, while the highest cumulative exergy destruction and exergy efficiency amounts 5923 kW and 80.72 % at 83.00 rpm. The analyzed marine steam turbine has higher cumulative exergy destruction in regards to developed power and lower exergy efficiency when compared with high power steam turbines from land-based thermal power plants.

Turbine lost power is defined with steam mass flows subtracted from the turbine and represents the additional power which can be developed for propulsion propeller drive if necessary. Analyzed turbine lost power at the highest propulsion propeller speed of 83.00 rpm is the highest and amounts 3339 kW. Steam content at the main propulsion turbine outlet (main condenser inlet) decreases during the increase in propulsion propeller speed. Analyzed turbine exergy flow streams can vary considerably, even for a small difference in propulsion propeller speed. The ambient temperature change does not affect significantly exergy efficiency or destruction of main marine propulsion steam turbine - what is an identical conclusion as for high power steam turbines from land-based steam power plants.

This analysis presented that LNG carrier crew should avoid running at partial main turbine loads during a major period of time - main turbine exergy efficiency significantly decreases at partial loads. Steam subtractions opening and turbine lost power at high loads leads to a possibility of turbine optimization. During the significant ambient temperature increase (for example summer operation in the Persian Gulf), the main propulsion steam
turbine is not the component for which can be expected notable decrease in efficiency or increase in losses.

Acknowledgment

The authors would like to extend their appreciations to the main ship-owner office for conceding measuring equipment and for all help during the exploitation measurements. A retired professor Vladimir Medica, Faculty of Engineering, University of Rijeka is gratefully acknowledged for helpful suggestions and discussions. This work has been fully supported by the Croatian Science Foundation under the project IP-2018-01-3739.

<table>
<thead>
<tr>
<th>NOMENCLATURE</th>
<th>Greek symbols:</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Abbreviations:</strong></td>
<td>( \varepsilon ) specific exergy, kJ/kg</td>
</tr>
<tr>
<td>HFO</td>
<td>( \eta ) efficiency, -</td>
</tr>
<tr>
<td>HP</td>
<td>Subscripts:</td>
</tr>
<tr>
<td>LNG</td>
<td>0 ambient state</td>
</tr>
<tr>
<td>LP</td>
<td>D destruction</td>
</tr>
<tr>
<td></td>
<td>ex exergy</td>
</tr>
<tr>
<td></td>
<td>IN inlet (input)</td>
</tr>
<tr>
<td></td>
<td>OUT outlet (output)</td>
</tr>
<tr>
<td></td>
<td>PL power lost</td>
</tr>
<tr>
<td></td>
<td>Subscripts:</td>
</tr>
<tr>
<td>( \dot{E} )</td>
<td>stream flow power, kJ/s</td>
</tr>
<tr>
<td>( h )</td>
<td>specific enthalpy, kJ/kg</td>
</tr>
<tr>
<td>( m )</td>
<td>mass flow rate, kg/s or kg/h</td>
</tr>
<tr>
<td>( p )</td>
<td>pressure, MPa</td>
</tr>
<tr>
<td>( P )</td>
<td>power, kJ/s</td>
</tr>
<tr>
<td>( \dot{Q} )</td>
<td>heat transfer, kJ/s</td>
</tr>
<tr>
<td>( s )</td>
<td>specific entropy, kJ/kg·K</td>
</tr>
<tr>
<td>( \dot{X}_{\text{heat}} )</td>
<td>heat exergy transfer, kJ/s</td>
</tr>
<tr>
<td>( T )</td>
<td>temperature, °C or K</td>
</tr>
</tbody>
</table>

REFERENCES


Marine Steam Turbine MS40-2 - Instruction Book For Marine Turbine Unit, HYUNDAI-MITSUBISHI, HYUNDAI HEAVY INDUSTRIES CO., LTD., ULSAN, KOREA, 2004., internal ship documentation


https://www.greisinger.de (accessed: 05.01.2018)

https://www.kyma.no (accessed: 05.01.2018)


AN ALTERNATIVE SOLUTION FOR INTERACTION OF FLEXURAL GRAVITY WAVES WITH BOTTOM MOUNTED CIRCULAR CYLINDER

UDC 530.122.1:534.182
Original scientific paper

Summary

The problem of interaction of the flexural gravity waves with the bottom mounted circular cylinder is discussed. Both the linear and second order problems are considered and they are solved by using the eigenfunction expansion principles together with the Boundary Integral Equation technique. The linear solution, which was proposed in the literature by different authors in the past, is explicitly recovered. For the second order problem, a new original solution strategy is proposed and it was shown that, in the case of water waves, a well known semi-analytical solution is also recovered, proving the generality of the proposed approach.

Key words: Flexural gravity waves; Second order; Green's function; Boundary Integral Equations; Eigenfunctions

1. Introduction

Interaction of gravity waves with different obstacles received much attention in the past both in the context of water waves [1, 2, 3] as well as in the context of flexural gravity waves [4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14]. Both the linear [1, 6, 7, 8, 11, 13] and higher order interactions [2, 3, 15, 16] were of concern. The problem of flexural gravity waves is relevant in the context of the ice structure interactions (both for the floating bodies as well as for the cracks which might occur in the ice sheets) and in the context of the elastic floating structures such as floating airports or the elastic plates lying at the water surface. Unlike the problem of water waves where the semi-analytical solution for the vertical circular cylinder, is well mastered and agreed within the community, the solution for the problem of flexural gravity waves is proposed in different forms by the different authors. In the present work an alternative solution is proposed for both the linear and the higher order problems. The potential flow assumptions are adopted and the problem is formulated in frequency domain. The solution methodology relies on the use of the eigenfunction expansion principles either directly or through the definition of the relevant Green’s function and the use of the Boundary Integral Equation (BIE) technique. For the case of linear wave scattering, a very similar work was reported in [13] for Cartesian
geometry and in [6] for circular geometry. The higher order solution is new and was developed based on the same principles as in [3].

2. Mathematical model

The basic configuration together with the different boundary conditions for the generic velocity potential \( \phi \) induced by the cylinder, is shown in Figure 1.

![Image of basic configuration and definitions](image)

Fig. 1 Basic configuration and definitions

The corresponding boundary value problem (BVP) has the following form:

\[
\begin{align*}
\Delta \phi &= 0 & -H \leq z \leq 0 \\
-\alpha \phi + \kappa \frac{\partial \phi}{\partial z} &= Q(r, \theta) & z = 0 \\
\frac{\partial \phi}{\partial r} &= \nu(a, \theta, z) & r = a \\
\frac{\partial \phi}{\partial z} &= 0 & z = -H \\
\phi &\to 0 & r \to \infty
\end{align*}
\]

(1)

where the operator \( \kappa \) is given by the following expression:

\[
\kappa = 1 - \alpha \frac{M}{\rho} + \frac{D}{\rho g} \Delta^2_0
\]

(2)

with \( M \) and \( D \) being the distributed mass and stiffness of the plate, respectively.
It is important to mention that, in the case of the flexural gravity waves, the BVP (1) the additional boundary conditions at the plate ends should be specified. At the cylinder, these conditions describe the way in which the plate is attached to the cylinder i.e. clamped, free, ... For example, in the case of the plate clamped to the fixed cylinder the conditions of the zero displacement and zero slope apply:

$$w(a, \theta) = 0, \quad \frac{\partial w}{\partial r}(a, \theta) = 0$$

(3)

The constant $\alpha$ and the functions $v(z, \theta)$ and $Q(r, \theta)$ are specified for each particular problem and here below we define them in the context of the second order wave body interaction theory.

2.1 Second order theory

The relevant second order theoretical model for monochromatic flexural gravity waves, was presented in [16] and will not be repeated here in details and we just mention few basic principles. We start by assuming that all the quantities of interest (velocity potential, plate deflection, pressure ...) can be developed into a perturbation series with respect to the small parameter $\varepsilon$ chosen to be the wave steepness:

$$\varphi = \varepsilon \varphi^{(1)} + \varepsilon^2 \varphi^{(2)} + o(\varepsilon^2) \quad , \quad w = \varepsilon w^{(1)} + \varepsilon^2 w^{(2)} + o(\varepsilon^2) \quad , \quad \ldots$$

(4)

This series is introduced into the initial fully nonlinear problem and, at the same time, the Taylor series expansion is used in order to express the different quantities at their instantaneous position as a function of their values at rest. After collecting the terms at different order we obtain:

$O(\varepsilon)$

$$\alpha^{(1)} = v, \quad \kappa^{(1)} = 1 - v \frac{M}{\varrho} + \frac{D}{\varrho g} \Delta_0^2, \quad w^{(1)} = -\frac{1}{i \omega} \frac{\partial \varphi^{(1)}}{\partial z}, \quad Q^{(1)} = 0$$

(5)

$O(\varepsilon^2)$

$$\alpha^{(2)} = 4v, \quad \kappa^{(2)} = 1 - 4v \frac{M}{\varrho} + \frac{2D}{\varrho g} \Delta_0^2$$

$$w^{(2)} = -\frac{1}{2i \omega} \left[ \frac{\partial \varphi^{(2)}}{\partial z} - \frac{1}{2} \nabla \varphi^{(1)} \nabla w^{(1)} + \frac{1}{2} w^{(1)} \frac{\partial^2 \varphi^{(1)}}{\partial z^2} \right]$$

(7)

$$Q^{(2)} = \frac{1}{2} \left[ \kappa^{(2)} \left( \nabla \varphi^{(1)} \nabla w^{(1)} - \nabla \varphi^{(1)} \nabla (\kappa^{(1)} w^{(1)}) \right) \right]$$

$$+ \frac{i \omega}{g} \nabla \varphi^{(1)} \nabla \varphi^{(1)} + \frac{\omega^2}{2g} w^{(1)} \frac{\partial \varphi^{(1)}}{\partial z} - \frac{1}{2} \kappa^{(2)} \left( w^{(1)} \frac{\partial^2 \varphi^{(1)}}{\partial z^2} \right)$$

(8)

With respect to the body boundary condition and the quantity $v(a, z, \theta)$ in (1) we should distinguish two types of problems namely: the radiation and the diffraction. In the case of the radiation the fluid is at rest and the normal velocity $v(a, z, \theta)$ is induced by the prescribed body
motion, while in the case of the diffraction the body is fixed and the normal velocity is induced by the incident wave. It is however clear that, in both cases, the resulting BVP is of the same form (1) so that exactly the same method can be used for its resolution.

3. Solution methodology

As already indicated, the solution methodology is based on the use of the eigenfunction expansion principles which are employed either directly or through the definition of the corresponding Green’s function in a way similar to [13]. In that respect we start by employing the method of separating variables which, for the flow symmetric with respect to x axis (θ = 0), allows writing the solution in the following form:

\[ \varphi(r, \theta, z) = \sum_{m=0}^{\infty} \epsilon_m \varphi_m(r, z) \cos m\theta \] (9)

where \( \epsilon_m = 1 \) for \( m = 0 \) and \( \epsilon_m = 2 \) for \( m > 0 \).

It can also be shown (e.g. [13]) that each Fourier mode \( \varphi_m(r, z) \) can be further developed into eigenfunction expansion in vertical direction as follows:

\[ \varphi_m(r, z) = \sum_{n=-\infty}^{\infty} f_n(z) \varphi_{mn}(r) \] (10)

where the functions \( f_n(z) \) denote the eigenfunctions in the vertical direction:

\[ f_n(z) = \frac{\cosh \mu_n(z + H)}{\cosh \mu_n H} \] (11)

The wave numbers \( \mu_n \) are roots of the following dispersion equation:

\[ \alpha = (\mathcal{M} + \mathcal{D} \mu_n^4) \mu_n \tanh \mu_n H \] (12)

where the notation \( \mathcal{M} = 1 - \alpha M / \varrho \) and \( \mathcal{D} = D / \varrho g \) was introduced in order to simplify the writing.

The dispersion equation has two real roots \((\pm \mu_0, \mu_0 > 0)\), infinite number of pure imaginary roots \((\pm \mu_n = \pm ik_n, k_n > 0, n = 1, \infty)\) and four complex roots \((\mu_{-4} = -a_0 - ib_0, \mu_{-3} = a_0 - ib_0, \mu_{-2} = -a_0 + ib_0, \mu_{-1} = a_0 + ib_0 \) with \( a_0 > 0, b_0 > 0)\). In the present analysis we follow the procedure from [8] and we restrict ourselves to the roots \( \mu_{-2}, \mu_{-1}, \mu_0, \mu_n = ik_n, n = 1, \infty \).

It is important to mention that the eigenfunctions \( f_n(z) \) are not orthogonal in a classical sense but they obey the following orthogonal relation:

\[ \int_{-H}^{H} f_m(z) f_n(z) dz + \frac{\mathcal{D}}{\alpha} (f'_m f'_n + f'_m f'_n) = \begin{cases} 0 & n \neq m \\ \frac{1}{2\mathcal{C}_n} & n = m \end{cases} \] (13)

The Green’s function corresponding to the BVP (1), with homogeneous condition at \( z = 0 \), can be derived following the method presented in [11, 13]. Here we skip the details and we simply present the final expression:
An alternative solution for interaction of flexural gravity waves with bottom mounted circular cylinder

\[ G(x; \xi) = \sum_{m=0}^{\infty} \varepsilon_m G_m(r, z; \rho, \zeta) \cos m(\theta - \vartheta) \] (14)

\[ G_m(r, z; \rho, \zeta) = -\frac{i}{2} \sum_{n=2}^{\infty} C_n \left\{ \frac{H_m(\mu_n r)J_m(\mu_n \rho)}{J_m(\mu_n r)H_m(\mu_n \rho)} \right\} f_n(\mu_n z) f_n(\mu_n \zeta), \quad \begin{cases} r > \rho \\ r < \rho \end{cases} \] (15)

This expression is valid for two arbitrary points \( x = (r, \theta, z) \) and \( \xi = (\rho, \vartheta, \zeta) \) in the fluid \(-H \leq (z, \zeta) \leq 0\). For the sake of clarity, the total potential \( \varphi \) is further decomposed in two parts \( \varphi = \varphi_B + \varphi_Q \) by defining the following BVP's:

\[
\begin{align*}
\Delta \varphi_B &= 0 & \Delta \varphi_Q &= 0 & r > a, -H < z < 0 \\
-\alpha \varphi_B + \kappa \frac{\partial \varphi_B}{\partial z} &= 0 & -\alpha \varphi_Q + \kappa \frac{\partial \varphi_Q}{\partial z} &= Q(r, \theta) & z = 0 \\
\frac{\partial \varphi_B}{\partial n} &= v(a, z, \theta) & \frac{\partial \varphi_Q}{\partial n} &= 0 & r = a \\
\frac{\partial \varphi_B}{\partial z} &= 0 & \frac{\partial \varphi_Q}{\partial z} &= 0 & z = -H \\
\varphi_B &\rightarrow 0 & \varphi_Q &\rightarrow 0 & r \rightarrow \infty
\end{align*}
\] (16)

In addition it should be noted that both potentials should also satisfy the conditions at the plate ends (3). The potential \( \varphi_B \) is called body perturbation potential and it represents the generic linear (first order) potential with homogeneous condition at the free surface. The potential \( \varphi_Q \) is called free surface perturbation potential and it satisfy the nonhomogeneous condition at the free surface together with the homogeneous condition at the cylinder. This potential is relevant for higher order problems \([2, 3, 15, 16]\).

3.1 Potential \( \varphi_B \)

The solution for the body potential \( \varphi_B \) will be obtained using two different methods namely: the direct eigenfunctions expansion method and the Boundary Integral Equation method. Both methods are described in details below.

3.1.1 Direct eigenfunctions expansion method

Since the potential \( \varphi_B \) satisfies the homogeneous condition at the free surface, it can be shown that the following eigenfunction expansion can describe any particular wave system generated by the bottom mounted vertical circular cylinder:

\[
\varphi_B(r, \theta, z) = \sum_{m=0}^{\infty} \varepsilon_m \varphi_{bm}(r, z) \cos m\theta = \sum_{m=0}^{\infty} \varepsilon_m \sum_{n=2}^{\infty} f_n(z) \beta_{mn} H_m(\mu_n r) \cos m\theta
\] (17)

where \( \beta_{mn} \) are the unknown coefficients and \( H_m \) denotes the Hankel function of the first kind.
It is important to note that the above expression (17) is written in compact form which means, in particular, that for \( n \geq 1 \) we have:

\[
 f_n(z) = \frac{\cosh i \mu_n(z + H)}{\cosh i \mu_n H} = \frac{\cos k_n(z + H)}{\cos k_n H},
\]

(18)

\[
 K_m(k_n r) = \frac{\pi}{2} i^{m+1} H_m(i \mu_n r), \quad I_m(k_n r) = i^{-m} J_m(i \mu_n r)
\]

(19)

We now apply the body boundary condition (16) to \( \varphi_B \) and we get:

\[
 \sum_{n=-\infty}^{\infty} f_n(z) \beta_{m_n} H_n'(\mu_n a) = v_m(a, z)
\]

(20)

where \( v_m(a, z) \) are the Fourier components of the body velocity.

After multiplying the above equation by \( f_k(z) \) and integrating over the depth, we obtain:

\[
 \sum_{n=-\infty}^{\infty} \beta_{m_n} \mu_k' H_n'(\mu_n a) \int_{-H}^{0} f_k(z) f_n(z) dz = \int_{-H}^{0} f_k(z) v_m(a, z) dz
\]

(21)

By using the orthogonality properties (13) this can be rewritten in the form:

\[
 \sum_{n=-\infty}^{\infty} \beta_{m_n} \mu_k' H_n'(\mu_n a) \left[ \frac{\delta_{kn}}{2\mathcal{C}_n} - \frac{\mathcal{D}}{\alpha} (f_k f_n + f_k f_n)' \right] = \int_{-H}^{0} f_k(z) v_m(a, z) dz
\]

(22)

which is the same as:

\[
 \frac{1}{2\mathcal{C}_k} \beta_{mk} \mu_k' H_n'(\mu_n a) - \frac{\mathcal{D}}{\alpha} \left[ f_k \frac{\partial^2 \varphi_{bm}}{\partial r \partial z} + f_k' \frac{\partial^4 \varphi_{bm}}{\partial r \partial z^3} \right] = \int_{-H}^{0} f_k(z) v_m(a, z) dz
\]

(23)

Knowing that \( f'_k = -\mu_k \tanh \mu_k H \) and \( f''_k = \mu_k^3 \tanh \mu_k H \) we can write the unknown coefficients \( \beta_{mk} \) in the following form:

\[
 \beta_{mk} = \frac{2\mathcal{C}_k}{\mu_k H_n'(\mu_n a)} \left[ \int_{-H}^{0} f_k(z) v_m(a, z) dz + \mu_k \tanh \mu_k H \frac{\mathcal{D}}{\alpha} \left( \mu_k^2 \frac{\partial^2 \varphi_{bm}}{\partial r \partial z} - \frac{\partial^4 \varphi_{bm}}{\partial r \partial z^3} \right) \right]
\]

(24)

The values of \( \frac{\partial^2 \varphi_{bm}}{\partial r \partial z} \) and \( \frac{\partial^4 \varphi_{bm}}{\partial r \partial z^3} \), at the connecting line \( (a, 0) \), are unknown in advance and can be found from the appropriate edge conditions. In order to simplify the notations, we introduce the following notations:

\[
 \gamma_m = \frac{\partial^2 \varphi_{bm}}{\partial r \partial z} (a, 0), \quad \sigma_m = \frac{\partial^4 \varphi_{bm}}{\partial r \partial z^3} (a, 0)
\]

(25)

When applying the edge conditions, and whatever the type of these conditions, the system of equations for \( \gamma_m \) and \( \sigma_m \) can be deduced in the following form:
An alternative solution for interaction of flexural gravity waves with bottom mounted circular cylinder

\[ J_1 \gamma_m + J_2 \sigma_m = J_3 \]  
\[ J_4 \gamma_m + J_5 \sigma_m = J_6 \]  

where \( J_n , n = 1,6 \) are the known coefficients [6].

Solution of this system gives the coefficients \( \gamma_m \) and \( \sigma_m \) so that the problem is formally solved.

### 3.1.2 Boundary Integral Equation method

After applying the Green’s theorem to the unknown velocity potential \( \varphi_b(x) \) and the Green’s function \( G(x;\xi) \), the following Boundary Integral Equation can be written:

\[
\left( 4\pi \varphi_b(x) \right)_0 - \int_{S_a} \varphi_b(\xi) \frac{\partial G(x;\xi)}{\partial \rho} dS = \int_{S_a} G(x;\xi) \frac{\partial \varphi_b(\xi)}{\partial \rho} dS + \int_{S_a} \left[ \varphi_b(\xi) \frac{\partial G(x;\xi)}{\partial \zeta} - G(x;\xi) \frac{\partial \varphi_b(\xi)}{\partial \zeta} \right] dS, \quad \begin{cases} r > a \\ r < a \end{cases}
\]

where \( S_a \) denotes the surface of the cylinder \( r = a \), \( S_f \) denotes the free surface \( z = 0 \), and where it was accounted for the fact that the integrals at infinity and at the sea bottom \( z = -H \) disappear.

For convenience we denote the free surface integral in (28) by \( I_{sr} \) and we rewrite it as follows:

\[
I_{sr} = \int_{S_a} \left[ \varphi_b \frac{\partial G}{\partial \zeta} - G \frac{\partial \varphi_b}{\partial \zeta} \right] dS = \frac{1}{\alpha} \int_{S_a} \left[ \kappa \frac{\partial \varphi_b}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} - \frac{\partial \varphi_b}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} \right] dS = \frac{D}{\alpha} \int_{S_a} \left[ \frac{\partial G}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} - \frac{\partial \varphi_b}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} \right] dS
\]

Now we use the following identity which is valid for two arbitrary harmonic functions \( \phi \) and \( \psi \):

\[
\int_S (\psi \Delta^2 \phi - \phi \Delta^2 \psi) dS = \int_C [\Delta \psi \frac{\partial \phi}{\partial n} - \Delta \phi \frac{\partial \psi}{\partial n} + \psi \frac{\partial \phi}{\partial n} - \phi \frac{\partial \psi}{\partial n} \Delta] dC
\]

where \( S \) denotes the closed surface and \( C \) its contour(s).

This allows rewriting the integral \( I_{sr} \) in the following form:

\[
I_{sr} = \frac{D}{\alpha} \int_{r=a} \left[ \Lambda_0 \frac{\partial G}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} - \Delta_0 \frac{\partial \varphi_b}{\partial \zeta} \frac{\partial G}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} + \frac{\partial G}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} \right] dC
\]

\[
= \frac{D}{\alpha} \int_{r=a} \left[ \frac{\partial G}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} - \frac{\partial \varphi_b}{\partial \zeta} \frac{\partial G}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} + \frac{\partial \varphi_b}{\partial \zeta} \frac{\partial \varphi_b}{\partial \zeta} - \frac{\partial \varphi_b}{\partial \zeta} \right] dC
\]
After exploiting the orthogonality properties of the Fourier series, we can write for one point inside the cylinder \([ r = a - \delta \) \((\delta > 0)\)], the following expression:

\[
-\int_{-H}^{0} \phi_{lm}(a, \zeta) \frac{\partial G_{m}(r, z; a, \zeta)}{\partial \rho} d\zeta = \int_{-H}^{0} G_{m}(r, z; a, \zeta) v_{m}(a, \zeta) d\zeta
\]

\[
-\frac{D}{\alpha} \left[ \frac{\partial^{3} G_{m}}{\partial \zeta^{3}} \frac{\partial^{3} \phi_{lm}}{\partial \rho \partial \zeta^{2}} - \frac{\partial^{3} \phi_{lm}}{\partial \zeta^{3}} \frac{\partial^{3} G_{m}}{\partial \rho \partial \zeta^{2}} + \frac{\partial G_{m}}{\partial \zeta} \frac{\partial^{3} \phi_{lm}}{\partial \rho \partial \zeta^{2}} - \frac{\partial \phi_{lm}}{\partial \rho} \frac{\partial^{4} G_{m}}{\partial \zeta \partial \rho \partial \zeta^{2}} \right]_{\rho=a, \zeta=0}^{\rho=a, \zeta=0}
\]

For the sake of clarity, we rewrite the above equation in compact form:

\[
-\mathcal{L}^{B} = \mathcal{P}^{B} + \frac{D}{\alpha} \left( \mathcal{Q}_{1}^{B} - \mathcal{Q}_{2}^{B} \right)
\]

and we explicit below the different terms:

\[
\mathcal{L}^{B} = \int_{-H}^{0} \phi_{lm}(a, \zeta) \frac{\partial G_{m}(r, z; a, \zeta)}{\partial \rho} d\zeta
\]

\[
= -\sum_{k=2}^{n} \beta_{mk} H_{m}(\mu_{k} a) \frac{i}{2} \mathcal{C}_{k} J_{m}(\mu_{k} r) \mu_{k} H_{m}(\mu_{k} a) f_{k}(\mu_{k} z) \frac{1}{2C_{k}} - \frac{D}{\alpha} \mathcal{Q}_{1}^{B}
\]

\[
\mathcal{P}^{B} = \int_{-H}^{0} G_{m}(r, z; a, \zeta) v_{m}(a, \zeta) d\zeta
\]

\[
= -\sum_{k=2}^{n} \frac{i}{2} \mathcal{C}_{k} J_{m}(\mu_{k} r) H_{m}(\mu_{k} a) f_{k}(\mu_{k} z) \int_{-H}^{0} f_{k}(\mu_{k} \zeta) v_{m}(a, \zeta) d\zeta
\]

\[
\mathcal{Q}_{1}^{B} = \left[ \frac{\partial^{3} \phi_{lm}}{\partial \zeta^{3}} \frac{\partial^{3} G_{m}}{\partial \rho \partial \zeta^{2}} + \frac{\partial G_{m}}{\partial \zeta} \frac{\partial^{3} \phi_{lm}}{\partial \rho \partial \zeta^{2}} - \frac{\partial \phi_{lm}}{\partial \rho} \frac{\partial^{4} G_{m}}{\partial \zeta \partial \rho \partial \zeta^{2}} \right]_{\rho=a, \zeta=0}
\]

\[
= \sum_{n=2}^{\infty} \beta_{mn} H_{m}(\mu_{n} a) \left[ f_{n}^{\mu_{n}} \frac{\partial^{3} G_{m}}{\partial \rho \partial \zeta^{2}} + f_{n}^{\mu_{n}} \frac{\partial^{4} G_{m}}{\partial \zeta \partial \rho \partial \zeta^{2}} \right]_{\rho=a, \zeta=0}
\]

\[
\mathcal{Q}_{2}^{B} = \left[ \frac{\partial^{3} G_{m}}{\partial \zeta^{3}} \frac{\partial^{3} \phi_{lm}}{\partial \rho \partial \zeta^{2}} + \frac{\partial G_{m}}{\partial \zeta} \frac{\partial^{3} \phi_{lm}}{\partial \rho \partial \zeta^{2}} - \frac{\partial \phi_{lm}}{\partial \rho} \frac{\partial^{4} G_{m}}{\partial \zeta \partial \rho \partial \zeta^{2}} \right]_{\rho=a, \zeta=0}
\]

\[
= \sum_{n=2}^{\infty} \frac{i}{2} \mathcal{C}_{n} J_{n}(\mu_{n} r) H_{n}(\mu_{n} a) f_{n}(\mu_{n} z) \left[ f_{n}^{\mu_{n}} \frac{\partial^{3} \phi_{lm}}{\partial \rho \partial \zeta^{2}} + f_{n}^{\mu_{n}} \frac{\partial^{4} \phi_{lm}}{\partial \rho \partial \zeta^{2}} \right]_{\rho=a, \zeta=0}
\]

With this in mind the equation (32) becomes:

\[
\sum_{n=2}^{\infty} \frac{i}{2} \mathcal{C}_{n} J_{n}(\mu_{n} r) H_{n}(\mu_{n} a) f_{n}(\mu_{n} z) \left[ \frac{1}{2C_{k}} \beta_{mk} \mu_{k} H_{m}(\mu_{k} a) - \int_{-H}^{0} f_{k}(\mu_{k} \zeta) v_{m}(a, \zeta) d\zeta - \frac{D}{\alpha} \left( f_{k}^{\mu_{k}} \frac{\partial^{3} \phi_{lm}}{\partial \rho \partial \zeta^{2}} + f_{k}^{\mu_{k}} \frac{\partial^{4} \phi_{lm}}{\partial \rho \partial \zeta^{2}} \right) \right]_{\rho=a, \zeta=0} = 0
\]
Since the above equation should be valid for any $r$ and any $z$ inside the cylinder, the only solution is that the term under the brackets is equal to zero. This leads exactly to the equation (23) which proves that the two solutions are identical.

4. Potential $\varphi_Q$

Due to the fact that the potential $\varphi_Q$ satisfies the non-homogeneous condition at the free surface, it is not possible to derive the explicit dependence of the velocity potential in radial direction. However, at each radial distance the eigenfunction expansions in vertical and circumferential directions can be applied. In that respect we can write for the velocity potential at the cylinder surface ($r = a$):

$$
\varphi_Q(a, \theta, z) = \sum_{m=0}^{\infty} \varepsilon_m \varphi_{qm}(a, z) \cos m\theta = \sum_{m=0}^{\infty} \varepsilon_m \sum_{n=-2}^{n} f_n(z) A_{mn} \cos m\theta
$$

(39)

The solution for this potential will be obtained using the boundary integral equation method only. In this case, the Green's theorem for a point inside the cylinder, gives:

$$
\int_{s_r} \varphi_Q(\xi) \frac{\partial G(x;\xi)}{\partial \rho} dS = I_{s_r}
$$

(40)

where $I_{s_r}$ is now:

$$
I_{s_r} = \int_{s_r} \left[ \varphi_Q \frac{\partial G}{\partial \zeta} - G \frac{\partial \varphi_Q}{\partial \zeta} \right] dS = \frac{1}{\alpha} \int_{s_r} \left[ \left( \kappa \frac{\partial \varphi_Q}{\partial \zeta} \right) \frac{\partial G}{\partial \zeta} - \left( \kappa \frac{\partial G}{\partial \zeta} \right) \frac{\partial \varphi_Q}{\partial \zeta} \right] dS
$$

(41)

$$
= \frac{1}{\alpha} \int_{s_r} \frac{\partial G}{\partial \zeta} Q dS + \frac{D}{\alpha} \int_{s_r} \left[ \frac{\partial G}{\partial \zeta} \Delta_0^2 \frac{\partial \varphi_Q}{\partial \zeta} - \frac{\partial \varphi_Q}{\partial \zeta} \Delta_0^2 \frac{\partial G}{\partial \zeta} \right] dS
$$

After using the orthogonality of the Fourier series, we can write for a point inside the cylinder the following expression:

$$
\int_{-H}^{0} \varphi_{qm}(a, \zeta) \frac{\partial G_m(r, z; a, \zeta)}{\partial \rho} d\zeta = -\frac{1}{\alpha \alpha} \int_{a}^{\infty} \frac{\partial G_m(r, z; \rho, 0)}{\partial \zeta} Q_m(q) d\rho
$$

$$
+ \frac{D}{\alpha} \left[ \Delta_0^2 \frac{\partial G_m}{\partial \rho} \frac{\partial \varphi_{qm}}{\partial \rho} - \frac{\partial \varphi_{qm}}{\partial \rho} \Delta_0^2 \frac{\partial G_m}{\partial \rho} + \frac{\partial G_m}{\partial \rho} \frac{\partial \varphi_{qm}}{\partial \rho} - \frac{\partial \varphi_{qm}}{\partial \rho} \frac{\partial G_m}{\partial \rho} \right]_{\zeta=0}^{\zeta=\infty}
$$

(42)

Similar to body perturbation potential, we rewrite the above expression in compact form as follows:

$$
\mathcal{L}^0 = -\frac{1}{\alpha \alpha} \mathcal{P}^0 + \frac{D}{\alpha} \left( \mathcal{Q}_1^0 - \mathcal{Q}_2^0 \right)
$$

(43)

where the different terms can be deduced in the following form:

$$
\mathcal{L}^0 = \sum_{k=-2}^{\infty} A_{mk} I_m(\mu_k r) \mu_k a H_1(\mu_k a) f_k(\mu_k z) \frac{1}{2\mathcal{C}_k} \frac{D}{\alpha} \mathcal{Q}_1^0
$$

(44)

$$
\mathcal{P}^0 = \int_{a}^{\infty} \frac{\partial G_m(r, z; \rho, 0)}{\partial \zeta} Q_m(q) d\rho
$$

(45)
\[
\sum_{k=-2}^\infty \frac{i}{2} \mathcal{C}_k J_m(\mu_k r) f_k(\mu_k z) \mu_k \tanh \mu_k H \int_\rho^\infty H_m(\mu_k \rho) Q_m(\rho) d\rho
\]  
(46)

\[
Q_1^Q = \sum_{n=-2}^\infty A_{mn} \left[ f_n^\prime \frac{\partial^2 G_m}{\partial \rho \partial \zeta} + f_n \frac{\partial^4 G_m}{\partial \rho \partial \zeta^3} \right]_{\rho = a \ z = 0}
\]  
(47)

\[
Q_2^Q = -\sum_{n=-2}^\infty 2 \mathcal{C}_n J_m(\mu_n r) H_m(\mu_n a) f_n(\mu_n z) \left[ f_n^\prime \frac{\partial^2 \phi_{nm}}{\partial \rho \partial \zeta} + f_n \frac{\partial^4 \phi_{nm}}{\partial \rho \partial \zeta^3} \right]_{\rho = a \ z = 0}
\]  
(48)

After collecting all the different terms we can rewrite the equation (42) as below:

\[
\sum_{k=-2}^\infty \frac{i}{2} \mathcal{C}_k J_m(\mu_k r) f_k(\mu_k z) \cdot \frac{1}{2C_k} A_{mk} \mu_k a H_m'(\mu_k a)
\]

\[
- \frac{\mu_k \tanh \mu_k H}{\alpha} \int_\rho^\infty H_m(\mu_k \rho) Q_m(\rho) d\rho - \frac{a H_m(\mu_k a)}{\alpha} \mathcal{D} \left( f_k^\prime \frac{\partial^2 \phi_{nm}}{\partial \rho \partial \zeta} + f_k \frac{\partial^4 \phi_{nm}}{\partial \rho \partial \zeta^3} \right)_{\rho = a \ z = 0} = 0
\]  
(49)

Since this equation should be valid for any \( r \) and any \( z \) inside the cylinder, the only solution is that the term under the brackets is equal to zero. Furthermore, knowing that \( f_k^\prime = -\mu_k \tanh \mu_k H \) and \( f_k = \mu_k' \tanh \mu_k H \) we can write the unknown coefficients \( A_{mk} \) in the following form:

\[
A_{mk} = \frac{-2C_k \mu_k \tanh \mu_k H}{\mu_k a H_m'(\mu_k a)} \cdot \frac{1}{\alpha} \left[ \int_\rho^\infty H_m(\mu_k \rho) Q_m(\rho) d\rho - a \mathcal{D} H_m(\mu_k a) \left( \mu_k^2 \frac{\partial^2 \phi_{nm}}{\partial r \partial \zeta} - \frac{\partial^4 \phi_{nm}}{\partial r \partial \zeta^3} \right)_{r = a \ z = 0} \right]
\]  
(50)

The quantities \( \frac{\partial^2 \phi_{nm}}{\partial r \partial \zeta} \) and \( \frac{\partial^4 \phi_{nm}}{\partial r \partial \zeta^3} \), are evaluated in the same way as in the first order case i.e. through the application of the edge conditions.

5. Some particular cases

The above described procedure is now applied to some particular wave body interaction problems. In order to demonstrate the generality of the proposed approaches, we first apply the methods to the case of the water wave interactions and we show that the classical results are recovered. After that the case of the linear wave diffraction is considered and it is shown that the classical solution from [1] is recovered at the end.

5.1 Water waves

In order to demonstrate the generality of the proposed approaches, here we apply the methods to the case of the water wave interactions and we show that the classical results are easily obtained by simply applying the expressions given here.
An alternative solution for interaction of flexural gravity waves with bottom mounted circular cylinder

Sime Malenica

In the case of the water waves we have $M = 0$ and $D = 0$ and, at the same time, the edge conditions \( (3) \) are no more relevant. With this in mind, the classical dispersion relation for water waves is easily recovered:

$$\alpha = \mu \tanh \mu H$$  \hspace{1cm} (51)

The solution of this equation gives one positive root \( (\mu_0) \) and an infinite number of purely imaginary roots \( (i\mu_n, n = 1, \infty) \).

The corresponding eigenfunctions in vertical direction keep the same form \( (11) \) and the orthogonality relation simplifies to:

$$\int_{-H}^{0} f_m(z)f_n(z)dz = \begin{cases} 0 & n \neq m \\ \frac{2C_n}{2C_n} & n = m \end{cases}$$  \hspace{1cm} (52)

The unknown coefficients for the body perturbation potential become:

$$\beta_{mn} = -\frac{2C_n}{\mu_n H_m(\mu_n \alpha)} \int_{-H}^{0} f_n(z)v_m(a,z)dz$$  \hspace{1cm} (53)

and those for the free surface perturbation potential become:

$$A_{mn} = -\frac{2C_n}{\mu_n \alpha H_m(\mu_n \alpha)} \int_{-H}^{0} H_m(\mu_n \alpha)Q_m(\theta)\rho d\rho$$  \hspace{1cm} (54)

These expressions are the same as those in [3].

5.2 Diffraction

Only the case of the linear diffraction of monochromatic waves is considered.

5.2.1 Flexural gravity waves

The incident wave field is defined as the progressive sinusoidal wave with the following expression for the corresponding velocity potential \( \varphi_I \):

$$\varphi_I = A\varphi(z) \sum_{m=0}^{\infty} \epsilon_m^m i^m J_m(\mu_l r) \cos m\theta$$  \hspace{1cm} (55)

where $A = \frac{-ig\zeta_A}{\omega}$ and $\zeta_A$ is the wave amplitude.

The body perturbation potential is given by \( (17) \) with the unknown coefficients $\beta_{mk}$ given by \( (53) \).

Knowing that:

$$\int_{-H}^{0} f_k(z)f_0(z)dz = \frac{\delta_{k0}}{2C_0} + \frac{D}{\alpha} (f_k^* f_0 + f_k f_0^*)_{z=0}$$  \hspace{1cm} (56)

we can easily deduce the following expression for $\beta_{mk}$:
\[
\beta_{mk} = -A_i^m \frac{J_m'(\mu_0 a)}{H_m'(\mu_0 a)} \delta_{k0} \\
+ \frac{2C_i \tanh \mu_0 H D}{H_m'(\mu_0 a)} \alpha \left[ A_i^m \mu_0 J_m'(\mu_0 a) \tanh \mu_0 H \left( \mu_k^2 + \mu_0^2 \right) + \left( \mu_k^2 \gamma_m - \sigma_m \right) \right]
\]

or:

\[
\beta_{m0} = -A_i^m \frac{J_m'(\mu_0 a)}{H_m'(\mu_0 a)} + \frac{2C_i \tanh \mu_0 H D}{H_m'(\mu_0 a)} \alpha \left[ 2A_i^m \mu_0 J_m'(\mu_0 a) \tanh \mu_0 H + \left( \mu_k^2 \gamma_m - \sigma_m \right) \right] \\
\beta_{mk} = \frac{2C_i \tanh \mu_0 H D}{H_m'(\mu_0 a)} \alpha \left[ A_i^m \mu_0 J_m'(\mu_0 a) \tanh \mu_0 H \left( \mu_k^2 + \mu_0^2 \right) + \left( \mu_k^2 \gamma_m - \sigma_m \right) \right]
\]

where \( k \neq 0 \).

It can be shown that this solution is identical to the solution proposed in [8].

5.2.2 Water waves

In the case of water waves the well known MacCamy and Fuchs solution is recovered:

\[
\beta_{m0} = -A_i^m \frac{J_m'(\mu_0 a)}{H_m'(\mu_0 a)}
\]

with \( \beta_{mk} = 0 \) for \( k > 0 \).

6. Conclusions

We presented here an alternative semi-analytical method for the solution of the interactions in between the flexural gravity waves and the bottom mounted vertical circular cylinder. The proposed method covers both the linear (body perturbation potential) and the higher order solution (free surface perturbation potential). The linear solution was obtained by two different methods namely direct eigenfunction expansion method and the boundary integral equation method, while the higher order solution was found using the boundary integral equation method only. For the linear case, it was shown that the final expressions are equivalent to other expressions proposed in the literature. The higher order solution for flexural gravity waves is new and was not reported before in the literature.

7. Acknowledgments

The support from a National Research Foundation of Korea (NRF) grant funded by the Korean Government (MSIP) through GCRC-SOP (Grant No. 2011-0030013) is acknowledged. Discussions and advices of Prof. Korobkin A.A. and Prof. Khabakhpasheva T. are also greatly appreciated.

REFERENCES


An alternative solution for interaction of flexural gravity waves with bottom mounted circular cylinder


Submitted: 07.12.2018. Šime Malenica, sime.malenica@bureauveritas.com
Bureau Veritas Marine & Offshore

Accepted: 15.01.2019. Paris, FRANCE
A FUZZY EVALUATION MODEL OF CHOOSING A MIDDLE MANAGER FOR AN INTERNATIONAL SHIPPING SERVICE PROVIDER

UDC 323.3-057.17
Professional paper

Summary

Choosing a middle manager with management competency and capabilities will have a decisive influence on the organization's development for international shipping service providers. There is ambiguity and uncertainty in the decision-making environment during the selection of a middle manager and many evaluation criteria must be considered. The main purpose of this article is to construct a fuzzy multiple criteria decision-making (MCDM) model for international shipping service providers to use when selecting a middle manager. First, some methods and concepts of the fuzzy theory are introduced in this article. Five steps of evaluation model of fuzzy MCDM algorithms are then proposed to choose a best middle manager. Finally, an international shipping case is presented and the proposed fuzzy MCDM model is illustrated step by step. It can be seen from the demonstration that this evaluation model can be used to effectively select the best middle manager.

Key words: fuzzy evaluation model; multiple criteria decision-making (MCDM); middle manager; international shipping service provider

1. Introduction

The rise of international trade and the thriving marine transportation sector not only have accelerated the promotion and integration of the world economy and culture, but also created a vast market for the shipping industry [1]. International shipping is facing a competitive environment based on the four RCs, namely, rising complexity, rapid change, radical challenges, and rising competition. The competition among shipping companies is becoming more and more intense [2]. However, the services of shipping companies are no longer limited to port-to-port services. The emphasis is now on door-to-door services extending from seaside to landside [3], which has given birth to competition issues in business logistics and shipping markets.

With the fast development of international economic activities in recent years, the various operations of international trade are required to become increasingly economic and efficient. The integrated operations of international shipping logistics systems [4] could be
regarded as an opportunity to strengthen international marketing. In addition, the growth of regional economies, the evolution of supply chain concepts, the progress of e-commerce, and the lifting of international financial and transportation controls have all contributed to the vigorous development of international logistics [1, 4]. In other words, with the development of international trade, many multinational companies have begun to use their companies as part of the layout of an international strategy that expects to achieve maximum profits through the international division of labour and production. Therefore, international shipping logistics has taken a pivotal role in global trade.

Due to increasingly fierce competition as well as diversified and rapid changes in international trade and shipping markets, how to provide customers with more comprehensive integrated logistics services is an important issue for international shipping service providers [4, 5]. International shipping service providers play an important role of third-party logistics providers (3PLs) and are also important logistics supporters in the international transport industry [5]. Therefore, in order to provide better shipping logistics services and enhance companies operational performance so as to expand their scale, companies must continuously recruit excellent personnel through internal and external efforts, which can help form an effective work team and develop more effective organizations [6]. The shipping service industry requires employees who can create organizational business value externally. Therefore, firms must be able to see market opportunities and maintain close interaction with their customers so as to establish a good external network. Internally, they must be able to identify and integrate talents with relevant functions in order to grasp market opportunities.

Having excellent manpower quality is a key factor of a company's success [7]. Enterprise managers [6] can be divided into first-line, middle-level, and high-level managers, which have different responsibilities at different levels. For example, first-line managers supervise the work of non-management employees on a day-to-day basis and are among the front-line managers who perform tasks. Middle-level managers are responsible for overseeing first-line managers and are responsible for finding the best ways to align human resources and other resources to accomplish organizational goals. High-level managers are responsible for setting organizational goals, determining how different departments interact with each other, and supervising the performance of middle-level managers. Among the three types of managers mentioned above, middle-level managers act as a link between high level and first-line management. Moreover, they act as important information transmitters between the organizational operations department and the decision-making departments in task assignment as well as policy communication and organizational execution [6, 8, 9]. Therefore, choosing a middle-level manager with management competency and capabilities [10] will have a decisive influence on an organization's development.

The selection of a middle manager is important for organizational development. However, it is not easy to choose a good middle manager, because the human resources (HR) department is constantly faced with the uncertainty of the environment when choosing a middle manager and many evaluation criteria must be considered. Therefore, the selection process of a middle manager is full of the characteristics of multiple criteria decision-making (MCDM) [10]. In addition, with the changes of group decision-making and the environment, the weight value of each criterion and its importance also consists of fuzzy and changing characteristics [11]. Due to the ambiguity of traditional decision-making methods in dealing with the criteria weight and the inaccuracy of the transmission of decision-making information, it is unable to adequately express the information implied by various evaluation plans and decision-making criteria. In order to properly integrate the opinions of the decision-making groups (or committees) formed by the relevant decision-making units, and to evaluate and rank the alternatives for the best solution, this study intends to apply the fuzzy set theory [12] and incorporate MCDM to establish a selection model for a middle manager to allow
international shipping service providers to find the best middle managers in a fuzzy environment.

The fuzzy MCDM method is valued by scholars in the decision making field and industrial circles and has been widely applied in numerous fields. In recent years, in terms of personnel selection, numerous studies [13-18] have adopted this evaluation method and it has received great attention from many human resource managers and scholars. Thus, this article will use the operation and concept of fuzzy MCDM to facilitate the evaluation of middle managers by applying this method. To sum up, the main purpose of this article is to construct a fuzzy evaluation model to facilitate the HR department of international shipping service providers to select the best middle managers. The following section introduces research methodology, and a fuzzy MCDM evaluation model is proposed in Section 3. The fourth section illustrates a numerical example, and the conclusion is presented in Section 5.

2. Methodology

2.1 The trapezoidal fuzzy numbers and their algebraic operations

In a universe of discourse $X$, a fuzzy subset $\tilde{A}$ of $X$ is defined by a membership function $\mu_{\tilde{A}}(x)$, which maps each element $x$ in $X$ to a real number in the interval $[0, 1]$. The function value $\mu_{\tilde{A}}(x)$ represents the grade of membership of $x$ in $\tilde{A}$.

A fuzzy number $\tilde{A}$ [19] in real line $\mathcal{R}$ is a trapezoidal fuzzy number if its membership function $\mu_{\tilde{A}} : \mathcal{R} \rightarrow [0, 1]$ is

$$\mu_{\tilde{A}}(x) = \begin{cases} (x-t^e)(t^a-t^e), & t^e \leq x \leq t^a \\ 1, & t^a \leq x \leq t^b \\ (x-t^a)(t^b-t^a), & t^b \leq x \leq t^d \\ 0, & \text{otherwise} \end{cases} (1)$$

with $-\infty < t^e \leq t^a \leq t^b \leq t^d < \infty$. The trapezoidal fuzzy number can be denoted by $(t^e, t^a, t^b, t^d)$.

According to extension principle [12], let $\tilde{A}_1 = (t^{e_1}, t^{a_1}, t^{b_1}, t^{d_1})$ and $\tilde{A}_2 = (t^{e_2}, t^{a_2}, t^{b_2}, t^{d_2})$ be trapezoidal fuzzy numbers, the algebraic operations of any two trapezoidal fuzzy numbers $\tilde{A}_1$ and $\tilde{A}_2$ can be expressed as

1. Fuzzy addition, $\oplus$:
   $$\tilde{A}_1 \oplus \tilde{A}_2 = (t^{e_1}+t^{e_2}, t^{a_1}+t^{a_2}, t^{b_1}+t^{b_2}, t^{d_1}+t^{d_2})$$

2. Fuzzy subtraction, $\ominus$:
   $$\tilde{A}_1 \ominus \tilde{A}_2 = (t^{e_1}-t^{e_2}, t^{a_1}-t^{a_2}, t^{b_1}-t^{b_2}, t^{d_1}-t^{d_2})$$

3. Fuzzy multiplication, $\odot$:
   $$k \odot \tilde{A}_2 = (kt^{e_2}, kt^{a_2}, kt^{b_2}, kt^{d_2}), \; k \in \mathcal{R}, \; k \geq 0; \quad \tilde{A}_1 \odot \tilde{A}_2 \equiv (t^{e_1}t^{e_2}, t^{a_1}t^{a_2}, t^{b_1}t^{b_2}, t^{d_1}t^{d_2}), \; t^{e_1} \geq 0, \; t^{e_2} \geq 0;$$

4. Fuzzy division, $\oslash$:
   $$(\tilde{A}_1)^{-1} = (t^{e_1}, t^{a_1}, t^{b_1}, t^{d_1})^{-1} \equiv (1/t^{d_1}, 1/t^{b_1}, 1/t^{a_1}, 1/t^{e_1}), \; t^{e_1} > 0;$$
\[ \tilde{A}_i \ominus \tilde{A}_j \cong (t^{i_1}/t^{d_1}, t^{i_2}/t^{d_2}, t^{i_3}/t^{d_3}), \quad t^{i_1} \geq 0, t^{d_1} > 0. \]

2.2 Linguistic variables

Zadeh [20] proposed the concept of linguistic variables, which is used to deal with problems that are too complex or too difficult to be properly described by traditional quantitative methods. Linguistic variables can provide a convenient quantitative syntax for complex or poorly defined descriptions. A linguistic variable is a variable expressed in words or natural sentences. For example, “importance” is a linguistic variable in which its value is spoken language rather than numerical values. The approximate reasoning of the fuzzy set theory can be used to reasonably express the linguistic values of “very unimportant,” “unimportant,” “normal,” “important,” and “very important.” In this article, the trapezoidal fuzzy number is used to convey the linguistic value of importance and superiority evaluation. For example, the linguistic value set of importance is \( W = \{ VL, L, M, H, VH \} \), while the linguistic value set of the superiority evaluation is \( S = \{ VP, P, F, G, VG \} \). The membership function for the linguistic values contained in the set \( W \) and \( S \) can be defined as follows: very low \((VL) = \) very poor \((VP) = (0, 0, 0.2, 0.3)\); low \((L) = \) poor \((P) = (0.2, 0.3, 0.4, 0.5)\); medium \((M) = \) fair \((F) = (0.4, 0.5, 0.6, 0.7)\); high \((H) = \) good \((G) = (0.6, 0.7, 0.8, 0.9)\); very high \((VH) = \) very good \((VG) = (0.8, 0.9, 1, 1)\). These trapezoidal fuzzy numbers can be referred to in Ghyym [21].

2.3 Ranking fuzzy numbers with the maximizing and minimizing sets

The ranking method developed by Chen [22], Kim and Park [23] and Chang and Chen [24] is adopted in this article, because it is easy and powerful.

Let \( \tilde{A}_i, i = 1, 2, \ldots, n \), be the fuzzy numbers. The membership functions can be denoted by \( \mu_{\tilde{A}_i} \). We define the maximizing set \( \tilde{G} = \{(x, \mu_G(x)) \mid x \in R\} \) with

\[
\mu_G(x) = \begin{cases} 
(x-x_i)/(x_r-x_i), & x \in [x_l, x_r], \\
0, & otherwise,
\end{cases}
\]

and the minimizing set \( \tilde{L} = \{(x, \mu_L(x)) \mid x \in R\} \) with

\[
\mu_L(x) = \begin{cases} 
(x-x_i)/(x_r-x_i), & x \in [x_l, x_r], \\
0, & otherwise,
\end{cases}
\]

where \( x_l = \inf D_i, x_r = \sup D_i, D = \bigcup_{i=1}^n D_i \) and \( D_i = \left\{ x \mid \mu_{\tilde{A}_i}(x) > 0 \right\} \).

Then, the value of optimistic ranking (we can call the optimistic utility) \( U^O_{\tilde{G}}(\tilde{A}_i) \) and the value of pessimistic ranking (we can call the pessimistic utility) \( U^P_{\tilde{L}}(\tilde{A}_i) \) of the fuzzy numbers \( \tilde{A}_i \) can be denoted by

\[
U^O_{\tilde{G}}(\tilde{A}_i) = \sup_x \left( \mu_{\tilde{A}_i}(x) \land \mu_G(x) \right) \quad (2)
\]

and

\[
U^P_{\tilde{L}}(\tilde{A}_i) = \sup_x \left( \mu_{\tilde{A}_i}(x) \land \mu_L(x) \right) \quad (3)
\]

where \( \land \) means the minimum operation and \( i = 1, 2, \ldots, n \).
Finally, we define the ranking value $U^R_i(\tilde{A}_i)$ of fuzzy numbers $A_i$ as

$$U^R_i(\tilde{A}_i) = \delta U^0_i(\tilde{A}_i) + (1 - \delta)U^L_i(\tilde{A}_i), \quad 0 \leq \delta \leq 1.$$  \hspace{1cm} (4)

The value of $\delta$ in the above formula is expressed as the total risk attitude index (TRAI) of the decision-makers (DMs), and TRAI reflects the risk-taking degree of the DMs. When $\delta < 0.5$, the total risk attitude of the DMs is pessimistic, which indicates that the DMs are risk-avers. When $\delta = 0.5$, the total risk attitude of the DMs is moderate, which indicates that the DMs are risk-neutral. When $\delta > 0.5$, the total risk attitude of the DMs is optimistic, which indicates that the DMs are risk-lovers.

The TRAI of DMs is an important issue in group decision-making. Ghyym [21] compared the measurement methods of the risk attitude of DMs. In general, the value of $\delta$ can be determined in two ways. The first method is to determine the value of $\delta$ by a single DM in the data output stage [23] according to the risk-taking degree of his/her final subjective cognition. For example, the value of $\delta$ can be 0.24, 0.5 or 0.76, etc. However, this approach is difficult to apply to problems in multi-person decision-making groups. Therefore, Chang and Chen [24] proposed another way of thinking. They believed that it is more reasonable for the value of $\delta$ to directly transmit the risk-taking degree of the group DMs at the data input stage. In this article, the method of Chang and Chen [24] is considered more reasonable after comprehensive consideration. Therefore, the method used to determine the value of $\delta$ developed by Chang and Chen [24] is the basis for evaluating the TRAI of the DMs or decision-making groups.

Furthermore, the ranking of the trapezoidal fuzzy numbers $\tilde{A}_i$ and $\tilde{A}_j$ are defined and based on the following rules:

1. $\tilde{A}_i > \tilde{A}_j \iff U^R_i(\tilde{A}_i) > U^R_j(\tilde{A}_j)$;
2. $\tilde{A}_i < \tilde{A}_j \iff U^R_i(\tilde{A}_i) < U^R_j(\tilde{A}_j)$;
3. $\tilde{A}_i = \tilde{A}_j \iff U^R_i(\tilde{A}_i) = U^R_j(\tilde{A}_j)$.

Let $\tilde{A}_i = (t^a_i, t^b_i, t^c_i, t^d_i), i = 1, 2, \ldots, n$, be $n$ trapezoidal fuzzy numbers. By employed the Eq. (1), (2), (3) and (4), the ranking value $U^R_i(\tilde{A}_i)$ of the trapezoidal fuzzy number $A_i$ can be denoted by

$$U^R_i(\tilde{A}_i) = \delta \left[ \frac{t^d_i - x_i}{x_r - x_i - t^b_i + t^d_i} \right] + (1 - \delta) \left[ 1 - \frac{x_r - t^c_i}{x_r - t^c_i + t^b_i - t^d_i} \right],$$  \hspace{1cm} (5)

where $x_i = \min \{ t^a_i, t^c_i, \ldots, t^x_i \}$, $x_r = \max \{ t^d_i, t^{x_i}, \ldots, t^{d_i} \}$ and $0 \leq \delta \leq 1$.

After obtaining the value of $\delta$, the ranking value of Eq. (5) can be calculated, and according to the above ranking rules, the priority of the $n$ trapezoidal fuzzy numbers can be determined.

3. The fuzzy evaluation model

A complete evaluation model of MCDM should include goals or objectives, alternatives, criteria or attributes, performance values, and the DMs’ preferences, etc. Based on this, the operational steps of the fuzzy evaluation model for an international shipping service provider
to select a middle manager in this article mainly includes five steps, which are explained as follows.

Step 1: Create a hierarchical structure.
Step 2: Obtain fuzzy weights for all criteria and sub-criteria.
Step 3: Evaluate the fuzzy ratings of all alternatives against all evaluation criteria.
Step 4: Evaluate comprehensive evaluation values for all alternatives.
Step 5: Select the best solution.

In order to allow readers to clearly understand the operational process of the model in this article, the flow chart of the fuzzy evaluation model is shown in Fig. 1.

3.1 Create a hierarchical structure

Because the hierarchical structure is the skeleton structure of the system, the primary task of the MCDM is to establish a structure of decision-making issues so that the relationships between different hierarchical structures can be systematically understood. The hierarchical structure of this article is shown in Fig. 2. In this architecture, tier 1 is the goal, and it is expected that the best one can be selected among the middle managers under evaluation; tier 2 is the $k$ main criteria for the selection; tier 3 is the $n_1 + \cdots + n_p + \cdots + n_k$ sub-criteria for all main criteria; and tier 4 are the $m$ alternatives.
In terms of the selection criteria and sub-criteria, because the factors influencing the selection of middle managers involve the skills, capabilities and competency of managers, based on the related management academic literature [4-10, 25-28] and the opinions of the heads of the HR departments in international shipping companies and management scholars, five assessment aspects and 20 assessment criteria are summed up in this article (their codes are marked in parentheses). The criteria and sub-criteria in this article are all subjective criteria.

(1) Leadership Competency (C1). This assessment aspect includes four management capabilities, including ‘the capability to effectively build team spirit and work atmosphere (C11),’ ‘the capability to positively motivate subordinates (C12),’ ‘the capability to influence subordinates to support the team (C13),’ and ‘the capability to impartially and objectively evaluate the performance of subordinates (C14).’

(2) Interpersonal Competency (C2). This assessment aspect includes four management capabilities, including ‘the capability to integrate and coordinate (C21),’ ‘the capability to lead team awareness (C22),’ ‘the capability to communicate in spoken language (C23),’ and ‘the capability to manage interpersonal networks perfectly (C24).’

(3) Administrative Competency (C3). This assessment aspect includes four management capabilities, including ‘the capability to effectively interpret relevant administrative information (C31),’ ‘the capability to manage crisis (C32),’ ‘the capability to transform conceptual schemes into executable strategic plans (C33),’ and ‘the capability to effectively manage and allocate available resources (C34).’

(4) Professional Competency (C4). This assessment aspect includes four management capabilities, including ‘the capability to thoroughly understand the work procedures of logistics and related practices (C41),’ ‘the capability to manage work pressure (C42),’ ‘the capability to use logistics expertise to enhance work efficiency (C43),’ and ‘the capability to have cross-divisional work experience (C44).’

(5) Conceptual Competency (C5). This assessment aspect includes four management capabilities, including ‘the capability to simplify complex issues (C51),’ ‘the capability to integrate resources within and outside related organizations (C52),’ ‘the capability to plan and organize (C53),’ and ‘the capability to properly understand the internal and external competitive environment (C54).’

3.2 Obtain fuzzy weights for all criteria and sub-criteria

The concept of linguistic variables in Section 2.2 is used to assist in the calculation of fuzzy weights. In this article, the fuzzy weights of all criteria and sub-criteria are obtained by using the arithmetic mean.

Let \( \tilde{w}_{ph} = (t^{p_1h}, t^{p_2h}, t^{p_3h}, t^{p_4h}) \), \( 0 \leq t^{p_1h} \leq t^{p_2h} \leq t^{p_3h} \leq t^{p_4h} \leq 1 \), \( p = 1, 2, \ldots, k; \) and \( h = 1, 2, \ldots, n \), represent the fuzzy weight given to the evaluation criterion \( C_p \) by decision maker \( D_h \). According to Zadeh’s extension principle [12], the average fuzzy weight of the evaluation criterion \( C_p \) can then be represented by

\[
\tilde{W}_p = \frac{1}{n} \otimes (\tilde{w}_{p1} \oplus \tilde{w}_{p2} \oplus \cdots \oplus \tilde{w}_{pn}) = (t^{p_1}, t^{p_2}, t^{p_3}, t^{p_4}),
\]

where \( t^{p_1} = \frac{1}{n} \sum_{h=1}^{n} t^{p_1h} \), \( t^{p_2} = \frac{1}{n} \sum_{h=1}^{n} t^{p_2h} \), \( t^{p_3} = \frac{1}{n} \sum_{h=1}^{n} t^{p_3h} \), \( t^{p_4} = \frac{1}{n} \sum_{h=1}^{n} t^{p_4h} \).
Let $\tilde{w}_{pjh} = (t^{e_p}, t^{a_p}, t^{b_p}, t^{d_p}), \quad 0 \leq t^{e_p} \leq t^{a_p} \leq t^{b_p} \leq t^{d_p} \leq 1, \quad p = 1, 2, \ldots, k; $  
$j = 1, 2, \ldots, n_p; \quad h = 1, 2, \ldots, n,$ represent the fuzzy weight given to the evaluation sub-criterion $C_{pj}$ by decision maker $D_h$. According to Zadeh’s extension principle [12], the average fuzzy weight of evaluation sub-criterion $C_{pj}$ can then be represented by

$$\tilde{W}_{pj} = \frac{1}{n} \bigotimes_{h=1}^{n} (\tilde{w}_{pjh} \oplus \tilde{W}_{pj2} \oplus \cdots \oplus \tilde{W}_{pjn}) = (t^{e_{pj}}, t^{a_{pj}}, t^{b_{pj}}, t^{d_{pj}}),$$

(7)

where $t^{e_{pj}} = \frac{1}{n} \sum_{h=1}^{n} t^{e_p}, \quad t^{a_{pj}} = \frac{1}{n} \sum_{h=1}^{n} t^{a_p}, \quad t^{b_{pj}} = \frac{1}{n} \sum_{h=1}^{n} t^{b_p}, \quad t^{d_{pj}} = \frac{1}{n} \sum_{h=1}^{n} t^{d_p}$.

3.3 Evaluate the fuzzy ratings of all alternatives against all evaluation criteria.

In terms of the evaluation of the performance value, the arithmetic mean and the concept of linguistic variables are still used to calculate the appropriateness rating in this study.

Let $\tilde{s}_{ipj} = (t^{e_i}, t^{a_i}, t^{b_i}, t^{d_i}), \quad 0 \leq t^{e_i} \leq t^{a_i} \leq t^{b_i} \leq t^{d_i} \leq 1, \quad i = 1, 2, \ldots, m; \quad p = 1, 2, \ldots, k; \quad j = 1, 2, \ldots, n_p; \quad h = 1, 2, \ldots, n,$ represent the appropriateness rating assigned to alternative $A_i$ by decision maker $D_h$ for the evaluation sub-criterion $C_{pj}$. According to Zadeh’s extension principle [12], the average fuzzy appropriateness rating of alternative $A_i$ can then be represented by

$$\tilde{S}_{ipj} = \frac{1}{n} \bigotimes_{h=1}^{n} (\tilde{s}_{ipjh} \oplus \tilde{S}_{ipj2} \oplus \cdots \oplus \tilde{S}_{ipjn}) = (t^{e_{ipj}}, t^{a_{ipj}}, t^{b_{ipj}}, t^{d_{ipj}}),$$

(8)

where $t^{e_{ipj}} = \frac{1}{n} \sum_{h=1}^{n} t^{e_i}, \quad t^{a_{ipj}} = \frac{1}{n} \sum_{h=1}^{n} t^{a_i}, \quad t^{b_{ipj}} = \frac{1}{n} \sum_{h=1}^{n} t^{b_i}, \quad t^{d_{ipj}} = \frac{1}{n} \sum_{h=1}^{n} t^{d_i}$.

3.4 Evaluate comprehensive evaluation values for all alternatives.

Let $\tilde{W}_p = (t^{e_p}, t^{a_p}, t^{b_p}, t^{d_p}), \quad p = 1, 2, \ldots, k,$ be the average fuzzy weight of the evaluation criterion $C_p$. Let $\tilde{W}_{pj} = (t^{e_p}, t^{a_p}, t^{b_p}, t^{d_p}), \quad p = 1, 2, \ldots, k; \quad j = 1, 2, \ldots, n_p,$ be the average fuzzy weight of the evaluation sub-criterion $C_{pj}$. Let $\tilde{S}_{ipj} = (t^{e_i}, t^{a_i}, t^{b_i}, t^{d_i}), \quad i = 1, 2, \ldots, m; \quad p = 1, 2, \ldots, k; \quad j = 1, 2, \ldots, n_p,$ be the average fuzzy appropriateness rating of alternative $A_i$. Then, the aggregation appropriateness ratings of the $n_p$ evaluation sub-criteria under the $p^{th}$ evaluation criterion for the $i^{th}$ alternative, can then be represented by $\tilde{R}_{ip}$. That is:

$$\tilde{R}_{ip} = \frac{1}{n_p} \bigotimes \left[ (\tilde{S}_{ip1} \otimes \tilde{W}_{p1}) \oplus (\tilde{S}_{ip2} \otimes \tilde{W}_{p2}) \oplus \cdots \oplus (\tilde{S}_{ipn} \otimes \tilde{W}_{pn}) \right]$$

$$p = 1, 2, \ldots, k; \quad i = 1, 2, \ldots, m.$$

(9)

According to the extension principle, we can represent $\tilde{R}_{ip}$ as $\tilde{R}_{ip} \simeq (t^{E_i}, t^{Q_i}, t^{G_i}, t^{Z_i})$, where $t^{E_i} = \frac{\sum_{p=1}^{n_p} t^{E_p} t^{a_p}}{n_p}, \quad t^{Q_i} = \frac{\sum_{p=1}^{n_p} t^{Q_p} t^{a_p}}{n_p}, \quad t^{G_i} = \frac{\sum_{p=1}^{n_p} t^{G_p} t^{a_p}}{n_p}, \quad t^{Z_i} = \frac{\sum_{p=1}^{n_p} t^{Z_p} t^{a_p}}{n_p},$ for $i = 1, 2, \ldots, m; \quad p = 1, 2, \ldots, k.$
Similarly, the aggregation appropriateness ratings of the \( k \) evaluation criteria for the \( i^{th} \) alternative can be represented by \( \widetilde{F}_i \). That is:

\[
\widetilde{F}_i = \frac{1}{k} \bigotimes \left( \left( \widehat{R}_{i1} \otimes \widehat{W}_1 \right) \oplus \left( \widehat{R}_{i2} \otimes \widehat{W}_2 \right) \oplus \cdots \oplus \left( \widehat{R}_{ip} \otimes \widehat{W}_p \right) \right),
\]

\[i = 1, 2, \ldots, m.\] (10)

According to the extension principle, we can represent \( \widetilde{F}_i \) as \( \widetilde{F}_i \cong (t^{xi}, t^{qi}, t^{ci}, t^{zi}), \)
where

\[t^{xi} = \sum_{p=1}^{k} t^{xip} t^{i}/k, \quad t^{qi} = \sum_{p=1}^{k} t^{qip} t^{i}/k, \quad t^{ci} = \sum_{p=1}^{k} t^{kip} t^{i}/k, \quad t^{zi} = \sum_{p=1}^{k} t^{zip} t^{i}/k, \]
for \( i = 1, 2, \ldots, m. \)

3.5 Select the best solution

Based on the ranking method in Section 2.3, for the aggregation appropriateness ratings \( \widetilde{F}_i \) in the above alternatives, the ranking value \( U^k_i (\widetilde{F}_i) \) can be represented as:

\[
U^k_i (\widetilde{F}_i) = \delta \left[ \frac{t^{zi} - x_i}{x_i - x_j - t^{ci} + t^{zi}} \right] + (1 - \delta) \left[ \frac{x_j - t^{yi}}{x_i - x_j + t^{ci} - t^{zi}} \right],
\]
where \( i = 1, 2, \ldots, m \), \( x_i = \min \{ t^{xi}, t^{zi}, \ldots, t^{yi} \} \), \( x_j = \max \{ t^{zi}, t^{yi}, \ldots, t^{zi} \} \), and \( 0 \leq \delta \leq 1. \)

The TRAI of the DMs in Eq. (11), i.e., the value of \( \delta \), must be obtained. According to Section 2.3, the information of the data input stage [24] is used to determine the value of \( \delta \). Therefore, according to the fuzzy MCDM model developed in this article (Section 3.2 to 3.4), the value of \( \delta \) of TRAI can be represented as:

\[
\delta = \frac{\delta_{w} + \delta_{w} + \delta_{s}}{(k \times n) + (n \times \sum_{p=1}^{k} n_p) + (m \times n \times \sum_{p=1}^{k} n_p)},
\]
where

\[
\delta_{w} = \sum_{p=1}^{k} \sum_{b=1}^{n_p} \frac{t^{u_{p\delta}} - t^{v_{p\delta}}}{(t^{d_{p\delta}} - t^{b_{p\delta}}) + (t^{v_{p\delta}} - t^{u_{p\delta}})},
\]
\[
\delta_{w} = \sum_{p=1}^{k} \sum_{j=1}^{n_p} \sum_{b=1}^{n_p} \frac{t^{u_{p\delta}} - t^{v_{p\delta}}}{(t^{d_{p\delta}} - t^{b_{p\delta}}) + (t^{v_{p\delta}} - t^{u_{p\delta}})},
\]
and

\[
\delta_{s} = \sum_{i=1}^{m} \sum_{p=1}^{k} \sum_{j=1}^{n_p} \sum_{b=1}^{n_p} \frac{t^{u_{p\delta}} - t^{v_{p\delta}}}{(t^{d_{p\delta}} - t^{b_{p\delta}}) + (t^{v_{p\delta}} - t^{u_{p\delta}})}.
\]

Finally, based on Eq. (11) and (12), the final ranking values \( U^k_i (\widetilde{F}_i) \) of \( m \) alternatives can be calculated. Therefore, the decision-making committee will determine the best alternative based on the ranking rules in Section 2.3.
4. The numerical illustration

In this section, a hypothetical case is taken as an example to illustrate the fuzzy MCDM selection model proposed in this study. The operational process is described as follows.

**Step 1.** Assume that the HR department of an international shipping company intends to promote one middle manager in the operation department. There are three experts, A, B, and C, who form a selection panel to select the best manager among the X, Y, and Z middle manager candidates. The selection criteria for this case are based on the five main criteria and 20 sub-criteria described in Section 3.1.

<table>
<thead>
<tr>
<th>Criteria / Sub-criteria</th>
<th>DMs</th>
<th>LVs</th>
<th>Fuzzy weights</th>
<th>Criteria / Sub-criteria</th>
<th>DMs</th>
<th>LVs</th>
<th>Fuzzy weights</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>A H</td>
<td></td>
<td>(0.667, 0.767, 0.867, 0.933)</td>
<td>C11</td>
<td>A H</td>
<td></td>
<td>(0.40, 0.50, 0.60, 0.70)</td>
</tr>
<tr>
<td></td>
<td>B H</td>
<td></td>
<td></td>
<td></td>
<td>B L</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C VH</td>
<td></td>
<td></td>
<td></td>
<td>C M</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C2</td>
<td>A M</td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.867)</td>
<td>C21</td>
<td>A H</td>
<td></td>
<td>(0.667, 0.767, 0.867, 0.90)</td>
</tr>
<tr>
<td></td>
<td>B VH</td>
<td></td>
<td></td>
<td></td>
<td>B M</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C H</td>
<td></td>
<td></td>
<td></td>
<td>C VH</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C3</td>
<td>A M</td>
<td></td>
<td>(0.533, 0.633, 0.733, 0.80)</td>
<td>C31</td>
<td>A M</td>
<td></td>
<td>(0.533, 0.633, 0.733, 0.833)</td>
</tr>
<tr>
<td></td>
<td>B VH</td>
<td></td>
<td></td>
<td></td>
<td>B H</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C M</td>
<td></td>
<td></td>
<td></td>
<td>C H</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C4</td>
<td>A H</td>
<td></td>
<td>(0.667, 0.767, 0.867, 0.933)</td>
<td>C41</td>
<td>A H</td>
<td></td>
<td>(0.467, 0.567, 0.667, 0.767)</td>
</tr>
<tr>
<td></td>
<td>B VH</td>
<td></td>
<td></td>
<td></td>
<td>B M</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C H</td>
<td></td>
<td></td>
<td></td>
<td>C M</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C5</td>
<td>A H</td>
<td></td>
<td>(0.533, 0.633, 0.733, 0.833)</td>
<td>C51</td>
<td>A VH</td>
<td></td>
<td>(0.733, 0.833, 0.933, 0.967)</td>
</tr>
<tr>
<td></td>
<td>B H</td>
<td></td>
<td></td>
<td></td>
<td>B VH</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C M</td>
<td></td>
<td></td>
<td></td>
<td>C H</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C11</td>
<td>A M</td>
<td></td>
<td>(0.667, 0.767, 0.867, 0.90)</td>
<td>C61</td>
<td>A M</td>
<td></td>
<td>(0.533, 0.633, 0.733, 0.833)</td>
</tr>
<tr>
<td></td>
<td>B VH</td>
<td></td>
<td></td>
<td></td>
<td>B H</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C VH</td>
<td></td>
<td></td>
<td></td>
<td>C H</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C12</td>
<td>A M</td>
<td></td>
<td>(0.333, 0.433, 0.533, 0.633)</td>
<td>C62</td>
<td>A M</td>
<td></td>
<td>(0.467, 0.567, 0.667, 0.767)</td>
</tr>
<tr>
<td></td>
<td>B L</td>
<td></td>
<td></td>
<td></td>
<td>B M</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C M</td>
<td></td>
<td></td>
<td></td>
<td>C M</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C13</td>
<td>A VH</td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.867)</td>
<td>C63</td>
<td>A VH</td>
<td></td>
<td>(0.667, 0.767, 0.867, 0.933)</td>
</tr>
<tr>
<td></td>
<td>B H</td>
<td></td>
<td></td>
<td></td>
<td>B H</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C M</td>
<td></td>
<td></td>
<td></td>
<td>C H</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C14</td>
<td>A M</td>
<td></td>
<td>(0.533, 0.633, 0.733, 0.80)</td>
<td>C64</td>
<td>A VH</td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.867)</td>
</tr>
<tr>
<td></td>
<td>B M</td>
<td></td>
<td></td>
<td></td>
<td>B VH</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C VH</td>
<td></td>
<td></td>
<td></td>
<td>C M</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C21</td>
<td>A M</td>
<td></td>
<td>(0.40, 0.50, 0.60, 0.70)</td>
<td>C71</td>
<td>A H</td>
<td></td>
<td>(0.40, 0.50, 0.60, 0.70)</td>
</tr>
<tr>
<td></td>
<td>B H</td>
<td></td>
<td></td>
<td></td>
<td>B L</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C L</td>
<td></td>
<td></td>
<td></td>
<td>C M</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C22</td>
<td>A M</td>
<td></td>
<td>(0.40, 0.50, 0.60, 0.70)</td>
<td>C72</td>
<td>A VH</td>
<td></td>
<td>(0.667, 0.767, 0.867, 0.90)</td>
</tr>
<tr>
<td></td>
<td>B M</td>
<td></td>
<td></td>
<td></td>
<td>B M</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C M</td>
<td></td>
<td></td>
<td></td>
<td>C VH</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C23</td>
<td>A M</td>
<td></td>
<td>(0.533, 0.633, 0.733, 0.833)</td>
<td>C74</td>
<td>A H</td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.867)</td>
</tr>
<tr>
<td></td>
<td>B H</td>
<td></td>
<td></td>
<td></td>
<td>B VH</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C H</td>
<td></td>
<td></td>
<td></td>
<td>C M</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C24</td>
<td>A VH</td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.867)</td>
<td>C75</td>
<td>A H</td>
<td></td>
<td>(0.40, 0.50, 0.60, 0.70)</td>
</tr>
<tr>
<td></td>
<td>B M</td>
<td></td>
<td></td>
<td></td>
<td>B L</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C H</td>
<td></td>
<td></td>
<td></td>
<td>C M</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
A fuzzy evaluation model of choosing a middle manager for an international shipping service provider

<table>
<thead>
<tr>
<th>Sub-criteria</th>
<th>DMs</th>
<th>LVs</th>
<th>Performance values</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>X</td>
<td>Y</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.20, 0.267, 0.40, 0.50)</td>
<td>(0.733, 0.833, 0.933, 0.967)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.20, 0.233, 0.40, 0.50)</td>
<td>(0.733, 0.833, 0.933, 0.967)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.267, 0.333, 0.467, 0.567)</td>
<td>(0.267, 0.333, 0.467, 0.567)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.867)</td>
<td>(0.667, 0.767, 0.867, 0.933)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.533, 0.633, 0.733, 0.80)</td>
<td>(0.533, 0.633, 0.733, 0.80)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.467, 0.567, 0.667, 0.767)</td>
<td>(0.733, 0.833, 0.933, 0.967)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.40, 0.50, 0.60, 0.70)</td>
<td>(0.40, 0.50, 0.60, 0.70)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.20, 0.267, 0.40, 0.50)</td>
<td>(0.467, 0.567, 0.667, 0.733)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.333, 0.40, 0.533, 0.633)</td>
<td>(0.333, 0.40, 0.533, 0.633)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.867)</td>
<td>(0.733, 0.833, 0.933, 0.967)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.40, 0.50, 0.60, 0.667)</td>
<td>(0.60, 0.70, 0.80, 0.833)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.267, 0.333, 0.467, 0.567)</td>
<td>(0.733, 0.833, 0.933, 0.967)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.867)</td>
<td>(0.667, 0.767, 0.867, 0.933)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.867)</td>
<td>(0.733, 0.833, 0.933, 0.967)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.40, 0.50, 0.60, 0.667)</td>
<td>(0.733, 0.833, 0.933, 0.967)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.20, 0.267, 0.40, 0.50)</td>
<td>(0.40, 0.467, 0.60, 0.667)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.533, 0.633, 0.733, 0.80)</td>
<td>(0.733, 0.833, 0.933, 0.967)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(0.60, 0.70, 0.80, 0.80)</td>
<td>(0.60, 0.70, 0.80, 0.80)</td>
</tr>
</tbody>
</table>
Ji-Feng Ding, Jung-Fong Kuo, Wen-Hui Tai

A fuzzy evaluation model of choosing a middle manager for an international shipping service provider

<table>
<thead>
<tr>
<th>B</th>
<th>G</th>
<th>G</th>
<th>VP</th>
</tr>
</thead>
<tbody>
<tr>
<td>C</td>
<td>G</td>
<td>G</td>
<td>P</td>
</tr>
</tbody>
</table>

$C_{53}$

<table>
<thead>
<tr>
<th>A</th>
<th>VP</th>
<th>VG</th>
<th>P</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>G</td>
<td>VG</td>
<td>P</td>
</tr>
<tr>
<td>C</td>
<td>P</td>
<td>G</td>
<td>VP</td>
</tr>
</tbody>
</table>

$C_{54}$

<table>
<thead>
<tr>
<th>A</th>
<th>VG</th>
<th>VG</th>
<th>VG</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>P</td>
<td>P</td>
<td>P</td>
</tr>
<tr>
<td>C</td>
<td>F</td>
<td>F</td>
<td>VP</td>
</tr>
</tbody>
</table>

**Step 2.** The three experts use the linguistic variables of Section 2.2 to evaluate the importance of the five main criteria and 20 sub-criteria, and then use the formula of Section 3.2 to obtain the fuzzy weights. The results are shown in Table 1. In addition, the three experts use the same method to evaluate the fuzzy ratings of all alternatives against all evaluation sub-criteria and use the formula of Section 3.3 to obtain the appropriateness rating. The results are shown in Table 2.

**Step 3.** Based on Eq. (9), the aggregation appropriateness ratings of the three candidates versus all evaluation sub-criteria can be obtained. The results are shown in Table 3. Furthermore, based on Eq. (10), we can obtain the final aggregation appropriateness ratings of the three candidates. The results are shown in Table 4.

**Table 3** The aggregation ratings of three candidates versus all evaluation sub-criteria

| $\bar{R}_{x1}$ | (0.170, 0.2455, 0.380, 0.4879) | $\bar{R}_{x2}$ | (0.3122, 0.4296, 0.5788, 0.6801) | $\bar{R}_{x3}$ | (0.1155, 0.1758, 0.3022, 0.4012) |
| $\bar{R}_{x2}$ | (0.1833, 0.2759, 0.40, 0.5284) | $\bar{R}_{x3}$ | (0.250, 0.3616, 0.4933, 0.6139) | $\bar{R}_{x3}$ | (0.1155, 0.1793, 0.3022, 0.4162) |
| $\bar{R}_{x3}$ | (0.2045, 0.2939, 0.4178, 0.5251) | $\bar{R}_{x4}$ | (0.2844, 0.3969, 0.5377, 0.6389) | $\bar{R}_{x4}$ | (0.1535, 0.2222, 0.3469, 0.4511) |
| $\bar{R}_{x4}$ | (0.2301, 0.3321, 0.4668, 0.5913) | $\bar{R}_{x5}$ | (0.3278, 0.4482, 0.6012, 0.7213) | $\bar{R}_{x5}$ | (0.1344, 0.2035, 0.3344, 0.4481) |
| $\bar{R}_{x5}$ | (0.2722, 0.3824, 0.5256, 0.6374) | $\bar{R}_{x5}$ | (0.3866, 0.5199, 0.6733, 0.7677) | $\bar{R}_{x5}$ | (0.1965, 0.2818, 0.4198, 0.5201) |

**Table 4** The final aggregation appropriateness ratings of the three candidates

| $\bar{F}_{x}$ | (0.1016, 0.1772, 0.2978, 0.4283) |
| $\bar{F}_{y}$ | (0.1507, 0.2510, 0.3936, 0.5308) |
| $\bar{F}_{z}$ | (0.0675, 0.1218, 0.2306, 0.3447) |

**Step 4.** Based on Eq. (12), we can obtain the TRAI of the three experts, i.e.,

$$\delta = \frac{7.8072 + 34.4985 + 89.8560}{5 \times 3 + 3 \times 20 + 3 \times 3 \times 20} = 0.5183;$$

therefore, according to the data at the data input stage, the overall risk attitude of the selection committee formed by these three experts is optimistic.

Then, by utilizing Eq. (11), we can obtain

$$x_{i} = \min\{0.1016, 0.1507, 0.0675\} = 0.0675,$$

$$x_{r} = \max\{0.4283, 0.5308, 0.3447\} = 0.5308,$$
A fuzzy evaluation model of choosing a middle manager for an international shipping service provider

Ji-Feng Ding, Jung-Fong Kuo, Wen-Hui Tai

\[
U_T^R(\tilde{F}_x) = (0.5183) \times \left[ \frac{0.4283 - 0.0675}{0.5308 - 0.0675 - 0.2978 + 0.4283} \right] + \\
(1 - 0.5183) \times \left[ 1 - \frac{0.5308 - 0.1016}{0.5308 - 0.0675 + 0.1772 - 0.1016} \right] = 0.4130,
\]

\[
U_T^R(\tilde{F}_1) = 0.5567, \\
U_T^R(\tilde{F}_2) = 0.2994.
\]

From the above results, the final ranking value of the three middle-manager candidates can be found. The results can be shown as \( U_T^R(\tilde{F}_1) > U_T^R(\tilde{F}_x) > U_T^R(\tilde{F}_2) \). Therefore, based on the ranking rules in Section 2.3, the HR department of the international shipping company will suggest that candidate \( Y \) is the best middle manager.

5. Conclusions

The international shipping logistics industry is an important logistics auxiliary for the international transportation industry. In order to provide better shipping logistics services, enterprises must recruit, cultivate, and retain outstanding talents continuously so as to form an efficient work team and achieve the organizational goal. Excellent manpower is also a key factor of a successful enterprise. In particular, middle managers play the role of transferring important information between the operational department and the decision-making department among the organizational managers. Therefore, selecting a middle manager with management competency and capability has a decisive impact on organizational development. Especially, HR departments often have to choose from many candidates during the process of employee evaluation and selection. In order to make a choice, it is necessary to set criteria for evaluation and comparison, so as to choose the most suitable manager from numerous candidates. In addition, a middle manager’s management competency and capability involves numerous evaluation dimensions and evaluation criteria, and the research scope and level covered by it is extensive and complex. As a result, the main purpose of this article was to establish a fuzzy evaluation model so that international shipping service providers can make the most suitable choice under a fuzzy environment.

In this article, a five-step fuzzy MCDM model was proposed to improve the quality of decision-making for choosing the best middle manager. In the proposed evaluation model, combining the academic literature and the scholars’ and experts’ opinions, a hierarchy structure with five assessment aspects, 20 assessment criteria and three candidates was constructed. The fuzzy weights for all assessment aspects and assessment criteria were obtained using the concept of linguistic variables and the arithmetic mean, as well as the evaluation of the performance value (the appropriateness rating) in this study. The comprehensive evaluation values for all alternatives were then evaluated. Based on the ranking method of the maximizing and minimizing sets, the best middle manager will be finally determined.

We applied a simulation example to interpret the calculation process of this fuzzy MCDM model. In this numerical case, a hierarchy structure was developed. Then, a three-member-committee was formed to thoroughly evaluate the three potential candidates of middle managers in order to select the most qualified one. In addition, the overall risk attitude of the selection committee is optimistic, which is based upon the procedure of the data input stage. The risk attitude indicates that the committee with three experts is risk-lovers. Hence, based on the proposed fuzzy MCDM evaluation model, the candidate \( Y \) is finally chosen as the best middle manager for the international shipping company. Moreover, in the end, the
A fuzzy evaluation model of choosing a middle manager for an international shipping service provider

Ji-Feng Ding, Jung-Fong Kuo, Wen-Hui Tai

The evaluation model and calculation process of this article were used to achieve the goal set by this paper.

The evaluation model developed in this article can be developed into a practical tool for business applications. In a fuzzy environment, companies can use this model to develop a decision support system to help them make decisions on related choices. Furthermore, the proposed fuzzy evaluation model can be applied in the similar decision-making problems [29, 30], such as partner selection of a strategic alliance in liner shipping carriers, best shipbuilding selection problem, best vessel selection, location choice of logistic centres, location selection of transshipment ports, and so on.

REFERENCES


A fuzzy evaluation model of choosing a middle manager for an international shipping service provider


Submitted: 22.06.2018.
Accepted: 05.02.2019.

107
In ice-infested waters, propellers of a polar ship are likely to be exposed to ice loads in different scenarios. Propeller milling with ice is one of the most dangerous cases for ice-propeller interaction. In this study, we try to simulate dynamic milling process of ice-propeller and reproduce resulting physical phenomena. Cohesive element method is used to model ice in the simulation. To simulate material properties of ice, an elastoplastic softening constitutive law is developed. Both crushing and fracture failures are included in the ice-propeller milling process. The ice loads in 6 Dofs acting on blades of a propeller are calculated in time domain. The average and standard deviations of simulated dominant ice loads are compared with those from model test. A good agreement is achieved. By varying propeller rotation speed, advance velocity and cutting depth on ice block, the sensitivity study has been carried out. The results show that dominant ice loads are affected much by the three parameters. It is shown that decreasing rotation speed, or increasing advance velocity and cutting depth may lead to higher ice loads. Care should be taken to avoid over-loading on propeller when operating in ice for polar ship.

**Key words:** propeller-ice milling; cohesive element; constitutive law; ice loads; propeller

1. **Introduction**

As global warming and ice shrinking in the polar waters, increasing voyages are expected to come for polar ships. As an essential equipment, propeller installed on polar ships may be subjected to ice loads in different ways since sea ice exists widely. It is not enough to check only the hydrodynamic loads on the propeller as studied by Ghassabzadeh et al. [1], but it is also necessary to estimate the ice loads on the propeller. The impact load could result in damage to structure of the propeller and endanger shipping safety. Therefore, it is worth to investigate the ice-propeller interaction.
Generally speaking, ice-propeller interaction could be divided into three scenarios. The first one is called ice blockage, where ice block keeps a certain distance from propeller. The existing of ice has an influence on stream field close to propeller. The propulsion efficiency would decrease accordingly. The second one is direct collision of ice pieces with propeller. In this case, the ice pieces are often small enough and taken as rigid bodies so that they are not able to break during collision. The last one is that propeller mills with large ice block. The resulting ice forces are usually an order of magnitude larger than the forces occurred in the first two cases. Consequently, it is meaningful to do research on propeller-ice milling process.

Since the 1990s, full scale and model scale tests for propeller-ice interaction have been performed and used for prediction of ice loads. As a part of Joint Research Project Arrangement between Canada and Finland, many researchers carried out full-scale measurements by using simplified blades and sea ice [2, 3, 4, 5, 6]. Considering the cost problem, scale-model tests were also carried out with refrigerated model ice [7, 8, 9, 10, 11]. As for numerical method, only a few researchers have studied ice-propeller milling in complex conditions. Wang et al. [12, 13] introduced a numerical prediction method to calculate the ice milling action. Ye et al. [14] and Wang et al. [15] investigated propeller-ice contact using peridynamics method, and focused on analyzing the effect of factors influencing the contact load. Khan et al. [16] developed a numerical tool to assess the effects of propeller-ice interaction on the loads acting on the propellers. The tool was then calibrated based on the results from model tests.

In this paper, a numerical model is built up by combining cohesive element method with an elastoplastic softening constitutive law to simulate dynamic propeller milling with ice process. In order to validate the accuracy of the present model, a verification study is accomplished by comparing simulated result with the measurement from model test carried out at the ice tank of Institute for Ocean Technology (IOT) by Wang et al. [11]. The main factors that may affect the milling process, including rotation speed of screw, advance velocity and cutting depth, are considered in the sensitivity analysis. Finally, some conclusions are drawn based on simulation results.

2. **Numerical models**

When the propeller blade contacts with the ice, the ice failure includes crushing, fracture and crack. Wang et al. [11] have found that ice cracking arises in the interaction between ice and propeller in ice milling tests, and Khan et al. [16] observed crushing happened in propeller milling ice tests when the blade edge cut the ice block. Such crushing and fracture failures are also found in the interaction between ship and level ice [17, 18, 19, 20]. These dis-continuous problems are difficult to be solved by simply using traditional FEM method that is based on the continuous assumption. Thus, cohesive element method (CEM) is introduced to solve the occurrence and propagation of ice fracture and crack during the ice-structure interaction.

Cohesive element method (CEM) is an extension of cohesive zone model (CZM) in FEM simulations. CZM was firstly proposed by Hillerborg et al. [21] to model the local crack propagation of an unreinforced concrete beam. Due to the good performance of CEM in modeling the initiation and propagation of crack, many researchers have introduced CEM into ice mechanics to simulate the fracture of ice material [22, 23, 24, 25].

In the framework of CEM, the ice elements are divided into two kinds of elements: bulk elements and cohesive elements. As shown in Fig. 1, the cohesive elements are inserted into every internal faces between the neighboring finite bulk elements. The bulk elements and the cohesive elements share nodes on the contact surfaces, thus deformation and stress could transmit directly between them. The cohesive elements respond to the external tensile force,
Numerical simulation of ice milling loads on propeller blade with cohesive element method

Feng Wang, Zao-Jian Zou, Li Zhou
Yang Wang, Hao Yu, Haihua Zhang

and cohesive surface will separate based on a traction-separation law. After reaching failure criteria, the cohesive surfaces are completely separated and the cohesive elements will be damaged. Thus, the crack is generated and propagates along the path of cohesive surfaces. On the other hand, the bulk elements also respond to the external force based on its own constitutive model.

Fig. 1. Illustration of bulk elements and cohesive elements

The most commonly used traction-separation law in CEM is linear softening model, which is firstly proposed by Mi et al. [26]. The function of linear softening model is listed as follows,

\[
T = \begin{cases} 
\frac{T_0 \delta_n}{\delta_n^f} & 0 \leq \delta_n < \delta_n^f \\
\frac{T_0}{(\delta_n^f - \delta_n)} (\delta_n - \delta_n^f) & \delta_n^f \leq \delta_n < \delta_n^f \\
0 & \delta_n \geq \delta_n^f 
\end{cases} \quad (1)
\]

\[
S = \begin{cases} 
\frac{S_0 \delta_i}{\delta_i^f} & 0 \leq \delta_i < \delta_i^f \\
\frac{S_0}{(\delta_i^f - \delta_i)} (\delta_i - \delta_i^f) & \delta_i^f \leq \delta_i < \delta_i^f \\
0 & \delta_i \geq \delta_i^f 
\end{cases} \quad (2)
\]

where \( T \) and \( S \) are the cohesive forces in normal direction (mode I fracture) and tangential direction (mode II fracture), respectively; \( \delta_n \) and \( \delta_i \) are crack separations in normal and tangential direction; \( T_0 \) and \( S_0 \) are the maximum stress in normal and tangential direction which are called fracture stress, while \( \delta_n^f \) and \( \delta_i^f \) are the corresponding crack separations, \( \delta_n^f \) and \( \delta_i^f \) are the maximum crack separations in normal and tangential direction. Fig. 2 shows the relationship between normal separation and cohesive force for mode I and II fracture. Taking mode I fracture as an example, \( T \) at the crack tip zone firstly linearly increases with the growth of \( \delta_n \) under the external loads. When \( T \) reaches maximum stress \( T_0 \), the crack begins to arise from this time. With the continuous development of \( \delta_n \), the material stiffness presents linear softening with the gradually expansion of the crack. When \( T \) decreases to zero, \( \delta_n \) reaches maximum crack separations \( \delta_n^f \), which means the cohesive element is damaged and
the adjacent bulk elements separate. The fracture energy \( GIC \) (\( GIIC \) for mode II fracture) is released during this process, whose value is equal to the area of the traction-separation curve.

\[
\text{Traction-separation curve:} \quad GIC = \int_{\delta_n^1}^{\delta_n} T \, d\delta_n
\]

**Fig. 2.** Relationship between cohesive force and crack separation (Left: model I; Right: model II)

Besides the fracture and crack, local crushing should also be considered in the simulations of ice-propeller interaction. However, the mesh applied in the FEM simulation is always too coarse to present the feature of local crushing, i.e., such microscopic failure is difficult to be explicitly modeled in the simulation. Hilding et al. [27] proposed a numerical approach, which took local crushing as a product of multiple micro-crack branching in the contact zone. An elastoplastic linear softening constitutive law, taken as a homogenization approach, was applied and have been proved to be effective by Wang et al. [25]. This constitutive law is applied to the ice bulk elements to model local crushing in the simulations, and its hardening curve is shown in **Fig. 3.** \( \sigma_Y \) is compressive strength which is the initial point of crushing. After experiencing a linear softening phase, crushed strain \( \varepsilon_c \) is obtained and henceforward the ice material is seen as totally crushed. When the strain comes to failure strain \( \varepsilon_f \), the ice bulk element is completely damaged and deleted.

\[
\begin{align*}
\sigma_Y & \quad \text{Crushing initial point} \\
\varepsilon_c & \quad \text{Softening phase} \\
\varepsilon_f & \quad \text{Totally crushed}
\end{align*}
\]

**Fig. 3.** Elastoplastic constitutive law [27]

3. **Numerical setup for propeller ice milling**

Numerical model was established according to ice milling model tests performed by Wang et al. [11] in the ice basin of Institute for Ocean Technology, National Research Council of Canada (IOT). The configuration of the model tests is shown in **Fig. 4.** Podded propulsor was fixed to a carriage and towed forward at a constant speed to mill intact ice block. The propeller was designed with reference to Stone Marine Meridian series [28]. The blades of the propeller were thicken for the safety of shipping in ice-covered waters. The
EG/AD/S ice was used in the experiments. The EG/AD/S ice is generated by the cooling of a diluted aqueous solution of ethylene glycol (EG), aliphatic detergent (AD), and sugar (S) under approximately -20°C, which can provide similar material properties with the first-year columnar sea ice.

![Fig. 4. The model test configuration of ice-propeller milling [11]](image)

There are four blades in total. The diameter of propeller is 300mm. The mean pitch/diameter ratio is 0.76 and the expanded blade area ratio is 0.669. The model of propeller after meshing is shown in Fig. 5. The propeller is taken as rigid body so that deformation of propeller under ice action is neglected. The material properties of the propeller are given in Table 1.

![Fig. 5. Propeller model after meshing (left: face; right: back)](image)

<p>| Table 1 Material properties of propeller model |</p>
<table>
<thead>
<tr>
<th>Items</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m³)</td>
<td>7850</td>
</tr>
<tr>
<td>Elastic modulus (GPa)</td>
<td>200</td>
</tr>
<tr>
<td>Poisson ratio</td>
<td>0.3</td>
</tr>
</tbody>
</table>

The model of ice block is built up as shown in Fig. 6. The size is 300mm in length, 70mm in width and height, which was determined by a prior verification study to eliminate the influence of boundary effect and meantime guarantee the calculation efficiency. The ice model was made of hexahedral ice bulk elements and cohesive elements. Both ice local crushing failure and fracture failure are considered. The elastoplastic constitutive law is used to simulate local crushing failure, while cohesive element is used to simulate fracture failure.
Table 2 presents the properties of ice bulk elements and cohesive elements applied in the simulations. The compressive strength and shear strength ($S_0$) are chosen based on the measured values in the model test provided by Wang et al. [11]. The other parameters are determined with reference to the work by Timco et al. [29] on the engineering properties of first-year sea ice.

<table>
<thead>
<tr>
<th>Bulk elements</th>
<th>Cohesive elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Items</td>
<td>Value</td>
</tr>
<tr>
<td>Density (kg/m$^3$)</td>
<td>910</td>
</tr>
<tr>
<td>Elastic modulus (GPa)</td>
<td>5</td>
</tr>
<tr>
<td>Poisson ratio</td>
<td>0.3</td>
</tr>
<tr>
<td>Compressive strength (KPa)</td>
<td>200</td>
</tr>
<tr>
<td>Crushed strain $\varepsilon_c$</td>
<td>0.35</td>
</tr>
<tr>
<td>Failure strain $\varepsilon_f$</td>
<td>0.5</td>
</tr>
</tbody>
</table>

The ice-propeller milling model is set up in LS-DYNA for numerical simulation. The propeller rotates at a constant angular speed of 5 rps clockwise and moves along minus X axis at 0.5 m/s. The blades mill ice with back side. The cutting depth is 35 mm, which is the same as model test setup. Moreover, in order to simulate propeller milling with infinitely wide ice block, the boundary of both sides of ice block are fixed in 6 degrees of freedom as shown in Fig. 7. The most commonly used contact algorithm built-in LS-DYNA is penalty function method, which is defined by the keyword of CONTACT. Command of CONTACT-ERODING-NODES-TO-SURFACE is suitable for modeling the contact between two bodies with considerably different stiffness, which is used to detect ice-propeller contact and calculate the interaction force. The friction coefficient between ice and propeller is set to 0.2. The ice-ice contact is detected by using another command named CONTACT-ERODING-SINGLE-SURFACE, which is suitable for modeling the contact between two bodies with similar stiffness. The corresponding friction coefficient is set to 0.1. The explicit time
integration algorithm is used in LS-DYNA and time step is set up automatically based on minimum element size.

4. Verification and validation of numerical methods

4.1.1 Sensitivity study on ice mesh size

It is important to study the effect of ice mesh size on simulation results. The ice mesh size refers to the length of ice bulk element and cohesive element, as shown in Fig. 6. In this section, sensitivity study is performed by changing ice mesh size. Then the simulated results are compared with model test results to validate the numerical model. The four kinds of ice meshes are studied with sizes of 2.5mm, 3mm, 3.75mm and 4.28mm, which are $d/120$, $d/100$, $d/80$ and $d/70$ respectively, where $d$ is the diameter of the propeller. They are labelled as S1, S2, S3 and S4 respectively. Fig. 8 shows the calculated resultant forces and moments at time interval 0.15s~0.20s. Only one blade milled ice block within the time interval. It can be found that both resultant forces and moments tend to rise unsteadily at the first stage and then decrease non-monotonously afterwards. Since there is only one blade interacting with ice, the global trend of the forces and moments is due to invasion and withdrawal of the blade in ice. The fluctuation of the force and moment signals is caused by local failures of ice block.

As the size of ice decreases, the ice loads tend to increase. However, this tendency becomes weak and the ice loads tend to be convergent when the size decreases to a certain value. It is clear from Fig. 8 that the simulated ice loads using S2 mesh are close to those using the finest mesh, S1 mesh. Table 3 shows the average value and the standard deviation of calculated ice loads. The relative errors compared to the corresponding ice loads using S1 mesh are also presented in brackets. It should be noted that the relative errors for S2 mesh are less than 5%. Therefore, balancing the computation efficiency and accuracy, S2 mesh is used in the following simulation cases.
Numerical simulation of ice milling loads on propeller blade with cohesive element method

(a) Resultant force

(b) Resultant moment

Fig. 8. Calculated time series of ice loads with different ice mesh sizes

Table 3 Comparison of ice loads for cases with different ice mesh sizes

<table>
<thead>
<tr>
<th>Mesh size</th>
<th>Resultant force (N)</th>
<th>Resultant moment (N·m)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mean</td>
<td>StD</td>
</tr>
<tr>
<td>S1</td>
<td>62.7 (-)</td>
<td>39.5 (-)</td>
</tr>
<tr>
<td>S2</td>
<td>59.8 (4.6%)</td>
<td>37.6 (4.8%)</td>
</tr>
<tr>
<td>S3</td>
<td>30.9 (50.7%)</td>
<td>20.6 (47.8%)</td>
</tr>
<tr>
<td>S4</td>
<td>16.7 (73.4%)</td>
<td>12.4 (68.6%)</td>
</tr>
</tbody>
</table>

4.1.2 Comparison with model test results

In the model tests carried out by Wang et al. [11], a load cell was installed on one of the four blades, which is also named as key blade. The resultant ice load and moment acting on the key blade from the simulations are used for comparison with model test results.
The time series of simulated and measured ice loads within 1 second are presented for comparison in Fig. 9. It can be seen that there are five typical peaks for both resultant forces and moments during the time interval. The peaks and trends of ice load signals from simulation agree well with model test results. Table 4 and Table 5 show the peak values and the moment when they occur for both simulated and measured ice loads. The relative errors between simulation and model test results are also included in the bracket. It is found that the peak values and time from simulation coincide well with model test results except the second peak. The relative errors between averaged ice forces and moments are as low as 0.4% and 9.1% respectively, which demonstrates that the simulated results are valid and effective. The trends of the simulated and measured ice loads for one blade cutting process differ from each other, which reflects the inherent random nature of ice-structure interaction process. It should be noted that the second peak of measured moment is obviously smaller than the other peaks. The reason may be that the material property of ice block changes during milling process. Moreover, during the time when the key blade is not interacting with the ice, it is exposed to hydrodynamic load at low level. Since the effect of seawater is not considered in the numerical model, the resulting load is taken as zero in the simulation.
Feng Wang, Zao-Jian Zou, Li Zhou
Yang Wang, Hao Yu, Haihua Zhang

Numerical simulation of ice milling loads on propeller blade with cohesive element method

Table 4 Comparison of peak values for ice force

<table>
<thead>
<tr>
<th>Peak</th>
<th>Experiment</th>
<th></th>
<th>Simulation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Time(s)</td>
<td>Value (N)</td>
<td>Time(s)</td>
</tr>
<tr>
<td>Peak1</td>
<td>0.156</td>
<td>155.9</td>
<td>0.149 (-4.5%)</td>
</tr>
<tr>
<td>Peak2</td>
<td>0.345</td>
<td>191.5</td>
<td>0.358 (3.8%)</td>
</tr>
<tr>
<td>Peak3</td>
<td>0.542</td>
<td>180.5</td>
<td>0.553 (2.0%)</td>
</tr>
<tr>
<td>Peak4</td>
<td>0.731</td>
<td>178.8</td>
<td>0.749 (2.5%)</td>
</tr>
<tr>
<td>Peak5</td>
<td>0.931</td>
<td>162.7</td>
<td>0.949 (1.9%)</td>
</tr>
<tr>
<td>Mean of 5 peaks</td>
<td>-</td>
<td>173.9</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 5 Comparison of peak values for ice moment

<table>
<thead>
<tr>
<th>Peak</th>
<th>Experiment</th>
<th></th>
<th>Simulation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Time(s)</td>
<td>Value (N·m)</td>
<td>Time(s)</td>
</tr>
<tr>
<td>Peak1</td>
<td>0.146</td>
<td>19.02</td>
<td>0.149 (2.1%)</td>
</tr>
<tr>
<td>Peak2</td>
<td>0.328</td>
<td>12.61</td>
<td>0.358 (9.1%)</td>
</tr>
<tr>
<td>Peak3</td>
<td>0.544</td>
<td>17.07</td>
<td>0.553 (1.7%)</td>
</tr>
<tr>
<td>Peak4</td>
<td>0.736</td>
<td>17.84</td>
<td>0.750 (1.9%)</td>
</tr>
<tr>
<td>Peak5</td>
<td>0.944</td>
<td>18.75</td>
<td>0.953 (0.9%)</td>
</tr>
<tr>
<td>Mean of 5 peaks</td>
<td>-</td>
<td>17.06</td>
<td>-</td>
</tr>
</tbody>
</table>

In order to give a deeper comparison of the simulated and measured ice loads for one blade cutting process, Fig. 10 shows the simulated and measured ice loads between 0.48s~0.60s as an example. The fluctuated behaviour of simulated ice loads is caused by the local crushing and fracture of ice material, which reflects well the actual ice failure mode during ice-propeller interaction. The curves of measured loads are much more smooth than the simulated ones. This indicates that the measured curves are the filtered results, however, their sampling and filtered criterion are unknown. In order to facilitate the comparison, the simulated results are filtered by a low frequency filter and the filtered results are also shown in Fig. 10. It can be seen that the shape and trend of main peak in the smoothed load curves obtained from simulation approach those measured in the experiment. The low peaks arisen in the experimental loads are caused by the hydrodynamic loads, which are not considered in the simulations. Thus, it is reasonable that the smoothed simulation load levels are slightly lower than the measured ones due to neglect of hydrodynamic loads in the simulations.
Numerical simulation of ice milling loads on propeller blade with cohesive element method

Feng Wang, Zao-Jian Zou, Li Zhou
Yang Wang, Hao Yu, Haihua Zhang

4.1.3 Analysis of simulated results

The stress nephograms at different time instants are given in Fig. 11, from which the physical process of propeller milling with ice can be observed clearly. At the initial stage of the interaction, the blade starts to intrude ice and crushes against the ice block to cut the ice. As the intrusion process continues, the edge of the blade keeps crushing with ice block and paves a way to pass through. Simultaneously, the contact area between ice block and blade surface increases as the propeller disc advances into ice block. The blade surface may collide with ice block and ice pieces chopped from milling process to induce additional contact force as well. The deformation of ice fragments during the ice crushing process can be clearly observed in Fig. 11. As the edge of blade leaves from ice block, a piece of ice is cut off from the original ice block. It is clear to see from Fig. 11(d) that the cutting edge of ice block after milling is irregular which shows that the fracture and cracking happen during milling process. The reason is that the relative speed between ice block and blade is high at the contact point when the screw rotates fast. This leads to a high strain rate that may exceed the limit (10^3) of ductile-brittle transition area according to Carney et al. [30]. Thus, the ice block behaves brittle failure mode. The ice pieces chopped from ice block would continue to collide with rotating blade and get shattered into smaller pieces. In addition, it can be seen from Fig. 11...
that the highest effective stress appears at the contact area between the blade and ice block. With the continuous intrusion of blade into the ice block, more medium effective stress zones discretely distribute in the whole area of the ice block. This result is reasonable and reflects the propagation of stress between ice elements within the ice block.

Fig. 11. Stress nephograms during dynamic propeller-ice milling process

**Fig. 12** shows several time series of ice forces and moments in 6 Dofs during time interval of 0.15s ~ 0.2s. It can be seen clearly that the contact force has a trend of increasing first and decreasing after a peak value. This is mainly dependent on contact area between ice block and blade. The contact force augments to maximum when the contact area reaches the largest. Moreover, increasing rate of ice force is higher than decreasing rate clearly. This may be affected by shape of the blade. On the other side, ice pieces cut off may still interact with propeller until they leave away from the propeller disk completely. The resulting impact force, as additional action, prevents the total ice load from dropping quickly. The longitudinal force $F_X$ and moment $M_Y$ are relatively larger than the other loads, which may threaten the structural safety of propeller. Therefore, these two loads are selected as the objects for comparison in the following study to analyze the influence of different key factors on the simulated loads.
5. Parametric study

Based on the numerical model, the effects of rotation speed, advance velocity and cutting depth on ice load in longitudinal and transverse directions are analyzed. The parameter setups of the simulation are kept the same as those used in Section 3 if not stated else. The calculated results from all cases are compared with the basic case. The average value and standard deviation of ice load components $F_X$ and $M_Y$ are of interest herein. The load components from other cases are shown in percentage compared to the results of the basic case whose components are taken as 100%.

5.1.1 Rotation speed

In this section, five different rotation speeds are used for investigating how much the ice load components would be affected. They are 3, 4, 5, 6 and 7 rps, where 5 rps was applied to the basic case. The average values and standard deviations of ice load components $F_X$ and $M_Y$ under different rotation speeds are given in Fig. 13. It is found that both load components $F_X$ and $M_Y$ show similar trend as rotation speed increases. The average and standard deviation descend successively as the rotation speed increase. Especially for the average, it decreases in
a linear way approximately. Fig. 14 presents the time series of $F_X$ and $M_Y$ with rotation speeds of 3, 5 and 7 rps within the same range of phase angle from simulations. It is seen from Fig. 14 that the effect of rotation speed is minor at the load increasing stage, but major in the load dropping stage. The more slowly the propeller rotates, the larger the peak value of load time series tends to become. This is mainly attributed to that it takes longer for full contact of ice-propeller under the condition of slow propeller rotation. Moreover, the possibility of chopped ice pieces colliding with blades increases, which may result in slow declining of ice loads when the blade leaves off ice block. Therefore, to speed up propeller rotation to some extent could be helpful to break ice block efficiently and also reduce ice loads to improve the safety of propeller operation in ice-covered waters.
5.1.2 Advance velocity

Five different advance velocities are used for investigating how much the ice load components would be influenced. The screw is advancing straightly to the ice block at 0.2, 0.35, 0.5, 0.65 and 0.8 m/s respectively, where 0.5 m/s is used in the basic case. Fig. 15 shows the curves of average values and standard deviations of ice load components $F_X$ and $M_Y$ as function of advance velocity. It is clear that both load components $F_X$ and $M_Y$ increase significantly as the advance velocity increases, especially for average ice load components.

Fig. 16 presents time series of ice load $F_X$ and $M_Y$ at advance velocities of 0.2, 0.5 and 0.8 m/s. We can see that the simulated ice loads are affected by relative velocity between ice and propeller strongly. During the ice milling process, the ice loads increase significantly with the increasing velocity. The peak of ice loads at the highest velocity 0.8 m/s is almost twice the peak loads from the basic case with 0.5 m/s. This may pose a great challenge to the safe use of propeller in ice. Therefore, it is suggested that the advance velocity should be well controlled to avoid damage of propeller.
5.1.3 Cutting depth

Five cutting depths are used to study the influence of cutting depth on ice loads. The cutting depths of 21, 24.5, 28, 31.5 and 35 mm are set up in the simulations, where 28 mm is used in the basic case. Fig. 17 shows variations of average values and standard deviations of ice loads as a function of cutting depth. It is clear that both load components $F_X$ and $M_Y$ increase significantly as the cutting depth increases, especially for average ice load components.

Fig. 18 presents time series of ice load $F_X$ and $M_Y$ at cutting depths of 21, 28 and 35 mm. It is seen that as the velocity increases, the ice load level increases at both descending and ascending stages. This trend is caused by the long interaction process and large contact area for ice and blade with big cutting depth. Moreover, the edge of blade enters ice block ahead of expected time when cutting depth become large. Simultaneously, it is found that ice loads decrease to zero at the same time for different cases. The reason is that at the exiting stage the edge of the blade leaves away from the ice block before the surface. From Fig. 11 (d), it can observed that the contact force is small and can be neglected though the surface still interacts with chopped ice pieces. Thus, the time when the load turns to zero depends on when the edge of blade exits from ice block. Fig. 19 shows the stress nephograms with different cutting depths at the same moment before the blade exits from ice block. From the figure, it can be found that the exiting time of the blade edge for different cutting depths is almost the same and thus the loads drop to zero simultaneously. This conclusion coincides with that obtained by Wang et al. [15].
Numerical simulation of ice milling loads on propeller blade with cohesive element method

Feng Wang, Zao-Jian Zou, Li Zhou
Yang Wang, Hao Yu, Haihua Zhang

Fig. 17. Statistics of ice load $F_X$ and $M_Y$ based on simulations with different cutting depths

Fig. 18. Time series of simulated ice load $F_X$ and $M_Y$ with different cutting depths

Fig. 19. Stress nephograms at the same moment with different cutting depths (a. 21mm; b. 28mm; c. 35mm)
6. Conclusion

In this paper, we develop a numerical method to simulate continuous ice-propeller milling process based on an elastoplastic softening constitutive law with cohesive element method, which is implemented in commercial finite element software LS-DYNA. The present numerical method models ice cracking and crushing behaviors happened in the ice-propeller interaction, and the numerical results are validated by comparing with the available experimental results. Then, the effects of rotation speed of screw, advance velocity and cutting depth on dominant ice load components are analyzed and discussed. The main conclusions can be drawn as follows:

(1) There exists a large fluctuation of simulated ice loads due to local failures of ice and small pieces of ice generated. The simulated time series of ice loads coincide well with model test results with respect to the peak values and the time when the peak values occur, indicating that the linear softening constitutive law used in cohesive element method is capable of modeling propeller-ice milling process.

(2) Ice loads are affected by setup of ice mesh size. The loads will increase with decreasing ice mesh size. The ice loads tend to be steady when the size increases to certain level.

(3) According to the sensitivity study, the ice loads are affected by rotation speed of screw, advance velocity and cutting depth to varying degrees in terms of average and standard deviation values. The dominant ice loads decrease as the rotation speed increases and increase as the advance velocity or cutting depth increases. The ice loads are influenced by the advance velocity most compared to the other two factors.

It is shown that the present numerical method has a good prospect in modeling ice-propeller interactions. However, it should be noted that all conclusions are drawn based on the present simulation. Besides, the sensitivity of ice properties to the ice loads is not investigated herein and should be studied in the future.

ACKNOWLEDGEMENTS

The corresponding author (third author) greatly acknowledges the supports of the National Natural Science Foundation of China (Grant No. 51809124, 51709136), NSFC-STINT (Grant No. 5181102016), Natural Science Foundation of Jiangsu Province of China (Grant No. BK20170576), Natural Science Foundation of the Higher Education Institutions of Jiangsu Province of China (Grant No. 17KJB580006) and State Key Laboratory of Ocean Engineering (Shanghai Jiao Tong University) (Grant No. 1704).

REFERENCES


Numerical simulation of ice milling loads on propeller blade with cohesive element method


Submitted: 19.11.2018. Feng Wang, wangfeng0630@hotmail.com
School of Naval Architecture, Ocean and Civil Engineering, Shanghai Jiao Tong University, Shanghai 200240, China

Zao-Jian Zou, zjzou@sjtu.edu.cn
School of Naval Architecture, Ocean and Civil Engineering, Shanghai Jiao Tong University, Shanghai 200240, China

Collaborative Innovation Center for Advanced Ship and Deep-Sea Exploration, Shanghai Jiao Tong University, Shanghai 200240, China

Zhou Li, zhouli209@hotmail.com (Corresponding Author)
Jiangsu University of Science and Technology, Zhenjiang 212003, China

Yang Wang, maible@sjtu.edu.cn
School of Naval Architecture, Ocean and Civil Engineering, Shanghai Jiao Tong University, Shanghai 200240, China

Hao Yu, yugong9281224@163.com
China Shipbuilding Industry Corporation Limited, Beijing 100097, China

Haihua Zhang, zhanghaihua@702sh.com
Shanghai branch, China Ship Scientific Research Center, Shanghai 200011, China
REEFER VESSEL VERSUS CONTAINER SHIP

Summary

In the paper is presented a study of the economics of refrigerated cargoes transport with two specialized types of ship. Two modern ship designs are developed for the transport of refrigerated cargo, primarily bananas. One design is a reefer vessel with the ability to load a significant number of refrigerated containers on the main deck. The other design is a container ship specialized for the transport of refrigerated 40 feet containers. Both designs have the capacity to load the same mass of refrigerated cargo. Basic characteristics of the designs are compared, which can be the basis for answering the question: Which type of vessel to choose for the carriage of refrigerated cargo?

Key words: refrigerated vessel; container ship; ship design

1. Introduction

In recent years a dilemma has emerged: Is it more economical to transport refrigerated cargoes with reefer vessels or container ships? Development of different ship designs dedicated for transportation of refrigerated cargoes is elaborated in [1], where reefer ships are divided on traditional reefer ship for carriage of reefer cargoes in bulk, pure pallet ships and freezers.

In [2] container ships are, for quite understandable reasons, included in the family of “Linear Dimension” ships. Later, the same author in [3] includes container ships in the family of “Volume, Area and Dimension-Based Design” ships. In [1] container ships dedicated for transport large number of refrigerated containers are defined as separate ship type, commonly referred to as “Reefer Containers”.

Over time, the design of reefer vessels is developing in the direction of loading more and more containers in the cargo holds and on the main deck, so the differences between these two types are increasingly reduced. Also, both types have their own unique characteristics, so it can be said that each type has a number of differences (resulting as advantages or disadvantages) over the other type of ship.
2. Basic Design Requirements

Both designs have been developed under the same basic design requirement: to be able to load 12,000 t of refrigerated cargo. Also, both designs have been developed with the intent of loading the maximum number of (exclusively) refrigerated containers. Designs development was carried out in close cooperation with the specialised shipping companies based on “traditional” ship design spiral method. Speed-power calculation was carried out following the speed-power approximation method published in [4, 5], modified for design of reefer vessel and container ship. Special attention has been paid to energy efficiency and all related aspects [5, 6]. Basic design requirements are as follows:

- Cargo: abt. 11,440 high-cube (height 2.4 m) EURO pallets (1.2 x 1.0 m)
- Maximum draught 11.0 m
- Main engine MAN B&W 8G50ME-C9.6 Tier III
- Selected maximum continuous rating (SMCR) 11,980 kW at 100 rpm
- Continuous service rating (CSR) = 90% SMCR
- 40 feet ISO refrigerated high-cube containers (40’ x 8’ x 9.5’)
- Electrical sockets for all deck containers
- Electrical sockets for all containers in cargo holds
- Accommodation for 25 crew + 6 Suez crew

Additional requirements for reefer vessel:
- Dimension of EURO pallet with stowage margin: 1.24 x 1.03 m
- Clear tween deck height 2.55 m
- Aluminium gratings height 150 mm
- 4 cargo holds, 16 compartments, 8 independent cooling zones
- 2 twin deck cranes, each 2 x 20 t SWL
- Cargo hatch covers of folding type
- Cargo holds air recirculation

Additional requirements for container ship:
- 4 single deck cranes, each 40 t SWL
- Cargo hatch covers of pontoon type
- Cargo holds ventilation

3. Basic Design of P105 Reefer Vessel 765,000 cu.ft.

Based on the design requirements, a modern design of container-oriented reefer vessel is developed. She is a single screw diesel engine driven Reefer/Container Vessel with bulbous bow, long forecastle, poop deck and transom stern, Figures 1 and 2. Living quarters including navigation bridge and engine room located aft. Double bottom, bilge and peak tanks intended for segregated ballast. Cargo space divided into four (4) cargo holds subdivided into eight (8) refrigerated compartments with separate air cooling.

Intended cargo: Bananas, fruit, frozen meat, fish, palletized cargo, 40 feet containers

Main particulars:
Length overall .......................................................... abt. 178.6 m
Length b.p. ............................................................... 168.0 m
Breadth, mld. ............................................................ 27.2 m
Depth, mld. .............................................................. 15.0 m
Draught design ......................................................... 8.8 m
Draught scantling ...................................................... 11.0 m
Deadweight design ................................................... abt. 13,400 t
Deadweight scantling ................................................. abt. 21,400 t
Reefer Vessel versus Container Ship

Predrag Ćudina, Ana Bezić

Capacities:
Refrigerated cargo holds, total ................................................................. abt. 850,000 cu.ft.
Refrigerated cargo holds (up to 2.55 m height) ........................................... abt. 765,000 cu.ft.
Cargo deck area, total .............................................................................. abt. 8,550 m²
Number of Euro pallets (1.2 m x 1.0 m) ...................................................... abt. 6,600 pcs
Number of containers, holds/deck ............................................................. abt. 82/242 FEU
Heavy fuel oil .......................................................................................... abt. 1,800 m³
MGO/MDO ............................................................................................... abt. 370 m³
Fresh water .............................................................................................. abt. 200 m³
Feed water ............................................................................................... abt. 40 m³
Segregated water ballast, total ................................................................. abt. 6,000 m³
Cruising range ......................................................................................... abt. 18,000 n. miles

Main engine: MAN-B&W 8G50ME-C9.6 Tier III
SMCR 11,980 kW at 100 rpm, CSR 10,780 kW
Guaranteed speed: 21.0 knots in trial conditions, design draught and CSR, i.e. 10,780 kW
M.e. consumption: abt. 42.4 t/day of fuel oil LCV 42,707 kJ/kg

Basic loading conditions are shown in the following table.

Table 1 Basic Loading Conditions of P105

<table>
<thead>
<tr>
<th>Loading condition</th>
<th>Payload (t)</th>
<th>Mean draught departure/arrival (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Banana pallets (6,600 pcs) + 242 deck FEU (each 28 t)</td>
<td>6,930 + 6,776</td>
<td>11.0/10.5</td>
</tr>
<tr>
<td>82 FEU in holds + 224 deck FEU (each 28 t)</td>
<td>2,296 + 6,272</td>
<td>9.7/9.2</td>
</tr>
<tr>
<td>Loading condition: design draught</td>
<td>Payload (t)</td>
<td>Mean draught (m)</td>
</tr>
<tr>
<td>Banana pallets (6,600 pcs) + 150 FEU (each 28 t) + 25% stores</td>
<td>6,930 + 4,200</td>
<td>8.8</td>
</tr>
</tbody>
</table>

Fig. 1 Midship Section of P105
Fig. 2 General Arrangement of P105
4. Basic Design of P109 Container Ship 742 FEU

Based on the design requirements, a modern design of Container Ship specialized for loading 40’ refrigerated containers is developed. She is a single screw diesel engine driven Container Ship with bulbous bow, long forecastle, poop deck and transom stern. Living quarters including navigation bridge and engine room located aft, Figures 3 and 4. Double bottom, double side and peak tanks intended for segregated ballast. Cargo space divided into four (4) cargo holds.

Intended cargo: Refrigerated 40’ ISO high-cube containers (40’ x 8’ x 9.5’)

Main particulars:

Length overall .............................................................. abt. 177.0 m
Length b.p. ........................................................................ 166.0 m
Breadth, mld. ..................................................................... 27.5 m
Depth, mld. .......................................................................... 15.0 m
Draught design ................................................................... 9.5 m
Draught scantling ............................................................... abt. 17.800 t
Deadweight design ............................................................... abt. 23.700 t
Deadweight scantling ........................................................... abt. 23.700 t

Capacities:

Cargo holds, total .............................................................. abt. 30,000 m³
Number of 28 t refrigerated containers, holds/deck .......... abt. 269/303 FEU
Heavy fuel oil .................................................................... abt. 1,650 m³
MGO/MDO .......................................................................... abt. 500 m³
Fresh water ......................................................................... abt. 200 m³
Feed water .......................................................................... abt. 40 m³
Segregated water ballast, total .......................................... abt. 8,700 m³
Cruising range ................................................................. abt. 17,000 n. miles

Main engine: MAN-B&W 8G50ME-C9.6 Tier III
SMCR 11,980 kW at 100 rpm, CSR 10,780 kW

Guaranteed speed: 19.4 knots in trial conditions, design draught and CSR, i.e. 10,780 kW

M.e. consumption: abt. 42.4 t/day of fuel oil LCV 42,707 kJ/kg

Basic loading conditions are shown in the following table.

Table 2 Basic Loading Conditions of P109

<table>
<thead>
<tr>
<th>Loading condition: scantling draught</th>
<th>Payload (t)</th>
<th>Mean draught departure/arrival (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>269 FEU in holds + 303 deck FEU (each 28 t)</td>
<td>16,016</td>
<td>11.0/10.6</td>
</tr>
<tr>
<td>Loading condition: design draught</td>
<td>Payload (t)</td>
<td>Mean draught (m)</td>
</tr>
<tr>
<td>269 FEU in holds + 241 deck FEU (each 28 t) + 25% stores</td>
<td>14,280</td>
<td>9.5</td>
</tr>
</tbody>
</table>
Fig. 3 General Arrangement of P109
5. Comments on Designs

Both designs have a number of similar characteristics: main dimensions are almost the same, general configuration is very similar: four cargo holds, long forecastle, poop deck, accommodation situated aft.

The designs differ in characteristics that are specific to each type of ship. Reefer vessel has two 2 x 20 t SWL twin deck cranes and folding type hatch covers. Container ship is equipped with four 40 t SWL single deck cranes and pontoon type hatch covers. Reefer vessel loads palletized cargo on tank top and three tween decks. Alternatively, 5 tiers of high-cube FEU can be loaded on the tank top. Container ship also can load in cargo holds 5 tiers of high-cube FEU.

When comparing these two designs, attention should be paid to a very significant difference: in a case of a reefer vessel payload consists of palletized cargo and containers, but in a case of container vessel the payload consists exclusively of containers. What that means?

One high-cube FEU can be loaded with 20 high-cube EURO pallets. If bananas are loaded, each pallet has a mass of abt. 1.05 t. The gross mass of loaded refrigerated container is abt. 28 t and net mass of pallets is abt. 20 x 1.05 = 21 t. It leads to a result that container ship when loading 11,440 banana pallets (572 FEU x 20 pallets), loads cargo mass of 16,016 t (572 FEU x 28 t).

Reefer vessel, when loading banana pallets in cargo holds plus refrigerated FEU on the main deck, loads cargo mass of 6,930 t (6,600 pallets x 1.05) plus cargo mass of 6,776 t (242 FEU x 28 t). In total, it loads cargo mass of 13,706 t, which is 2,310 t less than the cargo mass loaded by the container ship. Thereby, both ships load the same quantity of banana pallets, totally 11,440 pieces.
6. Analysis of the Economy of Ships in Navigation

Analysis is carried out for the case of fully loaded ships in navigation and for the case of sailing in the ballast condition. Since the purpose of this paper is to carry out a comparison of two ship design dedicated for carriage of refrigerated cargoes, rather than the total economy of shipping company, the analysis is limited to the fuel costs of a voyage. In addition, it is important to note that fuel cost is the dominant item in the voyage costs [8].

6.1 Fully Loaded Ships

Both ships are fully loaded to the scantling draught and are observed under the following conditions:
- Ships are fully loaded on scantling draught
- Ships are sailing at the same speed in the service conditions (15% sea margin)
- Consumption of each refrigerated FEU is 7 kW, diversity factor is 0.7
- Consumption of reefer vessel refrigerating plant is 700 kW
- Cooling down period for bananas is 24 h (1 day) for reefer vessel
- Consumption of reefer vessel refrigerating plant during cooling down period is 1,400 kW
- Consumption of reefer vessel cargo holds ventilation / air recirculation is 500 kW
- Consumption of container ship cargo holds ventilation is 500 kW
- Consumption of other consumers is 650 kW

For the purpose to determine needed main engine power for both ships, following speed-power estimations are carried out using methodology published in [9] and correlated with results of prototype ship model testing [10, 11].

![Fig. 5 Speed-Power Estimation of P109 (Container Ship) on Scantling Draught](image1.png)

![Fig. 6 Speed-Power Estimation of P105 (Reefer Vessel) on Scantling Draught](image2.png)

Calculation of daily fuel oil consumption for both ships is carried out and shown in Tables 3 through 5. It is based on the declared conditions and needed main engine power for service speed of 17.9 knots. Reefer vessel cargo refrigerating plant during cooling down period is consuming 1,400 kW (instead of 700 kW during the period after cooling down ends), Figures 5 and 6.
### Table 3 Basic Electric Power Calculation

<table>
<thead>
<tr>
<th>Consumer</th>
<th>P105 (Reefer Vessel)</th>
<th>P109 (Container Ship)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>No of Peak load Div.</td>
<td>Power No of Peak load Div.</td>
</tr>
<tr>
<td></td>
<td>[-] [kW] [ ] [-] [kW]</td>
<td></td>
</tr>
<tr>
<td>Cargo refrigerating plant</td>
<td>1 700 1.0 700</td>
<td>1</td>
</tr>
<tr>
<td>Refrig. during cool. down</td>
<td>1 1,400 1.0 1,400</td>
<td></td>
</tr>
<tr>
<td>Holds vent./air circulation</td>
<td>1 500 1.0 500</td>
<td>1 500 1.0 500</td>
</tr>
<tr>
<td>Refrigerated FEU</td>
<td>242 7 0.7 1,186</td>
<td>572 7 0.7 2,803</td>
</tr>
<tr>
<td>Other consumers</td>
<td>1 650 1.0 650</td>
<td>1 650 1.0 650</td>
</tr>
<tr>
<td>Total during sailing</td>
<td>3,036</td>
<td>3,953</td>
</tr>
<tr>
<td>Total: sailing + cool. down</td>
<td>3,736</td>
<td></td>
</tr>
</tbody>
</table>

### Table 4 Daily Fuel Oil Consumption of Fully Loaded Ships

<table>
<thead>
<tr>
<th>Consumer</th>
<th>P105 (Reefer Vessel)</th>
<th>P109 (Container Ship)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Power sfoc dfoc</td>
<td>Power sfoc dfoc</td>
</tr>
<tr>
<td></td>
<td>[kW] [g/kWh] [t/day]</td>
<td>[kW] [g/kWh] [t/day]</td>
</tr>
<tr>
<td>Main engine</td>
<td>9,080 158.3 34.5</td>
<td>10,780 162.0 41.9</td>
</tr>
<tr>
<td>Diesel generator plant</td>
<td>3,036 192.0 14.0</td>
<td>3,953 192.0 18.2</td>
</tr>
<tr>
<td>DG plant during cool. down</td>
<td>3,736 192.0 17.2</td>
<td></td>
</tr>
<tr>
<td>Total (LCV=42,700 kJ/kWh)</td>
<td>48.5</td>
<td>60.1</td>
</tr>
<tr>
<td>Total: sailing + cool. down</td>
<td>51.7</td>
<td></td>
</tr>
<tr>
<td>Total (HFO)</td>
<td>51.4</td>
<td>63.7</td>
</tr>
<tr>
<td>Total: sail.+cool.down (HFO)</td>
<td>54.8</td>
<td></td>
</tr>
</tbody>
</table>

It can be noticed that there is a significant difference in fuel oil consumption between reefer vessel and container ship. Calculation of fuel oil consumption in the case of an assumed route of 5,000 nautical miles (days in navigation = 5,000 nm / 17.9 kn / 24 h/day ≈ 12 days) is as follows.

### Table 5 Total Fuel Oil Consumption of Fully Loaded Ships

<table>
<thead>
<tr>
<th>Duration</th>
<th>Daily fuel oil consumption (dfoc)</th>
<th>Total fuel oil consumption</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[days]</td>
<td>[t/day]</td>
</tr>
<tr>
<td>P105 (Reefer Vessel)</td>
<td>1</td>
<td>54.8</td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>51.4</td>
</tr>
<tr>
<td></td>
<td>~ 620</td>
<td></td>
</tr>
<tr>
<td>P109 (Container Ship)</td>
<td>12</td>
<td>63.7</td>
</tr>
</tbody>
</table>

Difference of total fuel consumption is abt. 144 t. If we assume a heavy fuel oil price of abt. 400-420 US$/t, saving of reefer vessel in comparison with container ship is abt. 60,000 US$ for 12 days sailing in a fully loaded condition.
6.2 Ships Sailing on Ballast Draught

Both ships are loaded to the ballast draught and are observed under the following conditions:
- Both ships are sailing on ballast draught that ensure good propeller immersion
- Ships are sailing at the same speed in the service conditions (15% sea margin)
- Load of diesel-generator plant is 650 kW

For the purpose to determine needed main engine power for both ships, speed-power estimations are carried out. Achievable speeds in both trial and service condition are shown in Figures 7 and 8.

Calculation of daily fuel oil consumption for both ships is carried out and shown in Table 6. It is based on the declared conditions and needed main engine power for service speed of 19.1 knots.

**Table 6** Daily Fuel Oil Consumption when Sailing in Ballast

<table>
<thead>
<tr>
<th>Consumer</th>
<th>P105 (Reefer Vessel)</th>
<th>P109 (Container Ship)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Power [kW]</td>
<td>sfoc [g/kWh]</td>
</tr>
<tr>
<td>Main engine</td>
<td>9,040</td>
<td>158.3</td>
</tr>
<tr>
<td>Diesel generator plant</td>
<td>650</td>
<td>192.0</td>
</tr>
<tr>
<td>Total (LCV=42,700 kJ/kWh)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total (HFO)11</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

It can be noticed that there is a significant difference in fuel oil consumption between reefer vessel and container ship. Calculation of fuel oil consumption in the case of an assumed route of 5,000 nautical miles (days in navigation = 5,000 nm / 19.1 kn / 24 h/day ≈ 11 days) is as follows.
Table 7  Total Fuel Oil Consumption when Sailing in Ballast

<table>
<thead>
<tr>
<th></th>
<th>Duration [days]</th>
<th>Daily fuel oil consumption (dfoc) [t/day]</th>
<th>Total fuel oil consumption [t]</th>
</tr>
</thead>
<tbody>
<tr>
<td>P105 (Reefer Vessel)</td>
<td>11</td>
<td>39.5</td>
<td>~ 435</td>
</tr>
<tr>
<td>P109 (Container Ship)</td>
<td>11</td>
<td>47.6</td>
<td>~ 524</td>
</tr>
</tbody>
</table>

Difference of total fuel consumption is abt. 89 t. If we assume a heavy fuel oil price of abt. 400-420 US$/t, saving of reefer vessel in comparison with container ship is abt. 36,000 US$ for 11 days sailing in ballast condition.

6.3  P105 (Reefer Vessel) Sailing at Continuous Service Rating

Reefer vessel can sail at a higher speed that corresponds to the main engine continuous service rating. In that case, calculation of fuel oil consumption and economy of the voyage differs from previously shown calculation. This case is presented in Table 8.

Table 8  P105 (Reefer Vessel) Daily Fuel Oil Consumption at CSR

<table>
<thead>
<tr>
<th>Consumer</th>
<th>Scantling draught</th>
<th>Ballast draught</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Power [kW]</td>
<td>sfoc [g/kWh]</td>
</tr>
<tr>
<td>Main engine</td>
<td>10,780</td>
<td>162.0</td>
</tr>
<tr>
<td>Diesel generator plant</td>
<td>3,036</td>
<td>192.0</td>
</tr>
<tr>
<td>DG plant during cool. down</td>
<td>3,736</td>
<td>192.0</td>
</tr>
<tr>
<td>Total (LCV=42,700 kJ/kWh)</td>
<td></td>
<td>55.9</td>
</tr>
<tr>
<td>Total: sailing + cool. down</td>
<td></td>
<td>59.1</td>
</tr>
<tr>
<td>Total (HFO)</td>
<td></td>
<td>59.3</td>
</tr>
<tr>
<td>Total: sail.+cool.down (HFO)</td>
<td></td>
<td>62.6</td>
</tr>
</tbody>
</table>

Calculation of fuel oil consumption in the case of an assumed route of 5,000 nautical miles (days in navigation on scantling draught = 5,000 n.m. / 18.7 kn / 24 h/day ≈ 11 days, on ballast draught = 5,000 nm / 19.97 kn / 24 h/day ≈ 10.5 days) is given in Table 9.

Table 9  Total Fuel Oil Consumption when Sailing at Continuous Service Rating

<table>
<thead>
<tr>
<th>P105 (Reefer Vessel)</th>
<th>Duration [days]</th>
<th>Daily fuel oil consumption (dfoc) [t/day]</th>
<th>Total fuel oil consumption [t]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scantling draught</td>
<td>1</td>
<td>62.6</td>
<td>62.6</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>59.3</td>
<td>593</td>
</tr>
<tr>
<td>Ballast draught</td>
<td>10.5</td>
<td>47.6</td>
<td>~ 500</td>
</tr>
</tbody>
</table>

In the fully loaded condition the reefer vessel consumes abt. 656 t of HFO (instead of 620 when sailing at a speed of 17.9 kn) and abt. 500 t of HFO (instead of 435 when sailing at a speed of 19.1 kn). In this case, still exists significant difference of fuel consumption with respect to the container ship, in fully loaded condition difference is 764 – 656 = 108 t of HFO and in ballast condition difference is 524 – 500 = 24 t of HFO. At assumed HFO price of abt. 400-420 US$/t, saving of reefer vessel in comparison with container ship is abt. 44,000 US$ for sailing in fully loaded condition and abt. 10,000 US$ in ballast condition.
7. Final Considerations

Presented comparison of reefer vessel and container ship dedicated for reefer containers shows that exists significant difference in fuel oil consumption and voyage economics between these two ships. Reefer ship has better speed-power characteristics and lower diesel generator fuel oil consumption in laden condition. The reasons lie in the following basic facts:

- Reefer vessel has significantly lower block coefficient and, consequently, better speed-power characteristics and lower main engine fuel oil consumption
- Large cargo refrigerating plant on reefer vessel has better efficiency than adequate number of small refrigerated FEU aggregates

Advantages of container ship with respect to reefer vessel are also significant and they can be summarized as follows:

- Easier and faster cargo loading and unloading
- Lower price of container ship in relation to reefer vessel
- Easier and less expensive maintenance of cargo holds

Reefer vessel is more energy efficient, it can develop higher speed, so the journey takes less time; on the contrary, cargo loading in the cargo holds takes more time, cargo holds insulation can be damaged during cargo loading/unloading, especially if it is raining.

Container ship can load and unload the cargo faster and easier, but it has lower energy efficiency and, as a rule, it sails at lower speed, so the journey takes more time. It should be kept in mind that the design of a container ship must be adapted to the transportation of refrigerated containers: cargo holds must be equipped with electrical sockets for refrigerated containers, cargo holds must be mechanically ventilated to evacuate the waste heat of refrigerated containers aggregates and diesel generator plant must cover total power consumption, including refrigerated containers and cargo holds ventilation. All this affects the economy of operation of the ship, so it is completely understandable that it is not simple nor easy to choose a better ship type.

When talking about the price of container ship, it should be kept in mind that container ship dedicated to refrigerated containers must be specially designed and is significantly more expensive than “ordinary” container ship (cargo holds mechanical ventilation, “ordinary” container ships do not need mechanical ventilation of cargo holds). Diesel generator plan is several times larger and more expensive than on the “ordinary” container ships to be able to drive refrigerated containers aggregates.

As a conclusion, it could be said that in the carriage of refrigerated cargos container ships may have the advantage when the whole process of cargo transportation is considered, but also, that the economy of reefer vessels’ operation is so attractive that further development and improvement of their design can be expected.
REFERENCES


Submitted: 3.09.2018. Predrag Ćudina, predrag.cudina@yahoo.com
Accepted: 5.02.2019. Ana Bezić, ana.bezic71@gmail.com

Classis Llc Ship Design and Consulting, Rendićeva 18, Split, Croatia
<table>
<thead>
<tr>
<th>Title</th>
<th>Page Range</th>
<th>Type</th>
<th>Authors</th>
<th>Journal</th>
</tr>
</thead>
<tbody>
<tr>
<td>NUMERICAL AND EXPERIMENTAL CALCULATION OF ROLL AMPLITUDE EFFECT ON ROLL DAMPING</td>
<td>str.1-15</td>
<td>Original scientific paper</td>
<td>Burak Yıldız, Bekir Şener, Ahmet Yurtseven, Toru Katayama</td>
<td></td>
</tr>
<tr>
<td>EXPERIMENTAL STUDY OF WELDING RESIDUAL STRESS OF HIGH-STRENGTH SHIPBUILDING STEEL</td>
<td>str.17-32</td>
<td>Professional paper</td>
<td>Yongjin Guo, Luyun Chen, Hongdong Wang, Hong Yi</td>
<td></td>
</tr>
<tr>
<td>WATER ENTRY HYDROELASTICITY ANALYSIS OF LATTICE SANDWICH PANEL WITH IMPERFECTION: SIMULATION AND ENGINEERING MODEL</td>
<td>str.33-59</td>
<td>Professional paper</td>
<td>Wang Hao, Cheng Yuan-Sheng, Pei Da-Ming, Hao Wei-Wei, Gan Lin</td>
<td></td>
</tr>
<tr>
<td>NUMERICAL SIMULATIONS OF ADDED RESISTANCE IN REGULAR HEAD WAVES ON A CONTAINER SHIP</td>
<td>str.61-86</td>
<td>Original scientific paper</td>
<td>Young-Gill Lee, Cheolho Kim, Jeong-Ho Park, Hyeongjun Kim, Insu Lee, Bongyong Jin</td>
<td></td>
</tr>
<tr>
<td>RETROFITTING ANALYSIS OF TANKER SHIP HULL STRUCTURE SUBJECTED TO CORROSION</td>
<td>str.87-109</td>
<td>Original scientific paper</td>
<td>Davide Chichi, Yordan Garbatov</td>
<td></td>
</tr>
<tr>
<td>DESIGN AND MODEL TEST OF STRUCTURAL MONITORING AND ASSESSMENT SYSTEM FOR TRIMARAN</td>
<td>str.111-134</td>
<td>Original scientific paper</td>
<td>Haoyun Tang, Huilong Ren, Qi Zhong</td>
<td></td>
</tr>
<tr>
<td>ANOTHER BLOW ON THE TORN DOWN WALL-THE INCLINING EXPERIMENT</td>
<td>str.135-153</td>
<td>Review article</td>
<td>Selahattin Ozsayan, Metin Taylan</td>
<td></td>
</tr>
<tr>
<td>PISTON RING MATERIAL IN A TWO-STROKE ENGINE WHICH SUSTAINS WEAR DUE TO CATALYST FINES</td>
<td>str.155-169</td>
<td>Original scientific paper</td>
<td>Miroslav Vukićević, Nikola Račić, Špiro Ivošević</td>
<td></td>
</tr>
<tr>
<td>A LID APPROACH FOR PREDICTING WAVE INDUCED MOTIONS OF TRIMARAN IN REGULAR WAVES</td>
<td>str.171-185</td>
<td>Professional paper</td>
<td>Zongyu Jiang, Yun Gao, Jie Liu</td>
<td></td>
</tr>
</tbody>
</table>
Roll motion is still a challenging problem in naval architecture and an adequate prediction of this physical phenomenon is important because of its undesirable effects such as capsizing. There are several methods using linear potential theory to predict roll motion, such as strip method, however, the accuracy of the calculated results lag behind the accuracy of other degrees of freedom due to viscosity. Viscosity have an important effect on roll damping, especially near resonance, and as it is known, it is not included in potential flow methods. Vortex shedding is the main physical phenomena in viscous damping of the roll motion and it affects the flow velocity around the bilge. This may lead to pressure increase or decrease on the hull. In the present study, roll damping of a forced rolling hull with bilge keels at different roll amplitudes was calculated numerically by using an Unsteady Reynolds-averaged Navier–Stokes (URANS) solver. For the purpose of validation, forced roll experiments were carried out and the results were plotted next to numerical results. The generated vortices around the hull and bilge keel were observed in the URANS calculations. In the case of large roll amplitude motion, the vortex shedding from the bilge keel interacts with the free surface and leads to decrease on roll damping.

**Key words:** Roll damping; bilge keel; large amplitude, vortex shedding, computational fluid dynamics

1. **Introduction**

Roll motion is the oscillatory motion of a vessel along its longitudinal axis. As the transverse loads are governed by roll motion, large-amplitude roll motions compromise the stability of a floating body and also affects structural integrity, crew and passenger comfort, machinery operations and ultimately the safety of a vessel. If the roll motion can be predicted at an early design stages, the vessel can be designed to withstand with the imposed dynamic motions in response to the environmental loads. However, prediction of roll damping has always been a challenging task for naval architects. As indicated above, accurate estimation of the roll motion is of great importance; underestimation can potentially cause disasters while
its overestimation results in a commercially unviable design that leads am increase of the weight and limits operability of the floating body.

Ship motions are mainly damped by the generation of surface waves which radiate from the ship in most cases. Hence the viscous damping components can usually be ignored and potential flow methods can be used to sufficiently simulate the motions [1]. However, this is not valid for roll motion. Linear radiation damping is the part of the roll damping caused by radiated surface waves and can be calculated using linear potential theory. However, viscous effects like flow separation, vortex shedding cannot be computed in potential flow methods.

In addition, ship appendages can have a noticeable effect on roll damping. Especially bilge keels are the most simple and effective appendages creating additional damping as a result of flow separation from their edges. Bilge keels provide a vortex around the body which increases the viscous contribution to the total damping. The generated vortices mitigate the roll motion by transferring energy from the ship to the surrounding fluid.

Many researchers have studied the viscous roll damping prediction, e.g. Ikeda et al. [2, 3, 4] and Himeno [5]. The recommended procedures for roll damping estimation are empirical, based on model tests, full-scale tests or from a database developed from model tests. Considering the relative simplicity of these methods, they perform reasonably well in most cases especially for conventional hull shapes and bilge keel sizes. However, Ikeda’s method is typically valid only for small roll motions which were only performed for roll amplitudes up to 15 deg and later extended to 20 deg. Several improvements of the method have been proposed in recent years, both to make it simpler [6] and to add the capability for handling special cases such as shallow drafts [7] and high-speed vessels [8, 9]. In the present paper, the effects of coming close the bilge keel to the free surface have been also investigated which is usually neglected.

Model experiments are the most common and accurate way for roll damping calculations. However, they are time-consuming and expensive to investigate lots of different options, such as the effects of different bilge keel shapes and positions or different locations of center of gravity on roll damping. Alternatively, in recent years, computational fluid dynamics (CFD) tools have been applied for solving the ship roll motion extensively. Since one of the major component of roll damping comes from the viscous and eddy-making effects of the flow, CFD based Unsteady RANS solvers have the potential to produce superior roll damping predictions compared to existing methods since the effects due to viscosity, creation of vorticity in the boundary layer, vortex shedding and turbulence are naturally included in the calculations. The advantages, such as low cost and fast computational time compared to experiments, lead researchers to use CFD for the estimation of roll damping.

URANS-equation methods have been used primarily by Yeung and Ananthakrishnan [10] to solve the nonlinear viscous-flow problem associated with heaving motion of a two-dimensional floating cylinder. Their work has led to the use of URANS-equation methods in the analysis of roll motion; several researchers used URANS solvers to study the flow around two-dimensional oscillating cylinders [11, 12, 13]. Bassler [14] realized forced roll test and CFD simulations to investigate the effects of hull and bilge keel geometry on damping of large amplitude roll motion. Avalos et al. [15] carried out CFD simulations with their own two-dimensional Navier-Stokes solver for free roll decay of FPSO. They performed simulations with and without bilge keels and compared results with the experiments carried out by Oliviera and Fernandes [16]. They presented that vortex size and roll damping depends on the amplitude of roll motion and the width of bilge keel. By using a commercial CFD code, Van Kampen [17] revealed a practical method to evaluate roll damping and motions of an FPSO. He conducted simulations with different type bilge keels and riser balconies in waves and used results to modify traditional Ikeda’s method. Irkal et al. [18] investigated the
best configuration of a bilge keel to reduce roll motion with numerical simulations using a commercial RANSE solver. As a result of the simulations, they obtained velocity and vorticity distributions around the bilge keel alternatives and validated with PIV measurements. The effect of the shallow draft on roll damping was studied by Yıldız et al. [19] by using URANS method. They performed two-dimensional roll damping calculations for a forced rolling S60 midsection with bilge keels and validated the results with experiments. They presented that Ikeda’s method overestimates at shallow draft conditions due to not considering the interaction between free surface and bilge keel.

Ommani et al. [20] performed a two-dimensional numerical study on a horizontal, circular cylinder fitted with one bilge keel forced to rotate harmonically around its axis. They used a Navier-Stokes solver based on the Finite Volume Method and simplified the problem by neglecting the influence of nonlinear advection terms and viscous stresses in the free-surface zone. Three different drafts and one fully submerged case far away from the free surface were considered in their work and three different bilge keel shape were used. They validated the numerical results with experimental data and presented that the simplified modeling and relatively coarse mesh could result in accurate prediction both in terms of forces on the bilge keel and the pressure distribution on the hull surface. Yıldız and Katayama [21] also used a RANS solver to carry out CFD simulations of roll damping for a two-dimensional model of a ship with bilge keels at the various draft. They performed experiments to validate the numerical calculations and investigated the generated vortices and vortex shedding numerically at deep and shallow draft conditions. Ikeda’s method gives a constant value for each draft condition, however they observed that the core and strength of the vortex decrease when the bilge keel come close to the free surface at the shallow draft.

Although there have been many studies on roll damping estimation by using experiments or CFD methods, there is still a critical need for the development of methods for predicting large amplitude roll damping of ships with appendages. In the present study, the effect of roll amplitude on roll damping was investigated numerically and experimentally. The forced roll simulations were carried out to calculate the roll damping moment. The vorticity generation around the hull was visualized by using numerical solver. The effect of vortex shedding and free surface interaction were investigated at different roll amplitudes. It was observed that the roll damping decreased when the bilge keels come close to the free surface at large roll amplitudes. Numerical results showed good accuracy with the experimental results. The numerical and experimental uncertainty analysis have also been performed to endorse the accuracy of the results.

### 2. Determination of the roll damping coefficient

Roll damping is obtained by analysis of the measured roll moment and roll motion from experimental and numerical forced roll test. The roll angle $\phi(t)$ is forced by:

$$\phi(t) = f(t)\phi_0 \sin \omega t$$  \hspace{1cm} (1)

The equation of one-degree-of-freedom (1-DoF) forced roll motion is given in Eq. 2.

$$(I_{\phi\phi} + \alpha_{\phi\phi})\phi'' + B(\phi, \phi') + C(\phi) = M_E(t)$$  \hspace{1cm} (2)

where $I_{\phi\phi}$ is mass moment of inertia, $\alpha_{\phi\phi}$ is the added mass moment of inertia for roll motion, $B(\phi, \phi')$ is the damping moment, $C(\phi)$ is the restoring moment and $M_E(t)$ is the time history of the measured or computed moments and it is fitted by Eq. 3;

$$M_E(t) = M_c \sin(\omega t + \delta)$$  \hspace{1cm} (3)
by applying the Fourier analysis which is used as an equivalent linearization method for energy, $M_0$ is the roll amplitude and $\phi$ indicates the phase angle between the roll angle and the roll moment. Time history of the moments is obtained by experiments or CFD simulations, then $M_0$ and $\phi$ can be calculated by applying Fourier analysis to the time history of moments. The roll damping coefficient can be expressed as follow:

$$B_{++} = \frac{M_0 \sin(\phi)}{\rho g \omega}$$  \hspace{1cm} (4)

Finally, non-dimensional representations of the damping coefficient is found as follow:

$$\tilde{B}_{++} = \frac{B_{++}}{\rho V^2 WL} \frac{\rho V^2 WL}{g}$$  \hspace{1cm} (5)

2.1 Experimental investigation

The forced roll tests were carried out to obtain the roll damping coefficients experimentally. A two-dimensional (2D) midship section model with attached bilge keels was used for calculations. Table 1 shows the principal particulars of the model with bilge keels.

<table>
<thead>
<tr>
<th>Table 1 Principle particulars of 2D model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, L (m)</td>
</tr>
<tr>
<td>Breadth, B (m)</td>
</tr>
<tr>
<td>Depth, D (m)</td>
</tr>
<tr>
<td>Center of gravity, KG (m)</td>
</tr>
<tr>
<td>Block coefficient, C_B</td>
</tr>
<tr>
<td>Bilge radius</td>
</tr>
<tr>
<td>Length x Breadth (Bilge keel) (m)</td>
</tr>
</tbody>
</table>

During the forced roll tests, the model was only allowed to roll motion and the other motions were restricted. The sinusoidal 1-DoF forced roll oscillations were applied to the 2D model and the roll moments were measured. Fig. 1 shows the 2D model attached to the forced roll mechanism.
The single-degree-of-freedom strain gage sensors were embedded into the arms and used to obtain the moment measurements (Fig. 2).

![Fig. 2 Schematic view of forced roll mechanism](image)

The experiments were conducted with combinations of roll amplitudes to show the effect of roll amplitude on roll damping (Table 2). The roll moments were measured for different roll amplitudes and the roll damping coefficients were calculated.

<table>
<thead>
<tr>
<th>Test conditions</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Fn</strong></td>
<td>0.0</td>
</tr>
<tr>
<td><strong>Roll amplitude, Ø (deg)</strong></td>
<td>8.59, 11.38, 14.38, 20.0, 27.27</td>
</tr>
<tr>
<td><strong>Roll period, T (s)</strong></td>
<td>1.2</td>
</tr>
<tr>
<td><strong>Draft, d (m)</strong></td>
<td>0.042</td>
</tr>
</tbody>
</table>

The uncertainty of the test device was checked and the results were published in the previous paper by the authors. The detailed information about the uncertainty analysis can be found in [21].

2.2 CFD simulations

2.2.1 Numerical settings

The flow around the oscillating body was solved numerically by a commercial URANS solver. The numerical 2D model was forced to the sinusoidal roll motion at a fixed axis as it was applied in the experiments. The generated numerical geometry and mesh around the 2D hull is shown in Fig. 3. The mesh size around the bilge keels should be small enough to capture the generated vortices accurately as it is shown in Fig. 4. The volume of fluid (VOF) method was used to solve the multiphase flows with the free surface at the air-water interface. The k-ε turbulence model was selected for the calculations. The top boundary of the computational domain was defined as open to atmospheric pressure and the bottom boundary was selected as wall, both ends were defined as outflow boundary. The roll moments were calculated at the same conditions with experiments and the roll damping coefficients were obtained numerically.
2.2.2 Numerical verification study

The verification study has been made via Grid Convergence Method (GCI). This method, which depends on Richardson Extrapolation, was introduced by Roache [22] and then it was used in numerous studies. In the present study, the procedure developed by Celik et al. [23] was used for verification study. Grid spacing and time step size, two of the main sources of uncertainty, were selected for verification study and the other sources were neglected. The time step size and grid resolution were refined systematically with constant refinement factor $\sqrt{2}$ as usually selected for verification. Three mesh cases; coarse (a) 3.00 million of cells, medium (b) 4.24 million of cells, and fine mesh (c) 6.00 million of cells, are shown in Fig. 5. It should be noted that in grid spacing verification study, the refinements were implemented only the overset region. The time-step convergence study was conducted on the finest grid (Case C) by systematically refined time steps with refinement ratio equal to $\sqrt{2}$, i.e. 0.005 s, 0.007 s and 0.01 s. It should also be noted that in the grid spacing verification study time-step was selected as medium time step which equals to 0.007 s. However, in the time step convergence study, the grid spacing was selected fine grid.

For verification of each source (time step & grid spacing), three solutions were considered and only one source of uncertainty was changed.
Numerical and experimental calculation of roll amplitude effect on roll damping

For instance, the time step size was kept constant while grid spacing was being refined systematically. The methodology applied in this paper can be briefly expressed as follows [23]. The difference between the solution scalars (ε) should be determined by Equation (8):

\[ ε_{21} = φ_2 - φ_1 \quad ε_{32} = φ_3 - φ_2 \]  

In these equations, \( φ_1, φ_2 \) and \( φ_3 \) refer solution of course, medium and fine mesh grid or time-step size, respectively. Convergence condition of the numerical study can be calculated by Equation (9):

\[ R = \frac{ε_{21}}{ε_{32}} \]  

The convergence condition has to be between 0 and 1 to monotonic convergence [24]. The apparent order of p can be found by Equation (10) [23]:

\[ p = -\frac{\ln|ε_{32} / ε_{21}| + q}{\ln(r_{s})} \]  

Here,

\[ q = \ln \left( \frac{r_{21} - s}{r_{32} - s} \right) \]  
\[ s = \text{sgn}(ε_{32} / ε_{21}) \]  

The extrapolated value is:

\[ φ^{21}_{ext} = \left( r^p φ_1 - φ_2 \right) / \left( r^p - 1 \right) \]  

The approximate relative error and extrapolated relative error are:

\[ ε^{21}_a = \frac{φ_2 - φ_1}{φ_1} \quad ε^{21}_e = \frac{φ^{12}_{ext} - φ_1}{φ^{12}_{ext}} \]
Finally, the GCI index is calculated by:

\[
\text{GCI}_{\text{fine}}^{21} = \frac{1.25a^{21}}{r_1^{21} - 1}
\]  

(15)

<table>
<thead>
<tr>
<th></th>
<th>Grid Convergence</th>
<th>Time Step Convergence</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\phi_1$</td>
<td>0.03777</td>
<td>0.05267</td>
</tr>
<tr>
<td>$\phi_2$</td>
<td>0.03919</td>
<td>0.04056</td>
</tr>
<tr>
<td>$\phi_3$</td>
<td>0.03952</td>
<td>0.03952</td>
</tr>
<tr>
<td>R</td>
<td>0.185</td>
<td>0.079</td>
</tr>
<tr>
<td>$%GCI_{\text{FINE}}$</td>
<td>0.232</td>
<td>0.275</td>
</tr>
</tbody>
</table>

The uncertainty values of the roll damping coefficients are given in Table 3. The convergence was achieved at both grid and time-step uncertainty and percentages are at applicable levels for both cases.

3. Results and discussion

3.1 Roll moments and roll damping coefficients

CFD computations have been carried out for the hull at five different roll amplitude values and results have been compared with experimental results. The moments acting on the hull and bilge keels were computed separately while the hull was forced to the roll motion. Fig. 6 and 7 show the total (hull + bilge keel) moments and bilge keel moments obtained from numerical analyses at different roll amplitudes. As it is shown on the Fig.6, the total moment increases when the roll amplitude increases. However, the bilge keel moment decreases at the point where the bilge keel interacts with the free surface. This effect cannot be seen in the total moment figure because the damping moment is a small portion of total moment. The effect of free surface interaction can be seen in Fig. 7. The bilge keel moment increases until 20 deg. but decreases at 27.27 deg. where bilge keels come closer to the free surface.
The roll damping coefficients have been calculated numerically and were obtained experimentally by using moments acting on the hull and bilge keels. Fig. 8 shows numerical and experimental non-dimensional roll damping coefficients at different roll angles. A good agreement between numerical results and experiments at each roll amplitude was observed. The roll damping increased from small to moderate roll amplitudes and it decreased at the largest amplitude. When the roll amplitude increases, the bilge keel come close to the free surface and this cause the vorticity around the bilge keels vortex shedding which affect the force acting on the bilge keels and the roll damping.

The bilge keels provide the largest contribution to energy dissipation [5]. However, bilge keels are more effective when they are fully submerged, typically only for small and moderate roll motion, and at lower speeds [4, 5]. According to the results of numerical...
analyses, as shown in Fig. 9, the bilge keel roll damping shows the same tendency as the total roll damping to decrease at large amplitudes that the bilge keel come close to the free surface. Also, roll damping contribution to the total roll damping decreases when the bilge keel comes closer to the free surface. Fig.10 shows the percentage of the bilge keel damping on total roll damping obtained from numerical results at different roll amplitudes. The dark parts show the bilge keel damping contribution. At the large roll amplitude, the bilge keel contribution is less as it was mentioned by Ikeda [4]. In the previous paper [25], authors showed the comparison of roll damping coefficients calculated with numerical solver and Ikeda’s estimation method. Ikeda’s method overestimated the roll damping value at large roll amplitudes because the method does not account for the free surface effect.

Fig. 9 Numerical bilge-keel damping and total roll damping coefficients at different roll angles

Fig. 10 Percentage of the numerical bilge-keel damping on the total roll damping
3.2 Generation of the vortices around the hull with bilge keels

The generated vortices from the bilge keel causes to changes in the bilge keel force and so the roll damping. Vortex shedding is the main physical phenomena involved in the viscous damping and it affects the flow velocity around the body that may lead to changes in the pressure gradients on the hull. To investigate the effect of the roll amplitude on the roll damping, the generation of vortices near the bilge-keels were investigated numerically for different roll amplitudes. The blue colour in the figures denotes negative (clockwise) vorticity while the red colour denotes positive (counter-clockwise) vorticity and the vorticity (1/s) scale is same for all figures, ranging from -50 to 50. Fig. 11 shows the generated vortices around the hull and bilge keels at a 8.59 deg roll amplitude and a 20.0 deg amplitude. As shown, the size and core of the vorticity increase with increasing the roll amplitude. This explains how the roll damping increases when the roll amplitude increases. However, the roll damping decreases when the roll amplitude is 27.27 deg. As mentioned before, the bilge keel comes closer to the free surface at this position. The vorticity generation around the hull were compared for different roll amplitudes to investigate the vortex shedding and free surface interaction on roll damping.

![Fig. 11 Generation of vortices around hull at max roll speed (left: 8.59 deg, right: 20.0 deg)](image)

Vorticities around the bilge keel are shown in Fig. 12 a and b, for roll amplitudes of 20 deg and 27.27 deg, respectively. The vortices are shown for the half oscillation period for both conditions. As it can be seen in the Fig. 12, the vortices are strongest when the hull reaches its maximum roll velocity. The generated positive vortices start to shed while the hull is rolling. At the maximum roll amplitude, where the roll speed is zero, the positive shed vortices start to dissipate after the hull reaches the maximum roll amplitude. At this point, roll direction changes and negative vortex starts to occur from the tip of the bilge keels and grows gradually. The previously generated positive vortex interacts with the new generated positive vortex and dissipates into the surrounding fluid while the body reaches the maximum roll velocity. The more intense negative vortex is dragging the positive vortex that is less intense, as it has been also shown in Avalos et al. [15]. As the hull rolls to a maximum amplitude, the newly generated negative vortex starts to shed from the bilge keel. When the hull reverses its direction, a new positive vortex will start to occur from the tip of the bilge keels. As the roll motion progresses in time, a new vortex will be generated every half of an oscillation and a new cycle of vortex shedding will start.
Fig. 12 Generation of vortices and vortex shedding around the hull form for (a) 20.0 deg. (b) 27.27 deg. roll amplitude

Fig. 12b. shows the vorticity contours around the bilge keels for 27.27 deg roll amplitude where the bilge keel interacts with the free surface. As it is shown in the figure, the vortices start to shed earlier. After the hull reaches the maximum roll amplitude, the negative vortex starts to occur at the same point as for the 20 deg roll amplitude case. However, the generation of the vortexes is affected by the presence of the free surface and the vortex starts to shed just after the maximum roll amplitude. This interaction with the free surface does not allow the vortices to grow. It can be seen that the size of the vortex for the 20 deg amplitude motion is bigger than the large roll amplitude condition. This explains the decrease of bilge
keel moment at large roll amplitude. The damping from the bilge keel decreases when the vortices become weaker. It has been also observed that the free surface disturbance is stronger for the 27.27 deg amplitude motions as Himeno [5] cautioned that the bilge keel wave-making component cannot be neglected where bilge keel interacts with the free surface. It might be said that the wave-making damping increases when the bilge-keel component decreases at large roll amplitudes. Wave-making damping can be calculated by using the radiated wave amplitudes but it will be studied as a future work because the mesh around the free surface needs high quality to measure wave heights.

4. Conclusion

In this paper, a URANS numerical solver was used for the estimation of the effect of the roll amplitude on the roll damping. The unsteady flow around a forced rolling hull with bilge keels was simulated and the viscous-damping coefficient was computed for various roll amplitudes. The 1-DoF forced roll tests were also carried out to validate the numerical results. The generation of vortices around the hull during roll motion was visualized with numerical solver to show the interaction of the bilge-keel and free surface at large roll amplitudes. The URANS calculations and the experiments were done to validate the calculations. For the numerical uncertainties, an acceptable comparison error was calculated for the grid refinement and time step.

Numerical results showed good agreement at each roll angles with experimental results and several important observations were obtained from the numerical and experimental analysis. It has been shown that the total roll moment and roll moment due to the bilge keels are amplitude dependent. The total roll moment increased for each roll amplitude but the bilge keel roll moment decreased when the bilge keels approach the water/air interface due to the free surface interaction. The roll damping coefficients were obtained from calculated and measured roll moments. The total roll damping increased from small to moderate roll angles and it decreased at large roll amplitudes. The bilge keel interacted with the free surface at large roll amplitude and this interaction caused the decrease of the bilge keel damping contribution to the total roll damping. For small and moderate roll amplitudes, the relative contribution of the bilge keel to the total roll moment was over 60%. However, near and above the free surface threshold, the bilge keel contribution to the total unit started to decrease due to the interaction of bilge-keel and free surface. Numerical solver was used to capturing the effect of vortex shedding and free surface interaction on the roll damping due to the large roll amplitudes. The flow around the hull with bilge keels was visualized and the generation of vortices was shown for different roll amplitudes and it was shown that vortices sheds from the bilge keel are proportional to the amplitude of roll motion. It was observed that the strength and core of the vortices grow until the free surface interaction starts which leads to increase of roll damping. The roll damping coefficient starts to decrease when the bilge keel come closer to the free surface because vortex generation and shedding is affected by the presence of the free surface.
REFERENCES


Submitted: 27.05.2018. 
Burak Yıldız, buraky@yildiz.edu.tr
Yildiz Technical University, Naval Architecture and Maritime Faculty, 34349, Besiktas, Istanbul, Turkey.
Bekir Şener (*corresponding author), bsener@yildiz.edu.tr
Yildiz Technical University, Naval Architecture and Maritime Faculty, 34349, Besiktas, Istanbul, Turkey.
Phone: +90 212 3832852, Fax: +90 212 3833021
Ahmet Yurtseven, ahmety@yildiz.edu.tr
Yildiz Technical University, Naval Architecture and Maritime Faculty, 34349, Besiktas, Istanbul, Turkey.
Toru Katayama, katayama@marine.osakafu-u.ac.jp
Osaka Prefecture University, Graduate School of Engineering, Japan.
EXPERIMENTAL STUDY OF WELDING RESIDUAL STRESS OF HIGH-STRENGTH SHIPBUILDING STEEL

UDC 691.714:52-334.2
Professional paper

Summary

Ships are large welded structures. Welding residual stress (WRS) significantly impacts the mechanical properties of the structure. The plate and cylinder models welded by electrode welding (EW) and submerged arc welding (SAW) were designed to measure the WRS of high-strength shipbuilding steel. The WRS was tested by instrumented indentation method. The stress value of each measure point was proposed. The stress distribution patterns of different welding processes and models were analysed. This research can provide a reference for controlling the WRS of ships and improving shipbuilding quality.

Key words: high-strength shipbuilding steel; welding residual stress; instrument indentation test

1. Introduction

High-strength steel has favourable mechanical properties and is widely used in manufacturing ship structures [1, 2]. Under the same structural strength, using high-strength steel can reduce hull weight, increase carrying capacity and reduce operating cost slightly. Many structural parts of ships are welded by high-strength steel plates. Uneven heating during the welding process will cause local plastic deformation of the weldment, thereby resulting in WRS in the structure, which poses a potential threat to the safety of the structure and adversely affects the fabrication, performance, and service life of ships [3–7]. High-strength steel material is added to alloying elements on the basis of ordinary-strength steel, and its welding performance is different from that of ordinary-strength steel. Therefore, the WRS of high-strength shipbuilding steel must be explored to improve the construction quality and performance of ships.

Many studies have been conducted on the numerical simulation and measurement of WRS. Perić et al. [8] investigated the welding distortion in a T-joint fillet weld using an experimental method and finite element simulations, thereby verifying that 3D technology is an effective method for predicting welding distortions and residual stresses for large structures. The features of the WRS distribution near a weld end-start location have been
investigated through experiment and numerical simulation [9]. Deng and Murakawa [10] used large and small deformation theories to simulate WRS and deformation in a butt-welded joint. Deng et al. [11] used numerical simulation technology and experimental method to investigate the characteristics of WRS distribution in several typical welded joints, which are used in nuclear power plants. The WRS of the lap joints of thin steel sheets was measured by X-ray diffraction method and compared with the numerical simulation, thus confirming that the WRSs in the arc-welded lap joints are significantly influenced by the strength of the base metal and welding heat input [12]. Zhao et al. [13] applied finite element method to calculate the WRS for Invar steel of an LNG carrier containment system, whereas the x-ray diffraction method was applied to measure the longitudinal WRS of the welding joint. The research concluded that the effect of WRS cannot be neglected when the fatigue life of the LNG carrier containment system is assessed under a sloshing impact.

Several researchers have studied the influence of different welding processes on WRS. An X-ray diffraction technique was used to measure the surface residual stress distribution of high-strength steel by hybrid laser/gas metal arc welding in a butt joint configuration. The influence of welding parameters, including welding speed and wire feeding rate, on the residual stress distribution and mechanical properties of a hybrid weld has been experimentally studied [14]. Chen et al. [15] simulated six welding sequences to study their influence on the WRS of a stiffened plate structure. Kong and Kovacevic [16] confirmed that the residual stress concentration in the welded joint obtained by a hybrid laser–gas tungsten arc welding is smaller than when using laser welding alone. The surface residual stress evaluation for double-electrode welding has been studied [17]. The residual stress generated by the double-electrode welding is generally higher than that by single-electrode welding. Fu et al. [18] investigated the WRS and distortion in T-joint welds under various mechanical boundary conditions. The effects of preheating, interpass and transient-releasing temperatures on the residual stresses in the multipass welding of high-strength shipbuilding steel are investigated [19].

In general, WRS is a research hotspot in the field of materials and manufacturing. However, currently there are few measuring data for WRS of high-strength shipbuilding steel under the actual welding process. In order to solve this problem, we create the plate and cylinder models under the processes of EW and automatic SAW to measure the WRS distributions by the instrumented indentation method. The measuring data may contribute to conduct the analysis of mechanical properties of ship structures and quantitative calculation of WRS in shipbuilding. Meanwhile, this study can provide a reference for controlling the WRS of ships and improving the shipbuilding quality.

2. Measurement method of the WRS

In this research, instrument indentation method was adopted to measure the WRS. In Fig. 1, the principle of this method is that the indentation force required to reach a given depth \( h_m \) under residual stress is different from that in the stress-free state [20]. The slope of force-depth curve is changed by tensile residual stress \( F_T \) or compressive residual stress \( F_C \) in a target test piece.
Experimental Study of Welding Residual Stress of High-strength Shipbuilding Steel

Yongjin Guo, Luyun Chen
Hongdong Wang, Hong Yi

From the stressed and stress-free force-depth curves, the force difference between two curves can be calculated at a given indentation depth. The force difference can be correlated to residual stress by considering the stress state under the indenter tip. Only surface biaxial stresses can affect the shape of the force-depth curve [21, 22]. Biaxial stresses can be divided into mean stress (hydrostatic stress) and plastic-deformation-sensitive shear deviator stress terms described in Eq. (1) [23, 24].

\[
\begin{bmatrix}
\sigma_{res} & 0 & 0 \\
0 & p\sigma_{res} & 0 \\
0 & 0 & 0
\end{bmatrix} = \begin{bmatrix}
\frac{(1+p)}{3} & \sigma_{res} & 0 \\
0 & \frac{(1+p)}{3} \sigma_{res} & 0 \\
0 & 0 & \frac{(1+p)}{3} \sigma_{res}
\end{bmatrix} + \begin{bmatrix}
\frac{(2-p)}{3} & \sigma_{res} & 0 \\
0 & \frac{(2-p)}{3} \sigma_{res} & 0 \\
0 & 0 & \frac{(2-p)}{3} \sigma_{res}
\end{bmatrix}
\]

where \( \sigma_{res} \) is the residual stress, and \( p \) is the ratio of residual stresses along one direction to those along the normal direction, which can be determined by the indentation tests performed with a Knoop indenter. For welded steel joints, the value of \( p \) is recommended as 1/3.

The indentation test direction is defined as \( z \), Only the shear deviatoric stress component applied along the indentation test direction \( (\sigma_{z,d}) \) can influence the force-depth curve, and \( \sigma_{z,d} \) can be related to the force difference:

\[
\sigma_{z,d} = \frac{(1+p)}{3} \sigma_{res} = \frac{1}{\psi} \frac{(F - F_T)}{A}
\]

\[
\sigma_{z,d} = \frac{(1+p)}{3} \sigma_{res} = \frac{1}{\psi} \frac{(F - F_C)}{A}
\]

where \( \sigma_{z,d} \) is the shear deviatoric stress component applied along the indentation test direction, \( F \) is the test force, \( \psi \) is the ratio of average contact pressure to true stress and \( A \) is the contact area including residual stress.

From the relationship in which \( \sigma_{res} \) is proportional to the indentation force difference, the residual stress for tensile and compressive residual stresses can be expressed as follows:

\[
\sigma_{res} = \frac{1}{\psi} \frac{3}{(1+p)} \frac{(F - F_T)}{A}
\]
\[ \sigma_{res} = \frac{1}{\psi} \frac{3}{(1+p)} \frac{(F - F_c)}{A} \] (5)

The measurement method is easy to execute and has high measurement accuracy and speed. This method has reached a micro destructive level and can test the residual stress of a large gradient change. In addition, the test equipment is convenient and suitable for a shipyard, which has a high application value in shipbuilding.

3. Experimental scheme

3.1 Experimental material and welding process

The material of the base metal is high-strength low-alloy 10CrNiMoV steel, which is a reliable material for the key components in shipbuilding due to its superior mechanical properties and good weldability. The chemical composition and mechanical properties of the material are summarised in Tables 1 and 2.

<table>
<thead>
<tr>
<th>Table 1 Chemical composition of the base steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
</tr>
<tr>
<td>10CrNiMoV</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 2 Mechanical properties of the base steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thickness</td>
</tr>
<tr>
<td>14 mm</td>
</tr>
</tbody>
</table>

The low carbon content of the steel material ensures favourable weldability. The addition of Ni increases the toughness and plasticity of the steel and reduces the brittle transition temperature. The hardenability of steel is improved by adding appropriate Cr and Ni elements. Other alloy elements, such as Mn, Mo and V, ensure the favourable comprehensive mechanical properties of the material. The welding processes used in the experiment are EW and SAW. The automation degree is higher in SAW than in EW. SAW has a better welding quality and repeatability than EW. The main drawback of the SAW is that the parts of complex products cannot be easily welded because the welding head is difficult to retract in place. The EW equipment is low cost with favourable flexibility and conveniences. However, this equipment requires a high-operation technique for welders, and the quality stability of the weld is inferior. These welding methods are commonly used in the welding of ship structures.

The experimental models were welded in accordance with the specific welding process proposed by Zhao [25]. The double-v groove was created. The welding current of the SAW is 500 A, with the welding speed of 35 cm/min. The welding current of EW is 150 A. Table 3 lists the chemical composition of the welding rods.

<table>
<thead>
<tr>
<th>Table 3 Chemical composition of the welding rods</th>
</tr>
</thead>
<tbody>
<tr>
<td>Welding process</td>
</tr>
<tr>
<td>C</td>
</tr>
<tr>
<td>Si</td>
</tr>
<tr>
<td>Mn</td>
</tr>
<tr>
<td>Cr</td>
</tr>
<tr>
<td>Mo</td>
</tr>
<tr>
<td>Ni</td>
</tr>
<tr>
<td>V</td>
</tr>
<tr>
<td>Ti</td>
</tr>
<tr>
<td>S</td>
</tr>
<tr>
<td>P</td>
</tr>
</tbody>
</table>
3.2 Experimental model

(1) Model 1 is a steel plate that was welded by two identical plates using EW. As shown in Fig. 2, the length, width and thickness of the model are 1200, 800 and 14 mm, correspondingly. The weld bead is at the centre of the long side. Three measuring lines, labelled with Lines A, B and C, were designed on the top side of the plate. Three measuring lines, labelled with Lines A’, B’ and C’, were designed on the backside of the plate. Measuring lines A and A’ have 20 measuring points, and the other lines have 10 measuring points.

Fig. 2 Measurement model of a steel plate (a). Arrangement of measuring points (b).

(2) Model 2 is a steel plate that was welded by two identical plates using the SAW. Its geometric dimensions and the location of weld bead are the same as those of Model 1. Three measuring lines, labelled with Lines D, E and F, were designed on the top side of the plate. Three measuring lines, labelled with Lines D’, E’ and F’, were designed on the backside of the plate. Measuring lines D and D’ have 20 measuring points, and the other lines have 10 measuring points.

(3) Model 3 is a steel cylinder with length, inner diameter and outer diameter of 200, 336 and 350 mm. The centre of the model has two weld beads. The circumferential weld bead was welded by EW, and the axial one was welded by the SAW. Three measuring lines, labelled with Lines G, H and I, were designed for the axial weld. Line G has 20 measuring points, and the H and I have 10 measuring points. Two measuring lines were set for circumferential weld with Measuring lines J and K. Each line has 20 measuring points. The size and point layout of the model are illustrated in Fig. 3.
3.3 Experimental procedure

According to the instrumented indentation measurement scheme, the experimental procedure is presented as follows:

− Selection of measuring area
− Selection of the base material area
− Grinding of the measuring area
− Arrangement and marking of measuring points
− Installation and fixing of indentation testers
− Indentation measurement of the base material using a spherical indenter
− Indentation measurement of the measurement area using a spherical indenter.
− Indentation measurement of the measurement area using Vickers indenter
− Data processing and analysis

4. Experimental scheme

The results of Models 1, 2 and 3 are listed in Tables 4, 5 and 6, respectively.
### Table 4 WRS of the plate by EW

<table>
<thead>
<tr>
<th>Distance from centerline / mm</th>
<th>Line A</th>
<th>Line A’</th>
<th>Line B</th>
<th>Line B’</th>
<th>Line C</th>
<th>Line C’</th>
</tr>
</thead>
<tbody>
<tr>
<td>-77.5</td>
<td>119.46</td>
<td>-112.17</td>
<td>240.59</td>
<td>86.81</td>
<td>168.82</td>
<td>29.92</td>
</tr>
<tr>
<td>-57.5</td>
<td>103.11</td>
<td>-346.43</td>
<td>-177.31</td>
<td>-15.21</td>
<td>216.99</td>
<td>-41.71</td>
</tr>
<tr>
<td>-42.5</td>
<td>105.68</td>
<td>-249.56</td>
<td>-56.47</td>
<td>112.56</td>
<td>63.56</td>
<td>13.59</td>
</tr>
<tr>
<td>-37.5</td>
<td>3.75</td>
<td>-25.04</td>
<td>43.32</td>
<td>-0.09</td>
<td>78.63</td>
<td>-6.32</td>
</tr>
<tr>
<td>-32.5</td>
<td>50.66</td>
<td>-121.02</td>
<td>106.17</td>
<td>25.99</td>
<td>32.39</td>
<td>-12.83</td>
</tr>
<tr>
<td>-27.5</td>
<td>87.65</td>
<td>-68.5</td>
<td>17.93</td>
<td>74.67</td>
<td>183.45</td>
<td>-14.05</td>
</tr>
<tr>
<td>-22.5</td>
<td>101.75</td>
<td>-119.17</td>
<td>3.91</td>
<td>25.83</td>
<td>314.37</td>
<td>13.05</td>
</tr>
<tr>
<td>-17.5</td>
<td>17.13</td>
<td>-64.29</td>
<td>87.84</td>
<td>43.74</td>
<td>133.49</td>
<td>-48.65</td>
</tr>
<tr>
<td>-12.5</td>
<td>91.87</td>
<td>-66.28</td>
<td>79.84</td>
<td>75.63</td>
<td>167.86</td>
<td>-2.76</td>
</tr>
<tr>
<td>-7.5</td>
<td>352.97</td>
<td>-53.94</td>
<td>118.41</td>
<td>-17</td>
<td>611.29</td>
<td>300.69</td>
</tr>
<tr>
<td>2.5</td>
<td>81.77</td>
<td>277.2</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>12.5</td>
<td>25.53</td>
<td>56.5</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>17.5</td>
<td>188.51</td>
<td>-29.41</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>22.5</td>
<td>125.81</td>
<td>-85.44</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>27.5</td>
<td>-69.25</td>
<td>-51.08</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>32.5</td>
<td>34.24</td>
<td>-0.9</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>37.5</td>
<td>60.29</td>
<td>-180.03</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>42.5</td>
<td>37.61</td>
<td>-87.55</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>57.5</td>
<td>-14.53</td>
<td>-64.72</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>77.5</td>
<td>51.99</td>
<td>-153.3</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

### Table 5 WRS of the plate by SAW

<table>
<thead>
<tr>
<th>Distance from centerline / mm</th>
<th>Line D</th>
<th>Line D’</th>
<th>Line E</th>
<th>Line E’</th>
<th>Line F</th>
<th>Line F’</th>
</tr>
</thead>
<tbody>
<tr>
<td>-77.5</td>
<td>-26.22</td>
<td>-98.74</td>
<td>4.95</td>
<td>-56.03</td>
<td>8.9</td>
<td>28.7</td>
</tr>
<tr>
<td>-57.5</td>
<td>42.48</td>
<td>-29.61</td>
<td>-46.21</td>
<td>-34.18</td>
<td>-3.3</td>
<td>34.15</td>
</tr>
<tr>
<td>-42.5</td>
<td>43.75</td>
<td>-52.56</td>
<td>39.67</td>
<td>-24.6</td>
<td>42.6</td>
<td>-58.32</td>
</tr>
<tr>
<td>-37.5</td>
<td>63.73</td>
<td>-60.89</td>
<td>-43.15</td>
<td>15.1</td>
<td>-13.76</td>
<td>-18.33</td>
</tr>
<tr>
<td>-32.5</td>
<td>-10.85</td>
<td>-23.39</td>
<td>67.53</td>
<td>-19.91</td>
<td>62.5</td>
<td>-73</td>
</tr>
<tr>
<td>-27.5</td>
<td>10.9</td>
<td>-32.47</td>
<td>10.86</td>
<td>-19.59</td>
<td>29.23</td>
<td>-78.93</td>
</tr>
<tr>
<td>-22.5</td>
<td>59.36</td>
<td>-48.15</td>
<td>53.88</td>
<td>1.85</td>
<td>20.96</td>
<td>-38.53</td>
</tr>
<tr>
<td>-17.5</td>
<td>40.15</td>
<td>-33.96</td>
<td>-35.27</td>
<td>-32.97</td>
<td>39.52</td>
<td>-81.32</td>
</tr>
<tr>
<td>-12.5</td>
<td>-34.23</td>
<td>-94.06</td>
<td>-50.47</td>
<td>-20.71</td>
<td>-11.48</td>
<td>106.27</td>
</tr>
<tr>
<td>-7.5</td>
<td>142.29</td>
<td>92.98</td>
<td>186.98</td>
<td>107.98</td>
<td>160.97</td>
<td>177.45</td>
</tr>
<tr>
<td>2.5</td>
<td>155.8</td>
<td>112.36</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>12.5</td>
<td>-21.29</td>
<td>-53.73</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>17.5</td>
<td>-35.47</td>
<td>30.46</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>22.5</td>
<td>-51.95</td>
<td>48.32</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>27.5</td>
<td>-27.69</td>
<td>13.73</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>32.5</td>
<td>62.61</td>
<td>-11.65</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>37.5</td>
<td>35.81</td>
<td>-32.26</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>42.5</td>
<td>72.37</td>
<td>-49.82</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>57.5</td>
<td>80.42</td>
<td>-14.97</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>77.5</td>
<td>75.02</td>
<td>-13.86</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>
Experimental Study of Welding Residual Stress of High-strength Shipbuilding Steel

Table 6 WRS of the cylinder

<table>
<thead>
<tr>
<th>Distance from centerline / mm</th>
<th>Residual stress / Mpa</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Line G</td>
</tr>
<tr>
<td>-77.5</td>
<td>-76.89</td>
</tr>
<tr>
<td>-57.5</td>
<td>-125.96</td>
</tr>
<tr>
<td>-42.5</td>
<td>-111.26</td>
</tr>
<tr>
<td>-32.5</td>
<td>-133.52</td>
</tr>
<tr>
<td>-27.5</td>
<td>-120.02</td>
</tr>
<tr>
<td>-22.5</td>
<td>-73.91</td>
</tr>
<tr>
<td>-17.5</td>
<td>-90.74</td>
</tr>
<tr>
<td>-12.5</td>
<td>126.3</td>
</tr>
<tr>
<td>-7.5</td>
<td>201.96</td>
</tr>
<tr>
<td>-2.5</td>
<td>98.98</td>
</tr>
<tr>
<td>2.5</td>
<td>94.98</td>
</tr>
<tr>
<td>12.5</td>
<td>-34.04</td>
</tr>
<tr>
<td>17.5</td>
<td>-61.45</td>
</tr>
<tr>
<td>22.5</td>
<td>-54.69</td>
</tr>
<tr>
<td>27.5</td>
<td>32.09</td>
</tr>
<tr>
<td>32.5</td>
<td>-10.41</td>
</tr>
<tr>
<td>37.5</td>
<td>-48.64</td>
</tr>
<tr>
<td>42.5</td>
<td>-12.38</td>
</tr>
<tr>
<td>57.5</td>
<td>23</td>
</tr>
</tbody>
</table>

5. Discussion

5.1 Maximum and minimum WRS

Fig. 4 depicts the maximum residual tensile stress and residual compressive stress for each measuring line. The peak value of the tensile stress of most measuring lines is approximately 200 Mpa, and the maximum value exceeds 600 Mpa. Most of the residual compressive stress peaks are approximately −100 Mpa, and the maximum compressive stress exceeds −300 Mpa. Line C has no measured compressive stress value. The maximum tensile stress appears at −2.5 mm from the centreline of Line C, thereby exceeding the yield limit of the material. The condition can be attributed to two possible reasons. Firstly, the measured value may be the longitudinal residual stress of the weld bead. The mechanism is that the longitudinal shrinkage of the weld is constrained, and the longitudinal residual tensile stress exists in a narrow area near the weld. The value decreases with the increase in distance from the centreline. When no phase change occurs, the maximum value will reach or exceed the yield limit of the material. Secondly, the mechanical properties of the material during the welding heat process change with the increase or decrease in temperature, thereby directly affecting welding thermoplasticity, stress–strain process and magnitude of residual stress. The peak tensile stress in low carbon steel may approach the yield strength of the material.
5.2 Distribution characteristics of WRS

Fig. 5 demonstrates the results of Measuring lines A and D of the steel plates. The WRSs of the two lines vary with the distance from the centreline of the weld. The tensile stress appears near the centre section of the weld, and the peak values of the tensile stresses of the two lines are close to the centreline. This is due to the local heating of the heat source causes the thermal expansion of the joint and the surrounding area, thus resulting in compressive plastic deformation. When heat source arrives, the deformation reaches the maximum. The weld metal will crystallise, solidify and shrink whilst the heat source leaves. After welding, the residual stress will be tensile near the welding area. The peak value of Line A is 60% of the yield strength of the material; this value is 25% of Line D. A high tensile stress zone can easily cause cold cracks near the weld bead which depends on combining the basic welding metal and welding rods, as well as the process parameters. The residual stress decreases and stabilises with the increase in the distance between the measuring points and the centreline. The WRS distribution of the two lines is not completely symmetrical. The reason for which is related to the model parameters, welding process and measurement error.

Fig. 6 exhibits the results of longitudinal weld Measuring line G and circumferential weld Measuring lines J and K of the cylinder. The stress peak also appears in the area that surrounds the weld. The peaks of Measuring lines G and J are approximately 35% of the yield strength and that of Line K is approximately 20%. The residual stress distribution of Lines G and K is steep, thereby resulting in a large gradient residual stress. This result is due to welding is an abnormal local thermal process, accompanied by a highly nonlinear process of deformation. The high-temperature zone centred on the welding point is formed.
instantaneously, but the low-temperature zone may exist close to the peak of high temperature. A large temperature gradient exists between the two temperature zones, thus generating a large plastic strain gradient in the weld zone. Then, a large gradient residual stress will be generated around the weld zone.

Fig. 6 WRS distribution of Measuring lines G, J and K

5.3 Effect of welding processes

Fig. 7 displays that the residual stress curve of EW fluctuates considerably, and the condition of Measuring lines A’, B and B’ is obvious. Several extreme points can be observed, whilst the maximum tensile stress does not appear at the weld. The stress distribution of EW is irregular. The residual stress curve of the SAW is relatively stable, and the residual stress curve generally decreases with the increase in distance from the weld. The maximum tensile stress appears at the weld, that is, −2.5 mm from the centreline because EW is a manual operation whose process parameters are difficult to control. Residual stress is related to various factors, such as angle of welding, temperature and length-of-stay time at the same position. A relatively irregular distribution form is generated. The process parameters of the SAW are constant, thus resulting in a regular residual stress distribution. The stress amplitude of the EW varies significantly, whereas the amplitude of the SAW is small. The maximum values of tensile and compressive stresses appear through the EW, which generates large energy input in several areas during the welding process.
The measurement results of the different measuring lines of the same weld are compared. In Fig. 8, the residual stress distribution of each line is irregular. The residual stress of each line is either from tensile to compressive or gradually from compressive to tensile. The measured values of the measuring points with the same distance from the weld vary significantly. The peak tensile stress of Lines A and C appear at the weld, and the tensile stress of Measuring line C is relatively large. The peak compressive stress of Line B is at \(-57.5\) mm from the centreline. In Fig. 9, the stress distribution consistency of each line is improved, and the stress values of several points located from \(-10\) mm to \(-50\) mm are basically equal. The maximum tensile stress of each line is approximately 180 Mpa at \(-2.5\) mm from the centreline. The compressive stress is small and appears at measuring line E, which is \(-50.47\) Mpa. It can be seen that due to the various welding processes, the residual stress distributions of the same weld bead are different. The process stability is better in SAW than in EW.
5.4 Both sides of the plates

In Fig. 10, the WRS of both sides of the plates are different. In general, the trend of residual stress change on both sides is approximately the same, and the measuring stress is higher on the top side than on the backside. This result is due to the two sides have different cooling processes during the welding process, and weld shrinkage occurs. Moreover, the possible superposition of phase transition is also an important factor that contributes to the difference.
5.5 Comparison of the plates and cylinder

In Fig. 11, the WRSs are smaller in Measuring lines J and K of the circumferential weld than in the plate’s Measuring line A because the weld has undergone radial shrinkage deformation, thereby releasing a portion of the stress. The stresses of Lines J and K are 91.92 and −67.33 Mpa, correspondingly; these stress values are small because the residual stress of the circumferential weld is superimposed by two parts. Firstly, opposite thermal stresses of thermal expansion and contraction constantly exist on both sides of the cylinder considering the temperature difference between the outer and inner sides of the cylinder. Tensile stress is generated on the high-temperature side, whereas compressive stress is generated on the low-temperature side. Secondly, concave and additional bending moments and stresses are generated on the weld and its sides due to the cooling contraction of the weld. Tensile residual stress is generated at the inner side, whereas compressive residual stress is generated on the outer side. The thickness and diameter of the cylinder are small, and the bending stress will be small or even negative. The comparison of the measuring lines of the axial weld and plate indicates that the maximum WRSs are nearly equal. The tensile stress zone of the longitudinal weld is narrow and quickly transferred to the compressive stress zone with maximum stress of −133.52 Mpa. This result is related to the distribution of the temperature field during welding.
The temperature gradient in the weld and its narrow region is large. Furthermore, the peak temperature away from the weld is sharply decreased, and the gradient tends to be small.

6. Conclusions

The plates and cylinder experimental modes of high-strength shipbuilding steel are created to measure the WRS. The models are welded by EW and SAW. The WRS was measured by the instrumented indentation method.

Among the measured results, the maximum tensile stress is 611.29 MPa which exceeds the yield stress of the material. The maximum compressive stress is −346.43 Mpa. WRS mainly exists at the weld and heat-affected zone near the weld bead. Tensile stress is mainly generated at the weld, and compressive stress is generated away from the weld. Different welding processes will result in various forms of WRS distributions. The stress distribution of EW is minimally regular, and the process stability is better in SAW than in EW. Given the different welding and cooling processes, the WRSs on both sides of the plates are different. The measured stress is smaller in the circumferential weld than in the plates due to the shrinkage deformation of the weld of the cylinder. The longitudinal weld of the cylinder has a narrow tensile stress zone and a large gradient stress variation.

The instrumented indentation method has high accuracy and speed in measuring the residual stresses and is suitable for stress assessment at shipbuilding sites. WRS has an impact on ship performance. The measured data in this research can provide data basis for accurate prediction of ship structural safety, reliability, fatigue performance etc. According to the calculation results, corresponding process improvement measures may be taken to improve the shipbuilding quality.

Acknowledgments

The authors gratefully acknowledge the financial support of the Fund of State Key Laboratory of Ocean Engineering [GKZD010074] and Shanghai Sailing Program [18YF1411500].
REFERENCES


Lee, Young-Gill  
Kim, Cheolho  
Park, Jeong-Ho  
Kim, Hyeongjun  
Lee, Insu  
Jin, Bongyong

http://dx.doi.org/10.21278/brod70204

NUMERICAL SIMULATIONS OF ADDED RESISTANCE IN REGULAR HEAD WAVES ON A CONTAINER SHIP

UDC 629.5.015.2:629.5.017.2:629.544
Original scientific paper

Summary

Although shipping was not included in the final text of the Paris Agreement adopted in 2015, the IMO (International Maritime Organization) has been making efforts to cope with climate change by regulating the amount of air pollutants and greenhouse gas emissions from ships, and introduced EEOI (Energy Efficiency Operational Indicator) for ships in operation and EEDI (Energy Efficiency Design Index) for new ships. In order to meet these requirements, it is essential to estimate the resistance and propulsion performance of a ship in waves to predict minimum energy consumption rate through experiments or numerical simulations. However, most previous research on ship resistance has dealt with resistance in the calm condition. Nonetheless, ships sailing in real sea conditions experience more
resistance than in the calm condition due to winds and waves. The resistance induced by winds and waves are called air resistance and added resistance, respectively. In the case of high speed craft in waves, more than 30% of total resistance is taken up by added resistance ([1], [2]) and ship speed can be reduced by over 10% because of waves [3].

Research on added resistance of ships in waves has been carried out for decades. Research in early 1970 was performed based on experiments on the Series 60 ([4], [5]) and S175 container ship model ([6], [7]), etc. Later, Journee [8] systematically carried out added resistance experiments for four Wigley hull form and Kuroda, et al. [9] have recently studied added resistance based on experiments while changing the bow shape of a container ship. Also, Park, et al. [10] proposed a study on uncertainty for reliability evaluation of added resistance experimental values through repeated experiments in various cases [11].

Both the experimental studies and theoretical and numerical approaches have been performed. Most initial stage research based on potential theory can be classified into a momentum conservation or pressure integration methods based on analysis method. The momentum conservation method was proposed by Mauro [12] and it has been widely used because added resistance can be obtained with a relatively simple calculation and without computing pressures on ship surfaces. Although equations of the pressure integration method are complicated, it is useful for physical analysis [13]. In the past, strip theory was mostly used to calculate motion for added resistance analysis. However, thanks to improved computational performance, added resistance analysis using a 3D Green function method and a Rankine panel method in frequency domain, and a higher-order Rankine panel method are being studied [14]. Seo, et al. [15] dealt with changing patterns of added resistance due to changes in hull forms of KCS and KVLCC2 above water lines using a nonlinear motion analysis method with a time-domain Rankine panel method. Oh & Yang [16] developed added resistance computation module by using a modified radiated energy method and short wave correction method of NMRI so that the module can be used in the initial design or embedded module for navigation support systems. Jeong, et al. [17] performed sensitivity analysis of added resistance and weather factors in the representative sea state for initial hull form design stage and also analyzed added resistance for KVLCC2 and Supramax in waves by using a momentum conservation method.

Although methods based on potential theory have been useful in the realistic design stage, practical results cannot be obtained when the heights of incoming waves are high or ship motions become excessive [18]. In order to overcome this limitation, research on added resistance analysis using CFD (Computational Fluid Dynamics) taking viscosity into account due to recent improvements in hardware. Orihara and Miyata [19] studied on numerical simulation with WISDAM-X which developed by themselves using an overlapping grid system for analysis of added resistance and motions of an SR-108 container ship in regular head waves. Hu and Kashiwagi [20] used a Constrained Interpolation Profile (CIP) method and a rectangular grid system for analysis of added resistance. Visonneau, et al. [21] performed added resistance analysis in regular head waves by using ISIS-CFD, in which unstructured mesh and analytical weighting mesh deformation are adopted. Park, et al. [22] analyzed motions of a DTMB 5415 hull form in regular head waves by using WAVIS-6DoF with a fixed grid system and compared the results with experimental results. Yang, et al. [23] used a THINC (Tangent Hyperbola for Interface Capturing) method and a rectangular grid system based on a numerical method to analyze motions and added resistance of a Wigley III hull form and S175 container ship. Jeong & Lee [18] exploited a modified Marker-density method and a rectangular grid system for a vortex induced vibration (VIV) analysis for a circular cylinder also performed motions and added resistance analysis on Wigley III and KCS hull forms and compared the simulation results with public experimental results. Tezdogan et al. [24] performed a fully nonlinear unsteady RANS simulation to predict the
Numerical simulations of added resistance in regular head waves on a container ship

Young-Gill Lee, Cheolho Kim, Jeong-Ho Park

Hyeongjun Kim, Insu Lee, Bongyong Jin

ship motions and added resistance of a full scale KRISO Container Ship model and estimated the increase in effective power and fuel consumption due to its operation in waves. Oh, et al. [14] carried out added resistance model tests in waves with both the original and Ax-bow hull form of an AFRAMAX-Class tanker, performed numerical simulations by a using Rankine panel method based 3D time domain motion analysis program(WISH) and a commercial CFD program(STAR-CCM+). The simulation results were compared with the model test results. Seo & Park [11] used open source library OpenFOAM to compute added resistance and motion performance of a KCS hull form in regular head waves, and compared the results with public experimental results. Kim & Lee [25] also exploited OpenFOAM to analyze resistance performance of a KCS hull form depending on trim conditions in regular waves. In spite of the research being performed, high accuracy of numerical simulation remains a challenge.

In previous research, a wave generation zone to create waves and damping zones to prevent and cancel out reflection of waves are set when generating waves. A domain corresponding to 1-2 times the wave domain, except for the computational domain, has to be generally generated. Furthermore, when the body-fitted grid system is exploited for motion analysis in waves, the re-finining or moving of the grid system according to the moving body is required as well. The necessary algorithm is complicated and requires significant computational time [18].

In this research, added resistance in regular head waves on KCS was studied. The commercial CFD software STAR-CCM+ V11.06 was used for numerical analysis while not setting wave generating and damping zones, and a method that does not require refining and moving the grid system was adopted. The experiment was performed in a FORCE Technology organization and the simulation results were compared with the public results in a 2015 Tokyo workshop (Case 2.10) [26].

2. Numerical simulation methods

The mass conservation equation and the momentum conservation equations including turbulence model were calculated in the incompressible based unsteady state. For the coupling of velocity and pressure, a SIMPLE [27] method was used, and Standard K-Epsilon Low-Re model and wall function were selected as the turbulence model. In order to increase the convergence performance of linear algebraic equation AMG (Algebraic Multi-Grid) method [28] was used. And using Gauss-Seidel method the simultaneous linear equation solved. Fifth-order Stokes wave theory was applied to express waves and the wave forcing method, which is a built-in method of STAR-CCM+, was used to minimize wave reflections at the boundary, which should be non-reflecting. The governing equations and numerical schemes are represented in Table 1.

The mass conservation equation and the momentum equations, which are governing equations for numerical simulation, are expressed as shown in equation (1) and (2), respectively.

\[
\frac{d}{dt} \int_V \rho \, dV + \int_S \rho \vec{v} \cdot \vec{n} \, dS = 0 \tag{1}
\]

\[
\frac{d}{dt} \int_V \rho u_i \, dV + \int_S \rho u_i \vec{v} \cdot \vec{n} \, dS = \int_S \left( \tau_{ij} \vec{v}_j - p \vec{t}_i \right) \cdot \vec{n} \, dS + \int_V \rho g \vec{t}_i \, dV + \int_V q_i \, dV \tag{2}
\]

\(V\) is a finite volume limited by a closed surface \(S\), \(\vec{v}\) is the fluid velocity vector including velocity of \(u_i\) and the unit vector which is vertical to \(S\) and towards outer direction is \(\vec{n}\). \(t\) is time, \(p\) is pressure, \(\rho\) is fluid density, \(\tau_{ij}\) is viscous stress tensor and \(\vec{t}_i\) is the unit vector in the
Young-Gill Lee, Cheolho Kim, Jeong-Ho Park, Hyeongjun Kim, Insu Lee, Bongyong Jin

Numerical simulations of added resistance in regular head waves on a container ship

$x_f$-direction. $g$ is the gravitational acceleration and $q_i$ is the momentum source term. In this study, 2 numerical options which control the free surface wave around boundaries. One is wave damping [29] and the other is wave forcing [30] boundary option. A combination of linear and second-order damping, and one or a combination of both can be used for wave damping. The following equation (3) is used as the source term in equation (2) expressed in tensor form.

$$q_i^d = \rho (f_1 + f_2 | w |) \frac{x_d - x_{sd}}{x_{sd} - x_{sd}} w$$

$$\kappa = \left( \frac{x - x_{sd}}{x_{sd} - x_{sd}} \right)^{n}$$

Where $x$ is the vertical direction with regard to the free surface, $w$ is the z-direction velocity component, $x_{sd}$ is the starting point of wave damping propagating in the x-direction and $x_{sd}$ is the location where wave damping ends. $f_1$ and $f_2$ are the linear and the second-order damping coefficients, respectively. Factor $f_1$ and $f_2$, $n = 2$, are can be changed for control the strength of damping. As we can see in equation defining $\kappa = \left( \frac{x - x_{sd}}{x_{sd} - x_{sd}} \right)^{n}$, the value of $\kappa = \left( \frac{x - x_{sd}}{x_{sd} - x_{sd}} \right)^{n}$ can be changed on range of the damping zone applied in numerical domain.

Wave forcing is a technique blending the CFD solution and the undisturbed wave solution. This boundary option is the same concept as EOM (Euler Overlay Method) which is applied in research by Kim et al. [30]. It is used to prevent the scattered waves at the body from coming into the CFD domain after being reflected at the boundaries. In EOM method [30], additional source term is added to each conservation equation. The new source term is given by following.

$$S_{\phi}^* = S_{\phi} - \mu \rho (\phi - \phi_2)$$

$\phi_2$ is the reference value from theoretical wave solution. Navier-Stokes equations discretized at a certain distance are applied by other theoretical solutions, the calculation efficiency can be increased by reducing the size of the solution domain. Like EOM method, wave forcing can remove the problem related to wave reflection at the boundaries due to the damping characteristics of gradual constraint. Likewise wave damping, equation (4) is used as the source term in equation (2) expressed in tensor form. As equation (4) is the newly defined source term, it reduces the difference between the CFD solution and the undisturbed wave solution. $u_{tref}$ is a reference velocity component and $u_i$ is a velocity component.

$$\gamma = \gamma_{ref} \cos^2 \left( \frac{\pi}{2} + \frac{\pi}{2} \frac{x_{sd} - x_{sd}}{x_{sd} - x_{sd}} \right)$$

Equation (5) identifies the forcing coefficient $\gamma$, which has same purpose of $\mu$ in equation (4). The wave forcing technique enables gradual application of the constraint described above to dampen the difference between forced and recent solutions at each computational time by applying the characteristics of the spatial distribution of the $\gamma$ value. In equation (5), $\gamma_{ref}$ refers to the maximum value of $\gamma$. The forcing coefficient $\gamma$ represents the strength of forcing. It has a cosine form as shown in equation (5). The point where $\gamma$ has its maximum value spatially refers to the boundary surface at which it becomes the minimum at which forcing ends. If the value of $\gamma$ is at a maximum, which means the boundary, the
Numerical simulations of added resistance in regular head waves on a container ship
Young-Gill Lee, Cheolho Kim, Jeong-Ho Park
Hyeongjun Kim, Insu Lee, Bongyong Jin

Theoretical solution of the transport equation will become the same as the calculated solution of the transport equation at each computational time.

Table 1 Governing equations and numerical scheme

<table>
<thead>
<tr>
<th>Governing equations</th>
<th>Reynolds-averaged Navier-Stokes (RANS) equations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Free surface and wave</td>
<td>Volume of fluid method (VOF), Stokes’ 5th-order wave</td>
</tr>
<tr>
<td>Body motion</td>
<td>Dynamic fluid body interaction (DFBI), Heave and pitch free</td>
</tr>
<tr>
<td>Turbulent model</td>
<td>Standard K-Epsilon Low-Re, All y+ Wall Treatment</td>
</tr>
<tr>
<td>Mesh</td>
<td>Trimmed mesh &amp; prism layer with wall function</td>
</tr>
</tbody>
</table>

3. Principal particulars of KCS and numerical simulation conditions

The experiment was performed in FORCE Technology and a KCS (KRISO Container Ship) is used in both the numerical simulation and experiment. The scale of the model ship used in the simulation is 1/37.89, the main particulars and the geometry are represented in Table 2. and Fig. 1 Geometric model of KCS with rudder and section lines of the fore-body, respectively. The speed of the model ship is 2.006m/s, which is the design speed.

Assuming infinite water depth, simulation conditions were defined as shown in Table 3. Trim and sinkage were free and other motions were fixed. The waves were generated using the Stokes’ 5th-order wave.

In order to generate waves which have a significant effect on the accuracy of analysis, the grids were generated, varying with the height of each incoming wave. In this research, 10 to 20 grids were generated in the incoming waves along the vertical direction and the ratio of the horizontal and vertical grid size \( \Delta x/\Delta z \) was constrained at 8. The number of grids used and the size of vertical direction grid sizes used for wave generation are shown in Table 5 for each wave condition. The base size of grids used for all computations is 0.15m. In order to increase the accuracy of wave generation, a wave forcing technique was used [31].

The numerical computation domain used in this research for the resistance in calm water condition and the added resistance in waves is illustrated in Fig. 2. The origin of the coordinate system is \((0, 0, 0)\) which is at AP (After Perpendicular), Center line \((Y=0)\) and Base line \((Z=0)\), and the upward vertical direction was defined as +z. Time step size which is used in resistance calculation in calm water is chosen as 0.02s to restrict courant number under 1.0.

The boundary conditions on the simulation of the resistance calculation in the calm water condition and the added resistance calculations in waves are shown in Table 4. The wall boundary condition was applied to the hull surface and rudder in order to meet the non-slip condition. In the case of the resistance analysis in the calm water condition, \(L_{PP}/2\) of wave damping was applied at the inlet, outlet and side boundaries as shown in Fig. 2 [31]. For the added resistance analysis in waves, the wave forcing length was applied as shown in Fig. 3. Fig. 4 shows the generated grid for the added resistance analysis in waves. Fig. 5 represents the 2.5D grids which were generated to verify if the generated wave without ship hull was accurate and reproducible.
Numerical simulations of added resistance in regular head waves on a container ship

Table 2 Main particulars of KCS with rudder

<table>
<thead>
<tr>
<th>Main particulars</th>
<th>Full scale</th>
<th>Model scale</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length between perpendiculars, $L_{PP}$ (m)</td>
<td>230</td>
<td>6.0702</td>
</tr>
<tr>
<td>Maximum beam of waterline (m)</td>
<td>32.2</td>
<td>0.8498</td>
</tr>
<tr>
<td>Draft (m)</td>
<td>10.8</td>
<td>0.2850</td>
</tr>
<tr>
<td>Displacement volume (m$^3$)</td>
<td>52030</td>
<td>0.9571</td>
</tr>
<tr>
<td>Wetted surface area w/o rudder (m$^2$)</td>
<td>9424</td>
<td>6.6177</td>
</tr>
<tr>
<td>Wetted surface area of rudder (m$^2$)</td>
<td>115.0</td>
<td>0.0801</td>
</tr>
<tr>
<td>$LCB$ ($%L_{PP}$, fwd+)</td>
<td>-1.48</td>
<td>-1.48</td>
</tr>
<tr>
<td>Vertical Center of Gravity (from keel) (m)</td>
<td></td>
<td>0.378</td>
</tr>
<tr>
<td>Moment of Inertia ($K_{xx}/B$)</td>
<td>0.40</td>
<td></td>
</tr>
<tr>
<td>Moment of Inertia ($K_{yy}/L_{PP}$, $K_{zz}/L_{PP}$)</td>
<td>0.250</td>
<td>0.252</td>
</tr>
</tbody>
</table>

Table 3 Wave conditions

<table>
<thead>
<tr>
<th>Case no.</th>
<th>$\lambda/ L_{PP}$</th>
<th>Wave height (m)</th>
<th>Wave length, $\lambda$ (m)</th>
<th>Wave steepness</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td></td>
<td></td>
<td></td>
<td>Calm water</td>
</tr>
<tr>
<td>1</td>
<td>0.65</td>
<td>0.062</td>
<td>3.949</td>
<td>0.016</td>
</tr>
<tr>
<td>2</td>
<td>0.85</td>
<td>0.078</td>
<td>5.164</td>
<td>0.015</td>
</tr>
<tr>
<td>3</td>
<td>1.15</td>
<td>0.123</td>
<td>6.979</td>
<td>0.018</td>
</tr>
<tr>
<td>4</td>
<td>1.37</td>
<td>0.149</td>
<td>8.321</td>
<td>0.018</td>
</tr>
<tr>
<td>5</td>
<td>1.95</td>
<td>0.196</td>
<td>11.840</td>
<td>0.017</td>
</tr>
</tbody>
</table>

Table 4 Boundary conditions

<table>
<thead>
<tr>
<th>Boundary</th>
<th>Type</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>In calm</td>
<td>In wave</td>
</tr>
<tr>
<td>Inlet</td>
<td>Velocity inlet</td>
<td>Velocity inlet</td>
</tr>
<tr>
<td>Outlet</td>
<td>Pressure outlet</td>
<td>Pressure outlet</td>
</tr>
<tr>
<td>Top</td>
<td>Velocity inlet</td>
<td>Velocity inlet</td>
</tr>
<tr>
<td>Bottom</td>
<td>Velocity inlet</td>
<td>Velocity inlet</td>
</tr>
<tr>
<td>Side</td>
<td>Symmetry plane</td>
<td>Symmetry plane</td>
</tr>
<tr>
<td>Symmetry</td>
<td>Symmetry plane</td>
<td>Symmetry plane</td>
</tr>
</tbody>
</table>
Table 5 Number and size of mesh according to wave conditions

<table>
<thead>
<tr>
<th>Case no.</th>
<th>Total mesh number</th>
<th>Relative Z size (% of base size)</th>
<th>Mesh number in Z per wave height</th>
<th>Time steps size (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>abt. 2.87e+06</td>
<td>3.125</td>
<td>abt. 13</td>
<td>0.0025</td>
</tr>
<tr>
<td>2</td>
<td>abt. 3.81e+06</td>
<td>3.125</td>
<td>abt. 16</td>
<td>0.0025</td>
</tr>
<tr>
<td>3</td>
<td>abt. 4.48e+06</td>
<td>3.125</td>
<td>abt. 26</td>
<td>0.00125</td>
</tr>
<tr>
<td>4</td>
<td>abt. 1.60e+06</td>
<td>6.250</td>
<td>abt. 15</td>
<td>0.0025</td>
</tr>
<tr>
<td>5</td>
<td>abt. 1.66e+06</td>
<td>6.250</td>
<td>abt. 20</td>
<td>0.0025</td>
</tr>
</tbody>
</table>

Fig. 1 Geometric model of KCS with rudder and section lines of the fore-body

Fig. 2 Numerical domain in calm and wave conditions
4. Simulation results

4.1 Resistance analysis results in calm condition

Resistance analysis (Case 0) of KCS in the calm water condition was performed at the model ship speed of 2.006 m/s and approximately 1.15 million grids were used. Fig. 6 and Table 6 represent the comparison between total resistance, trim and sinkage results obtained from the numerical simulation and the model test. Since STAR-CCM+ uses the right-handed coordinate system, positive sinkage means upward and the positive value of trim means when
Numerical simulations of added resistance
in regular head waves on a container ship
Young-Gill Lee, Cheolho Kim, Jeong-Ho Park
Hyeongjun Kim, Insu Lee, Bongyong Jin

the bow part is lower than the stern part of the ship. The total resistance result obtained from numerical simulation is close to the model test result with a difference of around 2%. On the other hand, both sinkage and trim results show about a 10% difference. It seems the z-direction (0.009 m) size of the grid is comparatively large to compute changes in sinkage. If the z-direction size of the grid is reduced by considering sink-age changes, it is expected to decrease the difference with the model test results.

Fig. 6 Comparison of total resistance, sinkage and trim with experimental data

Table 6 Total resistance, Sinkage and trim in calm water (Case 0)

<table>
<thead>
<tr>
<th></th>
<th>EFD</th>
<th>CFD</th>
<th>Difference (% with EFD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total resistance [N]</td>
<td>51.591</td>
<td>52.627</td>
<td>-2.01</td>
</tr>
<tr>
<td>Sinkage [m]</td>
<td>-0.0126</td>
<td>-0.0113</td>
<td>10.27</td>
</tr>
<tr>
<td>Trim [deg]</td>
<td>0.1646</td>
<td>0.1799</td>
<td>-9.29</td>
</tr>
</tbody>
</table>

* Difference with EFD is \((E-S)/E \times 100\), where E is the EFD value, and S is the simulation value

4.2 Comparison of waves and added resistance according to the presence of wave forcing

The results according to the presence of wave forcing were compared with the results of Case 1. To prevent non-physical reflection at the boundary surface, wave damping was applied when wave forcing was not used and only applied at the outlet boundary for the added resistance analysis in waves. Wave damping length is from outlet boundary to \(L_{PP}/2\) which is 3m. When wave forcing is used, it is applied at the inlet, outlet and side boundary. The wave damping and wave forcing zones used for the added resistance analysis in wave are represented in Figs. 7 and 8.

Fig. 9 shows the measured time series values of the waves and the differences with the theoretical values are shown in Table 7. The location of measuring the generated waves is FP (Forward Perpendicular) of KCS when there is no ship hull and the wave height changes with
time were measured when the crest of the generated wave is located at FP for the accuracy of the waves.

From the results shown in Fig. 9 and Table 7, the difference due to the presence of wave forcing is not large when there is no ship. However, when there is a ship hull and wave forcing is applied, the wave approaching the ship hull seems to be excessively disturbed by the reflected waves by ship hull. In addition to that, the total resistance and motion response results showed continuous oscillation with a long period as shown in Fig. 10. Therefore, this research studied on the added resistance and the motion response with applying wave forcing method.

**Table 7** Wave amplitude in wave conditions regard with and without wave forcing

<table>
<thead>
<tr>
<th></th>
<th>Stokes’ 5th order wave</th>
<th>Difference (% with theory)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Theory</td>
<td>CFD with wave forcing</td>
</tr>
<tr>
<td>Crest</td>
<td>0.03174</td>
<td>0.03145</td>
</tr>
<tr>
<td>Trough</td>
<td>-0.03021</td>
<td>-0.03007</td>
</tr>
</tbody>
</table>

* Difference with theory is (T-S)/T × 100, where T is the theory value, and S is the simulation value

---

**Fig. 7** Wave damping zone for added resistance in wave condition

**Fig. 8** Wave forcing zone for added resistance in wave condition
Numerical simulations of added resistance in regular head waves on a container ship

Young-Gill Lee, Cheolho Kim, Jeong-Ho Park
Hyeongjun Kim, Insu Lee, Bongyong Jin

(a) Without wave forcing

(b) With wave forcing

**Fig. 9** Time history of wave elevation with and without wave forcing

(a) Total resistance
Young-Gill Lee, Cheolho Kim, Jeong-Ho Park, Hyeongjun Kim, Insu Lee, Bongyong Jin

Numerical simulations of added resistance in regular head waves on a container ship

4.3 Wave generation results

Applying wave forcing, the location of measurement for the generated waves is FP (Forward Perpendicular) of KCS when there is no ship hull and the wave height changes with time were measured when the crest of the generated wave is located at FP for the accuracy of waves. To judge the accuracy of the waves, the simulation results of waves progressing for 10 periods after 20 periods were compared with the theoretical values of the waves.

**Fig. 10** Comparison of total resistance, heave and pitch with and without wave forcing
Numerical simulations of added resistance in regular head waves on a container ship

Young-Gill Lee, Cheolho Kim, Jeong-Ho Park

Hyeongjun Kim, Insu Lee, Bongyong Jin

(a) Case 1 ($\lambda/L_{PP} = 0.65$)

(b) Case 2 ($\lambda/L_{PP} = 0.85$)

(c) Case 3 ($\lambda/L_{PP} = 1.15$)
Numerical simulations of added resistance in regular head waves on a container ship

Fig. 21 Time history of wave elevation under wave conditions

Table 8 Wave amplitude in wave conditions

<table>
<thead>
<tr>
<th>Case no.</th>
<th>Stokes’ 5th-order wave</th>
<th>Difference (% with theory)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Theory</td>
</tr>
<tr>
<td>1</td>
<td>Crest</td>
<td>0.03174</td>
</tr>
<tr>
<td></td>
<td>Trough</td>
<td>-0.03021</td>
</tr>
<tr>
<td>2</td>
<td>Crest</td>
<td>0.03991</td>
</tr>
<tr>
<td></td>
<td>Trough</td>
<td>-0.03806</td>
</tr>
<tr>
<td>3</td>
<td>Crest</td>
<td>0.06316</td>
</tr>
<tr>
<td></td>
<td>Trough</td>
<td>-0.05975</td>
</tr>
<tr>
<td>4</td>
<td>Crest</td>
<td>0.07656</td>
</tr>
<tr>
<td></td>
<td>Trough</td>
<td>-0.07236</td>
</tr>
<tr>
<td>5</td>
<td>Crest</td>
<td>0.10053</td>
</tr>
<tr>
<td></td>
<td>Trough</td>
<td>-0.09542</td>
</tr>
</tbody>
</table>

* Difference with theory is \((T - S)/T \times 100\), where T is the theory value, and S is the simulation value
4.4 Added resistance analysis results in waves

For the added resistance analysis, the regular waves shown in Table 3 were input. CFD calculation result of total resistance, pitch and heave motion RAO are shown in Fig. 12-14. Overall 5 cases, after 20 encounter periods, numerical results have their repeatability. Considering the accuracy of the waves, the average of the numerical simulation results over 10 periods after 20 were compared with the experimental results. Fig. 15-17 show the total resistance and motions of the ship, and Table 9 compares the simulation results of the total resistance with the 1 period average of the experimental results. $T_e$ in Figs. 15-17 is the encounter period, $\zeta_s$ is the wave amplitude, $k$ is the wave number and $\theta$ is the pitching angle which is expressed in radians. Since the average was provided for Case 3 instead of time series values due to the resonance period during the experiments, the simulation result was also compared by taking the average of the values.

From the total resistance results of Cases 1-5, the overall results are qualitatively similar to the experimental results except for Case 2. The difference between the total resistance with wave period in Table 9 and the experimental values is less than 4%. Heave and pitch results also show differences in Case 2 but are generally similar to the experimental results (Figs. 15-17).

![Time history of total resistance in wave conditions from 10 to 30 encounter period of CFD result](image)

**Fig. 32** Time history of total resistance in wave conditions from 10 to 30 encounter period of CFD result
Young-Gill Lee, Cheolho Kim, Jeong-Ho Park, Hyeongjun Kim, Insu Lee, Bongyong Jin

Numerical simulations of added resistance in regular head waves on a container ship

Fig. 43 Time history of heave motion in wave conditions from 10 to 30 encounter period of CFD result
Numerical simulations of added resistance in regular head waves on a container ship

Young-Gill Lee, Cheolho Kim, Jeong-Ho Park
Hyeongjun Kim, Insu Lee, Bongyong Jin

(c) Case 3 ($\lambda/L_{PP} = 1.15$)  
(d) Case 4 ($\lambda/L_{PP} = 1.37$)

(e) Case 5 ($\lambda/L_{PP} = 1.95$)

**Fig. 54** Time history of pitch motion in wave conditions from 10 to 30 encounter period of CFD result

**Table 9** Total resistance in wave conditions

<table>
<thead>
<tr>
<th>Case no.</th>
<th>Total resistance [N]</th>
<th>Difference (% with EFD)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>EFD</td>
<td>CFD</td>
</tr>
<tr>
<td>1</td>
<td>55.748</td>
<td>56.514</td>
</tr>
<tr>
<td>2</td>
<td>62.353</td>
<td>62.407</td>
</tr>
<tr>
<td>3</td>
<td>95.218</td>
<td>96.119</td>
</tr>
<tr>
<td>4</td>
<td>94.572</td>
<td>98.191</td>
</tr>
<tr>
<td>5</td>
<td>76.196</td>
<td>74.191</td>
</tr>
</tbody>
</table>

* Difference with EFD is $(E-S)/E \times 100$, where $E$ is the EFD value, and $S$ is the simulation value

(a) Case 1 ($\lambda/L_{PP} = 0.65$)  
(b) Case 2 ($\lambda/L_{PP} = 0.85$)

(c) Case 3 ($\lambda/L_{PP} = 1.15$), mean value of total resistance  
(d) Case 4 ($\lambda/L_{PP} = 1.37$)
Young-Gill Lee, Cheolho Kim, Jeong-Ho Park, Hyeongjun Kim, Insu Lee, Bongyong Jin

Numerical simulations of added resistance in regular head waves on a container ship

Fig. 65 Time history of total resistance in wave conditions

Fig. 76 Time history of heave motion in wave conditions
Fig. 87 Time history of pitch motion in wave conditions

Fig. 98 Time series for force and pitch of KCS in regular heading wave at Case 2 for EFD (symbol), reconstructed based on T2015 values (FORCE: red line) and reconstructed based on current values (blue line)

Fig. 18 shows the reconstructed total resistance and pitch motion from experimental values. The difference between Fig. 15 (b) and Fig. 17 (b) is considered to take place when reconstructing the values.

The transfer functions of those motions can be defined as Equation (6) and (7).

\[ TF_3 = \zeta_3/A \]  \hspace{1cm} (6)

\[ TF_5 = \zeta_5/KA \]  \hspace{1cm} (7)
Young-Gill Lee, Cheolho Kim, Jeong-Ho Park, Hyeongjun Kim, Insu Lee, Bongyong Jin

Numerical simulations of added resistance in regular head waves on a container ship

\( \zeta_3, \zeta_5 \) and \( A \) denote the amplitude of the first harmonic function of the heave, pitch motion and incident wave amplitude respectively and \( k \) means wave number.

Non-dimensional equation of the added resistance is defined as Equation (8).

\[
C_{aw} = \frac{(F_{wave} - F_{calm})}{\rho g \zeta^2 \frac{1}{L_P} B^2 W_L L_P}
\]

(8)

In the equation (8), \( F_{wave} \) means the time average value of resistance in the wave, and \( F_{calm} \) denotes the resistance in calm water condition.

Tables 10 and 11 compare the simulation and experiment result of the heave and pitch RAO (Response Amplitude Operator). Except for Case 2, the difference in heave and pitch RAO with wave period is less than 5%. The simulation results are confirmed to be similar to the experimental results (Fig. 19). The added resistance was compared in Table 12 and the differences between experiment and simulation were less than 10%. Case 5 where the wave length is about 2 times of the ship length showed the largest difference of approximately 12%. This result can also be found in Fig. 19 (c). In Case 3, which corresponded to resonance domain, the difference was slight, about 0.3%. Also, to judge of tendencies for the heave RAO, pitch RAO and the added resistance coefficient, only simulation results of 0.4 and 2.1 in \( \lambda/L_P \) had added. These results are shown in Tables 10-12 and Fig. 19.

### Table 10 Heave RAO

<table>
<thead>
<tr>
<th>Case</th>
<th>( \lambda/L_P )</th>
<th>( TF_3 )</th>
<th>Difference (% with EFD)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>EFD</td>
<td>CFD</td>
</tr>
<tr>
<td>1</td>
<td>0.40</td>
<td>0.007</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0.65</td>
<td>0.130</td>
<td>0.124</td>
</tr>
<tr>
<td>3</td>
<td>0.85</td>
<td>0.244</td>
<td>0.223</td>
</tr>
<tr>
<td>4</td>
<td>1.15</td>
<td>0.899</td>
<td>0.909</td>
</tr>
<tr>
<td>5</td>
<td>1.37</td>
<td>0.874</td>
<td>0.894</td>
</tr>
<tr>
<td>6</td>
<td>1.95</td>
<td>0.933</td>
<td>0.902</td>
</tr>
<tr>
<td>7</td>
<td>2.10</td>
<td></td>
<td>0.925</td>
</tr>
</tbody>
</table>

* Difference with EFD is (E-S)/E \times 100, where E is the EFD value, and S is the simulation value

### Table 11 Pitch RAO

<table>
<thead>
<tr>
<th>Case</th>
<th>( \lambda/L_P )</th>
<th>( TF_5 )</th>
<th>Difference (% with EFD)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>EFD</td>
<td>CFD</td>
</tr>
<tr>
<td>1</td>
<td>0.40</td>
<td>0.009</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>0.65</td>
<td>0.017</td>
<td>0.016</td>
</tr>
<tr>
<td>3</td>
<td>0.85</td>
<td>0.146</td>
<td>0.227</td>
</tr>
<tr>
<td>4</td>
<td>1.15</td>
<td>0.748</td>
<td>0.744</td>
</tr>
<tr>
<td>5</td>
<td>1.37</td>
<td>0.966</td>
<td>0.978</td>
</tr>
<tr>
<td>6</td>
<td>1.95</td>
<td>1.120</td>
<td>1.079</td>
</tr>
<tr>
<td>7</td>
<td>2.10</td>
<td></td>
<td>1.076</td>
</tr>
</tbody>
</table>
Numerical simulations of added resistance in regular head waves on a container ship

Young-Gill Lee, Cheolho Kim, Jeong-Ho Park
Hyeongjun Kim, Insu Lee, Bongyong Jin

Table 12 Added resistance coefficient in waves

<table>
<thead>
<tr>
<th>Case</th>
<th>$\lambda/L_{PP}$</th>
<th>$C_{mw} \times 10^3$</th>
<th>Difference (% with EFD)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>EFD</td>
<td>CFD</td>
<td></td>
</tr>
<tr>
<td>0.40</td>
<td>3.994</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0.65</td>
<td>3.433</td>
<td>6.47</td>
</tr>
<tr>
<td>2</td>
<td>0.85</td>
<td>6.457</td>
<td>9.12</td>
</tr>
<tr>
<td>3</td>
<td>1.15</td>
<td>9.760</td>
<td>0.31</td>
</tr>
<tr>
<td>4</td>
<td>1.37</td>
<td>6.967</td>
<td>-6.01</td>
</tr>
<tr>
<td>5</td>
<td>1.95</td>
<td>1.905</td>
<td>12.36</td>
</tr>
<tr>
<td>2.10</td>
<td>1.400</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

(a) Heave motion  
(b) Pitch motion  
(c) Added resistance coefficient

Fig. 109 Motion RAO and added resistance coefficient under different wave conditions

The proportion of average pressure and shear force in the average of the total resistance is compared in Table 13. Shear force is more dominant in the low period domain. On the other hand, pressure is dominant in the resonance domain. When taking a look at changes in pressure and shear force, the change in pressure due to wave length is larger than that of shear force.
Fig. 20 shows the time history of pressure and shear force in waves over one encounter period. The pressure components of Cases 1, 2 and 5 show a sinusoidal wave form, but the different shapes are shown in Cases 3 and 4. The moments when the pressure distribution acting on the hull in Cases 3 and 4 turn non-linear are shown in Figs. 21 and 22 in order to identify the causes of the difference in the shape of the time series.

Fig. 21 shows the pressure distribution on the ship surface when the $t/Te$ value changes from 0.7 to 0.8 of Case 3 in Fig. 20(c). Fig. 22 shows the pressure distribution on the ship surface when the $t/Te$ value changes from 0.6 to 0.7 of Case 4 in Fig. 20(d). When the pressure non-linearly increases, the impact form of pressure change can be seen at the knuckle located at the bow height of the deck. The high local pressure is acting at the end of the bow while the incoming crest encounters the bow. The impact and the high local pressure acting on the bow are the cause of a change in pressure that is larger than the changes in shear force in the resonance domain. The high pressure acting on the hull is caused at the knuckle and the bow above the waterline (Figs. 21 and 22). Therefore, if the bow shape above the waterline is improved, added resistance will be reduced.

Table 13 Average values of the resistance due to pressure and shear in waves

<table>
<thead>
<tr>
<th>Case no.</th>
<th>Total resistance [N]</th>
<th>Pressure resistance [N]</th>
<th>Pressure resistance / Total resistance [%]</th>
<th>Shear resistance [N]</th>
<th>Shear resistance / Total resistance [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>56.514</td>
<td>14.125</td>
<td>25.00</td>
<td>42.389</td>
<td>75.00</td>
</tr>
<tr>
<td>2</td>
<td>62.407</td>
<td>19.439</td>
<td>31.14</td>
<td>42.981</td>
<td>68.86</td>
</tr>
<tr>
<td>3</td>
<td>96.119</td>
<td>51.228</td>
<td>53.29</td>
<td>44.891</td>
<td>46.71</td>
</tr>
<tr>
<td>4</td>
<td>98.191</td>
<td>52.707</td>
<td>53.68</td>
<td>45.483</td>
<td>46.32</td>
</tr>
<tr>
<td>5</td>
<td>74.191</td>
<td>29.730</td>
<td>40.08</td>
<td>44.461</td>
<td>59.92</td>
</tr>
</tbody>
</table>

(a) Case 1 ($\lambda/L_{pp} = 0.65$)  (b) Case 2 ($\lambda/L_{pp} = 0.85$)  
(c) Case 3 ($\lambda/L_{pp} = 1.15$)  (d) Case 4 ($\lambda/L_{pp} = 1.37$)
Numerical simulations of added resistance in regular head waves on a container ship

Young-Gill Lee, Cheolho Kim, Jeong-Ho Park
Hyeongjun Kim, Insu Lee, Bongyong Jin

(e) Case 5 ($\lambda/L_{PP} = 1.95$)

**Fig. 20** Time history of pressure and shear force in waves over one encounter period

**Fig. 21** Dynamic pressure contours on the hull surface in wave ($\lambda/L_{PP} = 1.15$)
5. Conclusions

In this research, the numerical simulations of the added resistance and the motion response of the KCS hull form including the rudder in the regular head waves were conducted. The wave generation and damping zone were not set by using the wave forcing of the commercial CFD program STAR-CCM+. Also, the grid system did not need to refine and move by using DFBI.

The waves were compared with the Stokes’ 5th-order wave theory. In this research, 10 to 20 grids were generated in the incoming waves along the vertical direction and the ratio of the horizontal and vertical Δx/Δz was constrained at 8.

When the added resistance and motion response results depended on the presence of the wave forcing, the results without the wave forcing confirmed that the result showed long period oscillation. When a wave was generated by applying wave forcing, the difference between the simulation and theory was less than 1% and the wave generation was confirmed reproducible. In the comparison of the added resistance and motion response simulation results, each wave period total resistance difference was less than 4%. Also, the heave and pitch motion response showed similar results with those of the experiments. Differences were below 5%. Furthermore, the added resistance difference between the simulation and experimental results was around 10%. The average of pressure and shear force with regard to the total resistance were compared and the fact that the portion of shear force in total resistance is larger than that of pressure in the low period domain, confirmed the larger pressure portion around the natural period domain.

The reason for the non-linearity in the pressure distributions of case 3 and 4 was studied. The high pressure acting on the hull surface took place at the knuckle and the bow above the waterline, which confirms that the bow shape above the waterline has major effects on the added resistance.

The numerical method used in this research may be useful for calculating added resistance, motion response, etc. In this study, the numerical results of added resistance are compared with the experimental results and the difference is estimated to be within 10%, and it is judged that estimation of added resistance occurred by waves in the seaway is possible.
Numerical simulations of added resistance through CFD. This will increase the accuracy of estimating the power of the engine and thus contribute to a more accurate calculation of the EEOI factor. In further study, numerical simulations of the added resistance and motion response of a KCS hull form with a rudder at other incidence angles of waves will be conducted and the results will be verified by comparing with the public experimental results of the 2015 Tokyo workshop (Case 2.11).

ACKNOWLEDGMENT
This work was supported by INHA UNIVERSITY Research Grant.

REFERENCES
Numerical simulations of added resistance in regular head waves on a container ship

Young-Gill Lee, Cheolho Kim, Jeong-Ho Park, Hyeongjun Kim, Insu Lee, Bongyong Jin


Wang Hao  
Cheng Yuan-Sheng  
Pei Da-Ming  
Hao Wei-Wei  
Gan Lin

http://dx.doi.org/10.21278/brod70203

ISSN 0007-215X  
eISSN 1845-5859

WATER ENTRY HYDROELASTICITY ANALYSIS OF LATTICE SANDWICH PANEL WITH IMPERFECTION: SIMULATION AND ENGINEERING MODEL

UDC 62-419.5:629.5.015.4
Professional paper

Summary

In the present work, the three dimensional (3D) hydroelasticity characteristics of imperfect lattice sandwich panel (ILSP) subjected to water entry via analytical prediction and numerical simulations are proposed. Firstly, numerical investigations are performed on water entry characteristics based on Arbitrary Eulerian-Lagrange (ALE) coupling method for modeling fluid-structure interaction (FSI) at an impact velocity of 5.0m/s. The results show the impact pressure on total FSI surface of ILSP is generally lower than that of the perfect lattice sandwich panel. Then a novel semi-analytical method to calculate the elastic constants of ILSP is introduced. Based on this approach, an engineering computational model is developed to predict the deformation of ILSP, in which the total deformation is separated into two parts: local field deformation and global field deformation. Good agreement between the numerical and analytical results is achieved. And the effects of geometric parameters such as the thickness of face sheet, height of ILSP and relative density of core are discussed.

Key words: Imperfect lattice sandwich panel; Hydroelasticity; Fluid-structure interaction; Water entry

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>ILSP</td>
<td>Imperfect lattice sandwich panel</td>
</tr>
<tr>
<td>RM-ILSP</td>
<td>Random Moved ILSP</td>
</tr>
<tr>
<td>PM-ILSP</td>
<td>Partly Moved ILSP</td>
</tr>
<tr>
<td>$2\tau_s$</td>
<td>Duration of water entry loading</td>
</tr>
<tr>
<td>$f_s$</td>
<td>Natural vibration frequency</td>
</tr>
<tr>
<td>$\alpha, \beta$</td>
<td>Length (width) of structures</td>
</tr>
<tr>
<td>$K_1, K_2$</td>
<td>Dynamic coefficients</td>
</tr>
<tr>
<td>$p_{max}$</td>
<td>Peak impulsive pressure</td>
</tr>
<tr>
<td>$C_0$</td>
<td>Sound speed of air</td>
</tr>
<tr>
<td>$w_{max,G}$</td>
<td>Maximum global deflection of ILSP</td>
</tr>
<tr>
<td>$w_{max,L}$</td>
<td>Maximum local deflection of ILSP</td>
</tr>
<tr>
<td>$\rho_s$</td>
<td>Density of base solid material</td>
</tr>
</tbody>
</table>
1. Introduction

It has been revealed that slamming on ship bow section may cause serious damages. Thus, in the conceptual design of ship and offshore structures, the prediction of hydrodynamic pressure acting on an impacting body is very important. Early in 1929, Von Karman [1] firstly introduced a significant contribution on this subject. Following Karman's classic works on hull-water impact, a series of studies on the theoretical analysis of water entry problem have been reported (Wagner [2], Kapsenberg [3], Faltinsen [4-5] Morabito [6], Korobkin [7-8] and Abrate [9] et al.). Due to the complexities of this problem (such as multi-physics interaction phenomenon, free surface tracking et al), only a few special cases of wedges entry water vertically at a constant velocity can be solved analytically. So, water entry problem has been solved by some different numerical methods (e.g. Boundary Element Method (BEM) [10-11], Arbitrary Lagrange-Euler Method (ALE) [12-13], Volume of Fluid Method (VOF) [14-16], Smoothed Particle Hydrodynamics (SPH) [17-18], et al).

Most of previous investigations concern on the traditional ship and offshore structures such as flat plate [19], stiffened panel [20-21], V-type wedges [22-24], spherical ball [25-26], cylindrical projectile [26] and propeller blade wedge section [27] et al. In these studies, the common perspective of water entry phenomenon is a typical FSI problem. Obviously, the elasticity of the structures is expected to influence the results of this problem (here, which calls “hydroelasticity”). Some impressive investigations concern on the role of hydroelasticity when this effect is significant. Jones may be the first person to discuss this topic [28]. After that, Berezinski [29] performed an impressive review of hydroelastic behaviour for Wagner-type impact by using explicit FEM code LS-DYNA. Then, Stenius [30-31] also illustrated the hydroelastic effects are typically characterized by a relation between loading period and natural period of vibration of the structure. Based on model experiments and numerical hydroelastic analysis results of the symmetric wedges (made by composite materials), Panciroli et al. [32-34] showed that the hydroelastic effects are negligible when the ratio between the wetting impact time and the first wet natural period is greater than 5. However, most of these hydroelastic interaction investigations are limited in simple steel structures.

<table>
<thead>
<tr>
<th>$M_s$</th>
<th>Mass of structures</th>
<th>$p_\infty$</th>
<th>Atmosphere pressure,</th>
</tr>
</thead>
<tbody>
<tr>
<td>$t_f, t_b$</td>
<td>Top (bottom) face sheet thickness</td>
<td>$A, B$</td>
<td>Empirical parameters</td>
</tr>
<tr>
<td>$H_c$</td>
<td>Height of core</td>
<td>$\rho_w$</td>
<td>Initial density of fresh water</td>
</tr>
<tr>
<td>$t_c \times t_c$</td>
<td>Section cross area of core</td>
<td>$E_w$</td>
<td>Specific internal energy per unit mass</td>
</tr>
<tr>
<td>$v$</td>
<td>Water entry velocity</td>
<td>$a_1, a_2, a_3$</td>
<td>Constants for the fluid</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>Angle between face sheet and truss</td>
<td>$b_0, b_1$</td>
<td>Constants for the fluid</td>
</tr>
<tr>
<td>$d_{c1}$</td>
<td>Core cell length</td>
<td>$\rho_a$</td>
<td>Density of air</td>
</tr>
<tr>
<td>$d_{c2}$</td>
<td>Core cell width</td>
<td>$\gamma$</td>
<td>Heat capacities of the gas</td>
</tr>
<tr>
<td>$E^*$</td>
<td>Equivalent young modulus of ILSP</td>
<td>$G^*$</td>
<td>Equivalent shear modulus of ILSP</td>
</tr>
<tr>
<td>$\rho_0$</td>
<td>Relative density of core</td>
<td>$\eta$</td>
<td>Missing percents</td>
</tr>
<tr>
<td>$C_s^G$</td>
<td>Transverse shear rigidity matrix</td>
<td>$D_f$</td>
<td>Bending stiffness of face sheets</td>
</tr>
<tr>
<td>$D_1^*$</td>
<td>Bending rigidity ILSP lattice core</td>
<td>$\nu^*$</td>
<td>Poisson ratio</td>
</tr>
<tr>
<td>$k$</td>
<td>Missing percentage parameters</td>
<td>$E_a$</td>
<td>Specific internal energy of air</td>
</tr>
<tr>
<td>$P_{\text{perfect}}$</td>
<td>Pressure for perfect sandwich panel</td>
<td>$P_{\text{imperfect}}$</td>
<td>Pressure for imperfect sandwich panel</td>
</tr>
<tr>
<td>$T_{LP}$</td>
<td>Loading period</td>
<td>$T_{NP}$</td>
<td>First wet natural period of vibration</td>
</tr>
<tr>
<td>$P_{\text{max}, L}$</td>
<td>Central FSI surface peak pressure</td>
<td>$P_{\text{max}, G}$</td>
<td>Global FSI surface peak pressure</td>
</tr>
</tbody>
</table>
More recently, the interest of sandwich structures with various lattice core topologies has grown rapidly for their significant advantages of light weight, high specific stiffness and strength [35-37]. The static and dynamic mechanical behaviour of sandwich structures with lattice truss core is quite different from other traditional constructions, including those with honeycomb core or corrugated core. In the most previous studies, the static mechanism and energy absorption behaviour of sandwich structures with lattice truss core subjected to air explosion, underwater explosion and fragment penetration are widely concerned [38-42]. Up to date, water entry characteristics of sandwich structures with lattice truss core have rarely been studied. In our previous work [43], the dynamic responses of perfect lattice sandwich panel (PLSP) under water entry are investigated. But due to the manufacturing reasons, defects in the lattices may be either in the form of vacancies, owing to missing trusses, or topological imperfections owing to displaced nodes. Thus, we shall explore theoretically and numerically the hydroelastic mechanism of imperfect lattice sandwich panel (ILSP) based on some simple cases [44].

In the present study, the hydroelasticity characteristics of imperfect lattice sandwich panel (ILSP) are further discussed. A new computational method is utilized for determination of the vibration performance of ILSP (the random moved missing and partly moved missing cases are both considered here). And these two models are validated by comparing against vibration numerical analysis. Furthermore, in order to calculate the global deflection of partly moved missing ILSP, a novel computational method is derived by adopting the simple support boundary assumption for missing area [44]. Results from this proposed model agree well with those from the 3D FSI analysis. Then, the geometric parameters effects of ILSP such as the thickness of face sheet, the relative density of core, the height of ILSP core et al. are also examined. The proposed model may be considered as initial point for modeling the design of the lattice sandwich structures subjected to water entry.

2. Geometric Characteristics of Imperfection Lattice Sandwich Panel (ILSP)

2.1 Descriptions of Imperfection Lattice Sandwich (ILSP)

In the practical cases, the imperfection of manufacture widely exists. Generally, there are three types imperfection in manufacture process including the core truss missing, initial wave less buckling of core truss, jointing invalidation between trusses and face sheet. And the core truss missing may be the most important. Thus, this type imperfection is considered here. Based on our previous studies [43-44], when the non-dimensional criteria $2\tau c f_s > 4$ (here, $2\tau c$ is the duration of water entry loading, $f_s$ is the natural vibration frequency of structure including added mass effect) the hydroelastic effect of lattice sandwich panel is not significant. In the present paper, both $2\tau c f_s > 4$ and $2\tau c f_s < 4$ are considered. Some other researchers also give similar criteria to analysis the hydroelasticity of simple structures. Considering the added mass of the first vibration modal of structure for a v-shaped ship section, a similar definition given by Stenius et al. [30-31] is the ratio $T_{LP} / T_{NP}$. Where $T_{LP}$ is the loading period based on the Wagner theory and $T_{NP}$ is the first wet natural period of vibration. It is concluded that the hydroelastic effects are negligible and the response can be set as quasi-static for $(T_{LP} / T_{NP}) > 4$. Here, by referring a practical monolithic flat panel of cruiser [43], the ILSP with an exposed area $595.1 \, \text{mm} \times 270.5 \, \text{mm}$ is considered with $2\tau c f_s \sim 5$. Besides, the ILSP with an exposed area $1200.0 \, \text{mm} \times 1000.0 \, \text{cm}$ referred a stiffened plate of bulk carrier bottom structures is proposed with $2\tau c f_s \sim 1$. 

\[35\]
2.1.1 $2\tau_5 f_s \approx 5$ (the hydroelastic effect is not significant)

<table>
<thead>
<tr>
<th>Case</th>
<th>$a$ (cm)</th>
<th>$b$ (cm)</th>
<th>$\varphi$</th>
<th>$t_f$ (mm)</th>
<th>$t_b$ (mm)</th>
<th>$H_c$ (mm)</th>
<th>$t_c \times t_c$ (mm$^2$)</th>
<th>$v$ (m/s)</th>
<th>Percent</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>59.51</td>
<td>27.05</td>
<td>1.00</td>
<td>4.219</td>
<td>4.219</td>
<td>38.25</td>
<td>6.00$\times$6.00</td>
<td>5m/s</td>
<td>0.00%</td>
</tr>
</tbody>
</table>

$\eta(C2)=0.45\%, \eta(C3)=0.90\%, \eta(C4)=1.35\%, \eta(C5)=1.80\%, \eta(C6)=3.60\%,$

$\eta(C7)=5.40\%, \eta(C8)=9.09\%, \eta(C9)=9.54\%, \eta(C10)=10.91\%, \eta(C11)=16.36\%.$

ILSP consists of two thin face sheets attached to both sides of a lightweight lattice core with the total mass of perfect lattice core sandwich plate $M_s=67.6$ kg, see [43-44]), and the geometric description of ILSP is plotted in detail in Fig. 1. Here, the geometric parameters of referenced case C1 are $\varphi=\pi/4$, $d_{c1}=d_{c2}=d_{c}=54.100$ mm, $t_f=4.219$ mm, $t_b=4.219$ mm, $t_c=6.000$ mm, $H_c=38.250$ mm (for case C1, the lattice sandwich panel is perfect, please see Table 1). Two missing imperfection types are considered in this study: the first type is randomly moved missing imperfection, and the second type is partly moved imperfection. And the missing core cell (here it means the reduction in density) percentage varies from 0.45% to 16.36% (the cases C2~C11, please also see Table 1). The boundary condition of panel can be taken as clamped. Therefore, a sealed off round structure made by rigid materials (rigid out wall, see Fig. 1) is used to model clamped boundary condition. Particularly, for each random moved missing cases C2(R) ~ C11(R), the five random types are considered here. So, there are totally 50 random moved missing cases. For partly moved missing cases C2(P) ~ C11(P), to simplify the analysis, only the central partly moved missing is considered (see Fig. 1).

2.1.2 $2\tau_5 f_s \approx 1.0$ (the hydroelastic effect is significant)

In this case, the parameters of C1’ are $\varphi=49.88^\circ$, $d_{c1}=d_{c2}=d_{c}=100.000$ mm, $t_f=1.000$ mm, $t_b=1.000$ mm, $t_c=6.500$ mm, $H_c=90.000$ mm (for case C1’, the lattice sandwich panel is perfect,
Two missing imperfection types are also considered in this study; the first type is randomly moved missing imperfection, and the second type is partly moved imperfection. The missing core cell (here it means the reduction in density) percentage varies from 0.45% to 16.36% (the cases C2’~C11’, please see Table 2). The other conditions are same as 2\(\tau f_0\approx 5\).

<table>
<thead>
<tr>
<th>a (cm)</th>
<th>b (cm)</th>
<th>(\phi^2)</th>
<th>t (mm)</th>
<th>(t_f) (mm)</th>
<th>(t_b) (mm)</th>
<th>(H_c) (mm)</th>
<th>(t_c\times t_c) (mm(^2))</th>
<th>(v) (m/s)</th>
<th>Percents</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1’</td>
<td>120</td>
<td>100</td>
<td>49.88</td>
<td>1.00</td>
<td>1.00</td>
<td>90.00</td>
<td>6.5×6.5</td>
<td>5m/s</td>
<td>0</td>
</tr>
</tbody>
</table>

\(\eta(C2’)=0.42\%, \eta(C3’)=0.83\%, \eta(C4’)=2.50\%, \eta(C5’)=4.16\%, \eta(C6’)=5.833\%, \eta(C7’)=6.67\%, \eta(C8’)=7.50\%, \eta(C9’)=8.33\%, \eta(C10’)=12.50\%, \eta(C11’)=16.70\%.

3. Mathematical Formulation and Vibration Analysis of ILSP

3.1 Basics Solution Idea of ILSP Structural Dynamics Problem

3.1.1 Basics Solution Idea of Random Moved ILSP (RM-ILSP)

As the complexity of PM-ILSP, only the elastic constants of RM-ILSP are considered here. Due to the discrete stochastic characteristics, the homogenization method is adopted. According to the concept of continue damage mechanics (CDM), the elastic constants of RM-ILSP \((E^*, G^*)\) are given as follows.

\[
E^*, G^* \sim f(\varphi)\rho_0(1-\eta)^k
\]

where \(f(\varphi)\) is a dimensionless function of \(\varphi\) (the meaning of notation \(\varphi\) sees Fig. 1), \(\rho_0\) is the relative density of RM-ILSP and \(k\) is a dimensionless constant (if \(\eta=0.00\%\) means the perfect case).

\[
E^*, G^* \sim f(\varphi)\rho_0(1-\eta)^k \approx f(\varphi)\rho_0(1-k\eta)
\]

Here, the dimensionless function \(f(\varphi)\) and the relative density \(\rho_0\) can be referred in literature,

\[
f(\varphi) = \begin{cases} 
\sin^4 \varphi & \text{for } E^* \\
\frac{1}{8}\sin^2(2\varphi) & \text{for } G^* 
\end{cases}
\]

\[
\rho_0 = \rho_s \frac{4h_c^2}{d_c^2 \sin \varphi}(1-\eta)
\]

where the meanings of \(t_c\) and \(d_c\) can be seen in Fig. 1, \(l\) is the length of core truss, and \(\rho_s\) is of base solid material for ILSP.

By combining Eqs. (1)-(4) and neglecting the high order terms in the Taylor expansions, the expression of \(E^*\) and \(G^*\) can be written as follows,

\[
E^* = \rho_s \sin^3 \varphi \frac{4h_c^2}{d_c^2} [1 - (k+1)\eta], \quad G^* = \rho_s \frac{1}{2} \frac{\sin^2(2\varphi)}{d_c^2 \sin \varphi} \frac{h_c^2}{d_c^2} [1 - (k+1)\eta]
\]
In the previous study, Wallach and Gibson [45] has given an observation that Young’s modulus of lattice materials decreases linearly with increasing fraction of missing trusses, with a roughly 16.6% decrease in modulus for every 10% reduction in relative density. Thus, we assume that the value of $k$ is set to be 0.66 here. If the equivalent elastic constants $E^*$ and $G^*$ are calculated, the response of RM-ILSP is easy to obtained based on our previous study [43-44] which is a classic structural dynamic problem in linear region.

3.1.2 Basics Solution Idea of Partly Moved ILSP (PM-ILSP)

In this section, the boundary condition of partly moved ILSP (PM-ILSP) is analyzed. If the missing percentage of core truss is relative large, the above method will not be suitable. Thus, the local boundary effect must be considered. Based on the previous study [43-44], it shows that the core truss can be set as supported by rows of equidistant trusses or simply supported edges. To simplify the analysis, the shape of partly moved area is set to be rectangular, making the calculation of the vibration and static behavior of local partly moved area much easier.

3.2 Vibration Verification Analysis of ILSP

If the equivalent elastic properties of random moved ILSP are obtained from Eqs. (1)-(5), the fundamental bending vibration frequencies of RM-ILSP are calculated by three steps [43] as follows. The first and second step is to calculate the equivalent shear (bending) rigidity of ILSP lattice core. And the last step is to calculate the natural frequency of RM-ILSP. In this key step, an approximate method [43-44, 46-47] is adopted to analyze the fundamental vibration frequency. And the detailed expressions of the formulations are shown as follows.

Step 1: calculate the equivalent shear rigidity of RM-ILSP lattice core [43]

$$C_G^* = \frac{H_e + (t_f + t_b) / 2}{H_e} \begin{bmatrix} C_{55}^H & C_{54}^H & C_{45}^H & C_{44}^H \end{bmatrix}$$

where $C_G^*$ is the transverse shear rigidity matrix of ILSP core (the detailed expressions of the homogenized module of lattice core $C_{\alpha\beta}^{\mu H}$ ($\alpha, \beta=4, 5$) can be found in literature [43]).

Step 2: calculate the bending rigidity ILSP lattice core [43]

$$D_1^* = \frac{E^* \left[ \left( H_e + (t_f + t_b) \right)^2 - H_e^3 \right]}{12 \left( 1 - \nu^2 \right)} \times \left[ 1 - \frac{3(t_f - t_b)^2}{\left( \frac{H_c}{H_e + (t_f + t_b)} + \frac{H_c(t_f + t_b)}{H_e} \right)^2 \left( t_f + t_b \right)^2} \right]$$

Here, for the thin face sheets of ILSP, only bending deformation is considered (the bending stiffness $D_f$ of face sheets is included).

Step 3: calculate the natural frequency of ILSP [43].

The third step is to calculate the natural frequency of ILSP. In the present investigation, an approximate method is adopted to analyze the fundamental vibration frequency of the
ILSP. And the detailed steps of solving approximate equations can be found in our previous study [43] and classic literature [47].

Table 3 Vibration analysis of imperfect lattice sandwich plate

<table>
<thead>
<tr>
<th>$\eta$</th>
<th>1st (Analytical/FEM) Error</th>
<th>2nd (Analytical/FEM) Error</th>
<th>3rd (Analytical/FEM) Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00%</td>
<td>1691.1 1553.9 8.83% 1966.9 1858.9 5.81% 2421.2 2289.2 5.77%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.45%</td>
<td>1660.3 1552.2 6.96% 1943.7 1862.2 4.38% 2411.3 2275.1 5.99%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.90%</td>
<td>1654.1 1547.3 6.90% 1936.9 1851.6 4.61% 2408.2 2268.3 6.17%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.35%</td>
<td>1647.8 1541.9 6.87% 1930.1 1841.0 4.84% 2399.9 2262.0 6.10%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.80%</td>
<td>1641.6 1528.1 7.43% 1923.2 1838.0 4.64% 2391.5 2256.4 5.99%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.60%</td>
<td>1616.3 1523.2 6.11% 1895.6 1827.6 3.72% 2367.8 2235.9 5.90%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.40%</td>
<td>1590.7 1518.3 4.77% 1867.5 1799.3 3.79% 2333.5 2226.2 4.82%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9.09%</td>
<td>1538.2 1487.2 3.43% 1809.6 1746.6 3.61% 2262.6 2171.4 4.20%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9.54%</td>
<td>1531.5 1487.0 2.99% 1802.2 1746.0 3.22% 2253.6 2169.8 3.86%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10.91%</td>
<td>1511.3 1483.5 1.87% 1779.8 1741.5 2.20% 2226.1 2155.8 3.26%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>16.36%</td>
<td>1427.4 1423.0 0.31% 1686.4 1681.9 0.27% 2111.4 2113.4 -0.09%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3 continued...

(b) Partly moved missing case

<table>
<thead>
<tr>
<th>$\eta$</th>
<th>1st (Analytical/FEM) Error</th>
<th>2nd (Analytical/FEM) Error</th>
<th>3rd (Analytical/FEM) Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00%</td>
<td>1691.1 1553.9 8.83% 1966.9 1858.9 5.81% 2421.2 2289.2 5.77%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.45%</td>
<td>1660.3 1553.4 6.88% 1943.7 1851.2 5.00% 2411.3 2288.5 5.37%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.90%</td>
<td>1654.1 1552.2 6.56% 1936.9 1847.7 4.83% 2408.2 2268.7 5.31%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.35%</td>
<td>1647.8 1549.1 6.37% 1930.1 1833.6 5.26% 2399.9 2282.1 5.16%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.80%</td>
<td>1641.6 1544.7 6.27% 1923.2 1832.1 4.97% 2391.5 2183.5 9.53%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.60%</td>
<td>1616.3 1474.2 9.64% 1895.6 1776.1 6.73% 2367.8 1900.9 24.56%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.40%</td>
<td>1590.7 1374.2 15.75% 1867.5 1738.4 7.43% 2333.5 1818.1 28.35%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9.09%</td>
<td>1538.2 1148.4 33.94% 1809.6 1661.2 8.93% 2262.6 1738.2 30.17%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>9.54%</td>
<td>1531.5 1145.7 33.67% 1802.2 1660.6 8.53% 2253.6 1705.5 32.14%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10.91%</td>
<td>1511.3 1074.9 40.60% 1779.8 1192.6 49.24% 2226.1 1663.2 33.84%</td>
<td></td>
<td></td>
</tr>
<tr>
<td>16.36%</td>
<td>1427.4 681.75 109.37% 1686.4 1087.9 55.01% 2111.4 1314.0 60.68%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 2 Vibration of partly moved imperfect lattice sandwich plate

The vibration results for these two imperfection types are listed in Table 3. Compared with the FEM results in Table 3.a, the approximate analytical method can give a good prediction when the missing ratio is between 0.45%~16.36%, where the maximum error is less than 9.0%. For partly moved missing type, the vibration characteristics are complex (Table 3). The FEM results illustrated in Table.3.b show that the prediction method (Eqs. (1)-(7)) cannot reflect the vibration characteristics when the missing ratio is greater than 1.8%. And this is mainly caused by the local vibration mode due to the partly moved area. The
whole vibration mode seems a mixture of local-global vibration interaction (please see Fig.2). In practical, due to the complexity of boundary condition, the precision of analytical model is very difficult to achieve. As an initial estimation, the local vibration mode of missing area can be firstly calculated based on the model of section 3.1.2.

4. Engineering Prediction Model of Structural Deflection and Discussion

In this section, a novel engineering model is performed to investigate the deformation characteristics of ILSP when the structural deformation is in linear region as shown in Fig. 3. The total deformation field can be divided into two components:

1) The global bending deformation field of ILSP ($w_G$).

2) The local bending deformation field of each cell ($w_L$).

\[ w = K_1(t)w_G + K_2(t)w_L \]  
\[ (8) \]

where $K_1$ and $K_2$ are dynamic coefficient for total and local deformation respectively, when the dynamic effect is considered. And the calculation of dynamic deflection in linear elastic range under uniform impact pressure is given by following equation,

\[ K(t) = \frac{w_{\text{dynamic}}(t)}{w_{\text{static}}} = \frac{\omega}{p_{\text{max}}} \int_0^t p(\tau)\sin[\omega(t - \tau)]d\tau \]  
\[ (9) \]

where $K(t)$ is the dynamic inertial effect factor function varied with time, Substitute Eq. (9) into Eq. (8), it has,

\[ w = \frac{w_G \omega_G}{p_{\text{max},G}} \int_0^t p_G(\tau)\sin[\omega_G(t - \tau)]d\tau + \frac{w_L \omega_L}{p_{\text{max},L}} \int_0^t p_L(\tau)\sin[\omega_L(t - \tau)]d\tau \]  
\[ (10) \]

where $\omega_G$ is the first circular frequency of global ILSP, $\omega_L$ is the first circular frequency of ILSP local cell, $\omega_G$ can be calculated from Eqs. (1)-(7). The method of calculation for $\omega_G$ can be referred in [43].
In the water entry problem, generally, the compressibility effects can be divided as two parts. The first part of compressibility effects is the compressibility of water, while the second compressibility effect is the compressibility of air respectively. Some previous authors have included the effects of compressibility on the flow during impact [48-50]. The water entry bodies can be divided as three types, which are plate body, blunt body and pointed body. For flat or nearly flat bodies, the hydroelasticity of plate and the compressibility of air cushion should be included. For blunt body, the compressibility of liquid should be included in the first instants as the flow is accelerated from rest. After the fluid particles have reached the velocities associated with the subsonic stage of water entry compressibility can be ignored. As pointed out by Korobkin & Pukhnachov [48-49], there are many difficulties to describe and solve the water entry mathematical model, such as the multiphase, the fluid-solid interaction and the nonlinearity et al. One of these key difficulties is the compressibility of air cushion. To avoid this difficulty, based on the assumption the water impact pressure \( p_{\max} \) is the peak impulsive pressure for FSI surface which can be calculated in the following,

\[
p_{\max} - p_\infty = Av^B
\]  

where \( p_\infty \) is atmosphere pressure, \( A \) and \( B \) are also the empirical parameters. If the logarithm is used to the Eq. (10) left and right, we can obtain,

\[
\ln (p_{\max} - p_\infty) = \ln A + B \ln v
\]  

We see that the logarithm value of \( (p_{\max} - p_\infty) \) is linear with the logarithm value of water entry velocity \( v \). To calculate the water impact \( p_{\max} \), the two unknown parameters in the Eq. (12) are \( A \) and \( B \). And the parameters \( A \) and \( B \) can be determined from a least square linear fit based on a few numerical simulations or experiments. The values of \( A \) and \( B \) are constants though the range of water entry pressure varies widely. It must be pointed out that the coefficients \( A, B \) in Eq. (12) are different for the average (whole) FSI pressure and the central FSI pressure, respectively. For example, from our previous study [43], the performance of central FSI pressure for perfect lattice sandwich panels follows a linear trend, as determined from a least square linear fit,

\[
\ln (p_{\max,L} - p_\infty) = 4.05 + 1.44 \ln v
\]  

By means of the least square linear fit, the average (whole) FSI pressure for perfect lattice sandwich panels is also determined,

\[
\ln (p_{\max,G} - p_\infty) = 2.63 + 1.69 \ln v
\]  

The dynamic coefficient \( K \) is only associated with the fundamental circular frequency \( \omega \) and the impact duration time \( 2\tau_s \). For the average (whole) FSI pressure, a simplified expression to define the impulsive shape \( p_G(t) \) in Eq. (10) as following [43],

\[
p_G(t) = \begin{cases} 
\frac{\lambda_1G}{t^2} + \lambda_2G, & 0 \leq t \leq 2\tau_{s,G} \\
0, & 2\tau_{s,G} \leq t 
\end{cases}
\]  

where

\[
\lambda_1G = -\frac{p_{\max,G}}{\tau_s^2}, \quad \lambda_2G = \frac{2p_{\max,G}}{\tau_s}, \quad 2\tau_{s,G} \approx (2.1 \sim 2.6) \sqrt{\frac{a^2 + b^2}{C_0}}
\]  

where \( C_0 \) is the sound speed of air (\( C_0 = 340 \) m/s).

Similarly, for local FSI pressure, it has,
where

\[ \lambda_{1L} = -\frac{p_{\max,L}}{\tau_s^2}, \quad \lambda_{2L} = \frac{2p_{\max,L}}{\tau_s}, \quad 2\tau_{s,L} \simeq \frac{(2.1 \sim 2.6) \sqrt{a^2 + b^2}}{C_0} \]  

The last two unknown parameters \( w_G \) and \( w_L \) can be calculated from following,

\[ w_G = 4p_{\max,G}b^4 \sum_{m=1,3,\ldots} \left[ \frac{1}{m^2 \pi^2 (\alpha^2 + n^2)^2} - \frac{\delta_m}{m^2 \pi} \left\{ \frac{1}{\alpha} - \frac{\pi^2}{\alpha \eta_n^2 + \frac{m^2 \pi^2}{\alpha}} \right\} \right] G_n \]

\[ + \frac{1}{m^2} \left[ (\alpha_n th(\alpha_n) + 1) th(\alpha_n) - \alpha_n \right] p_{\max,G} = 0 \]

\[ \alpha_m - (\alpha_m th(\alpha_m) - 1) th(\alpha_m) + 2\delta_m \left\{ th(\alpha_m) - \frac{m\pi}{\eta_m} th(\alpha_m) \right\} H_m \]

\[ + \frac{1}{\alpha^2 n^2} \left[ (\alpha_n th(\alpha_n) + 1) th(\alpha_n) - \alpha_n \right] p_{\max,G} = 0 \]

\[ \alpha_m = \frac{m\pi b}{2a}, \quad \eta_m = \left[ m^2 \pi^2 + \frac{2}{\delta a (1-\nu)} \right]^{1/2}, \quad \delta_m = \frac{D_1}{C_g a^2}, \quad \delta_n = m^2 \pi^2 \delta_a \]

\[ \alpha_n = \frac{n\pi a}{2b}, \quad \eta_n = \left[ n^2 \pi^2 + \frac{2}{\delta_b (1-\nu)} \right]^{1/2}, \quad \delta_b = \frac{D_1}{C_g b^2}, \quad \delta_n = n^2 \pi^2 \delta_b \]
\[ w_L = 0.005507 \frac{p_{\text{max},L} D^*_c}{D^*_1} \]  

where \( C^*_G \) is the equivalent shear stiffness of lattice core, \( D^*_1 \) are the bending stiffness of ILSP, \( w_L \) is the local maximum deformation, \( p_{\text{max},L} \) is the maximum value of water entry pressure loading on central cells, \( d_c \) is the width of cell (see Fig. 1) and \( D_f \) is the flexural rigidity of cell face sheet. And the other nomenclatures have the same meanings as mentioned in the previous study.

\[ D_f = \frac{E t^3}{12(1-\nu^2)} \]  

If the detailed expression of \( \omega_G, \omega_L, w_G, w_L, p_{\text{max}}, p(t) \) and \( 2\tau_e \) are obtained from Eqs. (1)-(25), the value of dynamic coefficient \( K \) can be easily obtained. And the maximum value of \( K(t) \) at \( t = t_0 \) is commonly considered, which satisfies the following conditions \[43, \]

\[ \frac{dK(t)}{dt} = 0 \bigg|_{t=t_0}, \quad \frac{d^2K(t)}{dt^2} < 0 \bigg|_{t=t_0} \]  

5. Fluid-Structures Interaction Numerical Simulation

5.1 Numerical Analysis Model

![Numerical simulation model](image)

Fig. 4 Numerical simulation model

The top (bottom) face sheet and corrugated core are modeled as a plane plate using 52000 quadrilateral shell elements (shell 163, KEYHOFF formulation, hourglass control, five degrees of freedom per node \( u_x, u_y, u_z, \theta_x, \theta_y \), finite membrane strains elements, with 5 integration points). The detailed Lagrange FE model of ILSP can be seen in Fig. 4. In the simulation, the Lagrange material of the ILSP is modeled to be S304 steel. In order to consider the strain rate effect, Cowper-Symonds model is adopted. And the material property constants can be referred in [43-44].

To model the fluid inside and outside the ILSP, two Euler domains are used. The outer domain has the ILSP surface (including top face sheet, bottom face sheet and out-off rigid wall) as part of the fluid boundary, Euler material is outside the ILSP surface and there is no material inside the ILSP surface. The contents inside the ILSP are modeled in the inner domain and this domain is also enclosed by the ILSP out surface. Therefore both Euler domains use the ILSP surface as part of their enclosure. Here, the outer boundary of the outer domain is given by a sufficiently large fixed box. Here, pressure at the water Euler domain is set to the hydrostatic pressure by using the *INITIAL_HYDROSTATIC_ALE keyword cards. The Euler mesh contains the water and the air on the top of the water.
The fluid mesh used for this problem are consisted of a block of elements, with the dimensions are 2.5m*2.5m*4.0m. And this fluid block of water and air are meshed with 100*100*90 hexahedron elements, for a total of 900000 fluid elements (Fig. 4). The finest grid size of Euler domain is 0.005 m in this simulation. All the boundary conditions for our fluid mesh are given a "flow" boundary condition by adopting non-reflecting boundary keyword card. In order to model fresh water, a polynomial equation of state was conducted. This state equation (EOS) of fresh water relates the pressure in the fluid to the acoustic condensation $\mu$ and the specific internal energy $E$ by:

$$P = \begin{cases} a_1 \mu + a_2 \mu^2 + a_3 \mu^3 + \left( b_0 + b_2 \mu + b_3 \mu^2 \right) \rho_0 E_w (\mu > 0) \\ a_1 \mu + \left( b_0 + b_1 \mu \right) \rho_0 E (\mu < 0) \end{cases}$$

(27)

where $\mu = (\rho - \rho_w) / \rho_w$, $\rho_w$ is the initial density of fresh water, $E_w$ is the specific internal energy per unit mass, and $a_1$, $a_2$, $a_3$, $b_0$, $b_1$ are constants for the fluid. And the upper equation applies to a fluid in a compressed state, while the lower applies to a fluid in an expanded state. The constants for this equation for fresh water are provided in literature [43-44].

The gamma law gas model is adopted for the EOS of air,

$$P = \left( \gamma - 1 \right) \rho_a E_a$$

(28)

where $\rho_a$ is the density of air, $\gamma$ is the heat capacities of the gas and $E_a$ is the specific internal energy of air. The initial pressure of air is set to $1.0 \times 10^5$ Pa.

In this analysis, the Arbitrary Lagrange-Euler (ALE) Coupling algorithm is used. The Lagrangian and Eulerian meshes are geometrically independent, and interact via coupling surface. The coupling surface 'cuts across' Eulerian elements which contain multi-material air and water, changing their volume and surface areas. As the finite element mesh deforms under the action of the impact pressure from the Eulerian mesh, the resulting FEM deflection then influences subsequent material flow and pressure forces in the Eulerian mesh, resulting in automatic and precise coupling of FSI. In general, it is known that the results based on ALE algorithm are sensitive to the Euler mesh density. So the Euler mesh needs to be fine enough to capture the highest gradients in the pressure fields, yet a coarser mesh is favorable in terms of computational cost. In addition, the selection of the contact stiffness in penalty is based on contact algorithm, it is required the maximum pressures are approximately known ahead. The non-physical contact penetration can be controlled. To verify the reliability of the developed finite element model, compared with experimental results, the water entry analyses of a circular monolithic plate are illustrated in our previous study [19, 43].

5.2 Basic Water Entry Mechanism and Impact Pressure

5.2.1 Basic Water Entry Mechanism

As an illustration, the case of C1’ is analyzed here. The entire water entry process can be divided into three stages as follows.

- First Stage; Structure begins to drop → Air compression begins;
- Second Stage; Air compression begins → Structure begins to contact water;
- Third Stage; Structure begins to contact water → Structure immerses in water.

In this case, the duration predicted by numerical simulation $2\tau = 11.5$ ms coincides with the empirical estimations ($2\tau = 10.7$ ms) from Eqs. (11)-(12). The contacting water time $t$ observed by the numerical simulation is $\sim 19.0$ ms. The result from the simulation shows that when $t > 25.0$ ms, the water jet begins to split into two parts as plotted Fig.7. Furthermore, the shape of the water entry jet calculated by the ALE coupling algorithm cannot capture the spatter of the droplets. From the authors’ view, this phenomenon is probably caused by the
instability of fluid flow. Some small perturbations may be from some uncontrollable practical factors, such as the minor asymmetries in the simulation.

Fig. 5 The basic water entry process of ILSP

Fig. 6 Volume distribution isosurface of water entry process
From the simulation results, it is obtained that the global air cushion is significant from Fig. 5 (please see the black dash line). Due to the global elastic deformation, the shape of this air cushion is not rectangular. And the fluid materials volume distribution isosurface also shows the local air cushion exists during the initial water entry duration (Fig. 6). This local cushion is caused by lattice cell local deformation which has significant effects on the water entry pressure and dynamic responses of ILSP. Because of the existence of this local air cushion, which plays a role as a buffer device, the peak value of average FSI pressure of contacted-water wet face sheet is lower than that of monolithic plate which has been certificated in our previous study [43-44].

As the effects of local air cushion, the local hydroelasticity is important in the dynamic responses of ILSP. Due to the local loading, the high frequency responses of pressure are also induced as shown later in the pressure-time history curve (please see in Fig. 8). And this local high frequency response is more significant if the loss imperfection is more seriously (please see in Fig. 9). So, the whole response of ILSP is the combination of the multiple global deformations, the lattice cell local deformations, the air cushion and the free surface interaction. Furthermore, some other strong nonlinear phenomenon such as flow turbulence and multiphase flow mixture et al. phenomenon may also exist which cannot captured by present analytical solution and numerical simulation. And it must be studied by future more finely experimental measurement of micro-scale flow structures in FSI surface.

5.2.2 Impact Pressure Results for 2fs~5 (the hydroelastic effect is not significant)

The relation between core truss missing ratio and water entry pressure is shown in Fig. 7 for RM-ILSP and PM-ILSP when water entry velocity is 5m/s. According to the results as plotted in Fig. 7, by means of the least square fit, the following observations can be drawn.
For RM-ILSP:

\[ P_{\text{imperfect}} = P_{\text{perfect}} \left( 1 + 0.31\eta \right) \quad \text{for center} \]  
\[ P_{\text{imperfect}} = P_{\text{perfect}} \left( 1 - 0.45\eta \right) \quad \text{for whole FSI Surface} \]  

For PM-ILSP:

\[ P_{\text{imperfect}} = P_{\text{perfect}} \left( 1 - 0.57\eta \right) \quad \text{for center} \]  
\[ P_{\text{imperfect}} = P_{\text{perfect}} \left( 1 - 0.81\eta \right) \quad \text{for whole FSI Surface} \]

![Fig. 8 Comparison of perfect and imperfect water entry pressure curves for RM-ILSP](image)

The local FSI pressure at the center point increases slowly (<4%) when the missing ratio varies from 0.45% to 16.36%. But the water entry pressure of total FSI surface for RM-ILSP decreases linearly with fraction of missing trusses, with a roughly 4.5% (from 212KPa to 195KPa) respectively in pressure for every 10% reduction in relative density. This is mainly because the core of RM-ILSP becomes softer when \( E^* \), \( G^* \) are smaller. Compared with the Eq. (29) and Eq. (31), it shows that the changing trend of missing ratio is different between RM-ILSP and PM-ILSP. This phenomenon reveals that the random missing truss could change the space distribution of water entry pressure. And the local area of RM-ILSP may be becoming ‘harder’. This conclusion also certificated from Fig.7 (a) which shows the discrete pressure value of central point. And the maximum difference between the lower point and upper point value is about 80KPa (~15%). This gives a conclusion that the core topology
structure is very important in the FSI. The least square fit data also illustrate that the decrease changing trend of PM-ILSP is faster than that of RM-ILSP. Compared with Fig. 8 (a) and Fig. 9 (a), the pressure-time curve shows that the oscillation amplitude of PM-ILSP is greater in the whole water entry process. The same conclusion of total FSI pressure also can be summarized from Fig. 8 (b) and Fig. 9 (b).

5.2.3 Impact Pressure Results for $2\tau_f \sim 1.0$ (the hydroelastic effect is significant)

Fig. 10 Relation between missing percentage and water entry pressure (2$\tau_f \sim 1$)

For RM-ILSP:

\[ P_{\text{imperfect}} \approx P_{\text{perfect}} \quad \text{for center} \quad (33) \]
\[ P_{\text{imperfect}} = P_{\text{perfect}} \left(1 - 0.221\eta\right) \quad \text{for whole FSI Surface} \quad (34) \]

For PM-ILSP:

\[ P_{\text{imperfect}} \approx P_{\text{perfect}} \quad \text{for center} \quad (35) \]
\[ P_{\text{imperfect}} = P_{\text{perfect}} \left(1 - 0.638\eta\right) \quad \text{for whole FSI Surface} \quad (36) \]

When the hydroelastic effect is significant, the numerical simulation results reveal that the random missing imperfection effect on the dynamic response of RM-ILSP is complex as shown in Fig. 10 (a). When the missing percentage $\eta$ is less than 5%, the local FSI water entry pressure seems rise up. But when $\eta > 5\%$, the local FSI pressure is becoming lower while this oscillation amplitude is minor. The same phenomenon also can be found in the response of PM-ILSP. According to the numerical results in Fig. 10 (b), the changing trend is decreasing
about 9.8% for whole FSI surface when $\eta=16\%$. Compared with Figs. 10 (a) and (b) (here, also compared with Eqs. (34) and (36)), an approximate linear decreasing relation between missing rate and pressure value can be given.

![Fig. 11 Comparison of perfect and imperfect water entry pressure curves for RM-ILSP](image1)

![Fig. 12 Comparison of perfect and imperfect water entry pressure curves for PM-ILSP](image2)

In Fig. 11 and Fig. 12, the water entry pressure time history curves are given for local coordinate and whole surface. Compared the water entry curves detailed, the random missing imperfection also has little effect on dynamic response. But the effect of local partly missing imperfection is more significant (Fig. 12). Compared with Fig. 9 and Fig. 12, both for local central point pressure and total FSI surface pressure, the smoothness of pressure curve is more significant when $2\tau_{fs} \sim 1.0$. This may the high frequency pressure response components would be excited by smaller core cell.

5.3 Structural Deflection Response of ILSP

5.3.1 $2\tau_{fs} \sim 5$ (the hydroelastic effect is not significant)

For RM-ILSP, compared with FSI simulation results, with water entry speed $v=5$ m/s, the maximum deformations by using above FE analysis model are illustrated in Fig. 13. It is clear that the changing trend of simulation result is consistent well with theoretical predictions when the reduction of relative density is lower than 10–11%. But the error of this analytical method is greater than 15% when the missing percentage is greater than 12%. This is mainly because the precision of first homogenization approximation in Eq. (4) is not precise enough when the reduction of relative density becomes sufficiently high. Another reason is probably the local deflection of imperfection cell. For PM-ILSP, a detailed comparison analysis in Fig.
shows that the maximum deflection has a rapid increase when \( \eta = 9-10\% \). Thus, this result indicates that there exists a critical missing ratio point.

\[ \text{Fig. 13 Relation between missing percentage and water entry pressure (2}\tau_sf~5) \]

\[ \text{(Left: RM-ILSP, right: PM-ILSP)} \]

5.3.2 \( 2\tau_sf~1.0 \) (the hydroelastic effect is significant)

As shown in Fig. 14, the critical deformation point also exits when hydroelastic effect is significant both for RM-ILSP and PM-ILSP. For RM-ILSP, the critical point is nearly equal to 12%. But for PM-ILSP, the critical point is nearly equal to 2%. The computational results reveal that the maximum deformation is increasing very fast (~300%) for PM-ILSP when the missing percentage is about 1%. This means the maximum deformation is insensitive with imperfection ratio. Compared with Fig.13 and Fig. 14, the different critical deformation point exists when hydroelasticity indicator \( 2\tau_sf \) is different. For RM-ILSP cases, the critical deformation point of \( 2\tau_sf~1.0 \) (about 12%) is very close to that of \( 2\tau_sf~5.0 \) (about 10~11%). But for PM-ILSP cases, the critical point between \( 2\tau_sf~1.0 \) (about 2%) and \( 2\tau_sf~5.0 \) (about 10~11%) is very variable. This may the effects of different missing trusses region.

\[ \text{Fig. 14 Relation between missing percentage and water entry pressure (2}\tau_sf~1) \] (Left: RM-ILSP, right: PM-ILSP)

6. Geometric Parameters Effects Study

When the hydroelastic effect is not significant (\( 2\tau_sf~5 \)), the effects of parameters can be referred our previous investigation [44]. Thus, only \( 2\tau_sf~1 \) is considered here. Both for RM-ILSP and PM-ILSP, three parameters including the face sheet thickness \( t_b \) (\( t_f \)), the height \( H_c \) and the section cross area \( t_c^*t_c \) are considered here.
6.1 The face sheet thickness $t_b$ ($t_f$)

For RM-ILSP, as plotted in Fig. 15, when the top face sheet thickness $t_f$ varies from 0.5mm to 5.5mm, the central FSI pressure $P_c$ rises about 11% even $\eta=12.50\%$. Furthermore, when $t_f$ is greater than 0.5mm, the $P_c$ is nearly about 550KPa. The whole FSI surface pressure $P_w$ rises about 12% when $t_f<1.5$mm. But when $t_f>1.5$mm, the $P_w\sim370$KPa. Moreover, when $t_f$ changes, the characteristic of maximum deformation is given in Fig. 16. When $t_f$ varies from 0.5mm to 5.5mm, the deformation value becomes lower dramatically as the bending stiffness increases fast. Particularly, this trend is not significant when $t_f>2.5$mm. Compared with $t_f$, the effect of $t_b$ is significant (Fig. 17). As plotted in Fig. 17, $P_c$ increases fast when $t_f<2.5$mm while this trend is becoming slow if $t_f>2.5$mm. Thus, the maximum deformation decrease fast when $t_f<2.5$mm (Fig. 18). The effects of face sheet thickness $t_b$ ($t_f$) for perfect lattice sandwich panel has been investigated by many previous scholars in other field such as bending, vibration and blast et al [38-41], some similar conclusions have been given.

For PM-ILSP, if $\eta=12.50\%$, the central FSI pressure rises $P_c$ about 3% when the top face sheet thickness $t_f$ varies from 0.5mm to 5.5mm, while the FSI pressure $P_c$ increases 5%. That means the change of $t_f$ has little effect on the water entry pressure. And this trend also exits in the analysis of deformation. This is mainly because the majority of deformation is local deformation for water contacted surface. Though the bending stiffness increases fast, the global deformation has little effect on whole deformation. Compared with Fig. 16 and Fig. 19, the effect of $t_f$ is more significant for RM-ILSP cases. We think this may the average of random missing plays a key role in the response which makes ILSP core softer. While the central effect of local missing is more important for PM-ILSP cases.
6.2 The core height $H_c$

For RM-ILSP, $H_c$ has significant effect on the dynamic behavior. As shown in Fig. 20, the water entry FSI pressure is decreasing about 40% when $H_c$ increases from 70.00 to 110.00 mm. And the changes for the deformation of ILSP is dramatically (Fig. 21) from 50.0 mm to 5.0 mm. But this trend becomes slow when $H_c > 90.0$ mm. This can be interpreted as the local buckling phenomenon occurs when the $H_c$ is relative high. Compared with Fig. 20(a) and Fig. 20(b), the decreasing ratio of central pressure value and total FSI pressure value is nearly the same (~30%). The minimum deflection critical point in Fig. 21 shows that the optimization is needed in the parameter design of ILSP. Moreover, it shows that the effects of $H_c$ on the changing trend of water entry pressure is more dramatically than that of $t_b(t_f)$. And the conclusion is also suited for the effects of deflection characteristics.

For PM-ILSP, the changing trend of dynamic behavior curve is similar with RM-ILSP. When $20.0 \text{ mm} < H_c < 40.0 \text{ mm}$, the water entry pressure decreasing trend is more dramatically...
(nearly ~50%) as shown in Fig. 22. Furthermore, the changing trend seems more complex when 40.0 mm<\(H_c<60.0\) mm. We think the basic reason for this phenomenon is may be the changing of pressure spatial distribution. For the characteristics deformation of PM-ILSP, we find it decreases dramatically when \(H_c<90.0\)mm (Fig. 23). This may be caused by two reasons. The first is the deceasing of water entry pressure; the second is the increasing of bending stiffness. Compared with Fig. 21 and Fig. 23, it seems minimum deflection critical point is not existed for PM-ILSP.
6.3 The section cross area $t_c^*t_c$

For random moved ILSP, the last parameter is the section cross area $t_c^*t_c$. According to the numerical simulation results, the changing characteristic is rather different when $t_c$ increases from 5.0mm to 7.0mm. As shown in Fig. 24, the central water entry pressure $P_c$ is decreasing fast when $t_c<6.0$mm. But the changing trend is becoming increasing when $t_c>6.0$mm. This means the water entry pressure also has a critical point when $t_c$ changes. This may be caused by the non-uniform distribution of water entry pressure due to local stiffened truss. For global FSI pressure $P_w$, the changing trend of is increasing fast when $t_c>6.5$mm. This increasing trend is slow when $t_c<4.5$mm. Compared in Fig. 24, the changing trend between $P_c$ and $P_w$ is different. As we known, the pressure $P_c$ is a local indicator of pressure distribution; while the pressure $P_w$ is a global indicator of pressure distribution. The structural deformation of ILSP is also sensitive when $t_c$ is changing. As shown in Fig. 25, the maximum deformation is decreasing fast when $t_c$ changes from 2.5mm to 6.5mm. From the numerical simulation, the local truss buckling occurs when $t_c$ is about equal to 2.5mm.
4.5 5.0 5.5 6.0 6.5 7.0 7.5
tc (mm)

540
570
600
630
660

54
57
60
63
66

5.4
5.7
6.0
6.3
6.6

4.5
5.0
5.5
6.0
6.5
7.0
7.5
tc (mm)

180
190
200
210
220

For PM-ILSP, compared with Fig. 24 and 26, the changing trend of water entry pressure for central point is different with that of RM-ILSP. For the central pressure $P_c$, the water entry pressure is decreasing slowly from 650KPa to 540KPa. But the $P_w$ is increasing about 30KPa (Fig. 26). Compared with the effect of $H_c$, it shows that the effect of $t_c^*$ is minor. Thus, in the parameter design of ILSP, the section area can be seen as a secondary consideration. As plotted in Fig. 27, the changing trend of deformation curve also concludes that the $t_c$ has effect on the dynamic response of PM-ILSP, which is similar with that of RM-ILSP. The central deformation of PM-ILSP is decreasing about 10% when $t_c$ is increasing from 5.0mm to 7.0mm.

7. Conclusion

In this study, the hydroelastic behavior of lattice sandwich panels with imperfection (ILSP) subjected to water entry is investigated both analytically and numerically. A novel engineering estimation method is proposed to calculate the dynamic response characteristics of ILSP. The FSI numerical model is presented to obtain the detailed water entry behavior for ILSP. To identify the effect of missing ratio on the water entry characteristics, various imperfection scenarios are simulated. Finally, the effects of key parameters such as the face sheet thickness, core height and section of core truss are discussed. This study provides an engineering method for water entry problem of sandwich panels how hydroelastic pressure and maximum structural deformation are affected by the missing imperfection during the manufacture. And the following conclusions could be drawn from the semi-analytical method and numerical simulation results.

(1) When the hydroelastic effect is not significant ($2\tau_{fs}\sim 5$), the random missing imperfection has little effect on water entry pressure of local central point if the missing percentage varies from 0.45%-16.36%; though a more significant effect on the whole water
entry pressure. When the core of ILSP becomes softer, the water entry pressure of central point will decrease by about 10%.

(2) When the hydroelastic effect is significant ($2\tau_0 f_s^f \sim 1$), the random missing imperfection also has little effect on water entry pressure of local central point if the missing ration is varied from 0.45%-16.36%. With the increasing of missing percentages, the water entry pressure decreases. An oscillation phenomenon is identified in the water entry pressure curve. And the water entry pressure of central point decreases about 6.4% when the missing percentage is 16.36%.

(3) The results obtained by engineering estimation model agree well with that of numerical simulation when the imperfection type is random missing and the missing percentage is lower than 12%. But if the missing percentage is greater than 12%, the error becomes larger. And this is mainly caused by the first order approximation in the assumption.

(4) A very interesting phenomenon is that a critical missing percentage exists both for random missing and partly moved imperfection types. If the missing percentage is greater than the critical one, the maximum deformation increases dramatically. Thus, in the engineering practice, the missing percentage must be limited in the critical region which must be emphasized.

(5) The parametric study results show that the thickness of water contacted face sheet, the core height and the section of core truss has significant effect on the water entry response characteristics. If the core truss buckling occurs in the water entry process, this effect is more significant. On the contrary, the top face sheet has neglected effect on the response of ILSP.

Acknowledgement

The authors are pleased to acknowledge the financial support of the National Natural Science Foundation of P.R. China (Contract No. 51079058, 51279065) and Basic Defense Research Program (Contract No. JCKY201606C003).
REFERENCES


Davide CHICHI  
Yordan GARBATOV

http://dx.doi.org/10.21278/brod70205

ISSN 0007-215X  
eISSN 1845-5859

RETROFITTING ANALYSIS OF TANKER SHIP HULL STRUCTURE SUBJECTED TO CORROSION

UDC 621.78.019.84: 004.413.4: 629.5.018.4

Original scientific paper

Summary

The objective of the study presented here is to investigate the efficiency in recovering the structural capacity of a double bottom side girder plate of an oil tanker, accounting for the probability of failure and cost associated with the retrofit or substitution of the plate. The side girder includes a manhole shape opening, and it is subjected to a uniaxial compressive load and random non-uniform corrosion degradation. The Monte Carlo simulator models the non-uniformity of the corrosion degradation. Four cases are considered for the retrofitting process: the replacement of the entire plate, reinforcement by two longitudinal stiffeners, two longitudinal and two transversal stiffeners, a flange on the opening. Twelve scenarios are selected and analysed. Four strategies of accessing the space where the side girder is located to perform the retrofit and replacement are considered: no opening, access from the deck of the vessel, access from the side of the vessel, access from the bottom of the vessel. The First Order Reliability Method is used to estimate the reliability of the different solutions towards time. The cost and associated risk assessment are performed to compare the scenarios and determine the most convenient one. A comparison of the most advantageous solutions and the worst one is conducted considering the probability of failure, cost and associated risk.

Keywords: Corrosion; Ultimate Strength, Retrofitting, Cost-Benefit, Risk

1. Introduction

Marine structures operate in a harsh environment and are subjected to degradation during their service life. This deterioration leads to two main aspects of the maritime industry: safety and costs. On one side, Classification Societies Rules [1] indicate necessary parameters to assure the safety of a vessel under the structural point of view such as maximum corrosion wastage allowed, minimum sectional moduli, etc., on the other side the different subject involved in the industry tries to contain the cost associated with safety.

The classical theory of system maintenance describes the failure of components by probabilistic models, often Weibull family, which represents failure rates in operational phases and the ageing phases of the life of components as described in various textbooks, such as in [2-4].

Probabilistic models to describe failure components and demonstrated their application to structural maintenance of ships that are subjected to corrosion and fatigue damage have been
presented in [5-9] used and work presented in [10-12] proposed the planning of structural maintenance of ships based on structural reliability approaches and the concept of Bayesian analysis to the inspection procedure is applied in [13].

Fujita, et al. [14] proposed an adaptive strategy for inspection and repair where the inspection time and the decision criteria for repair can be optimised concerning the total cost and Lotsberg and Kirkemo [15] proposed a method based on probabilistic analysis combined with a resource allocation technique.

Fujimoto and Swilem [16] created a model to find the optimal inspection strategy to minimise the expected costs of inspections employing a Markov Chain Model to describe the entire probabilistic structure of the deterioration process and Madsen [17] applied stochastic models to the study of fatigue crack propagation and inspections.

Faber, et al. [18] presented a simplified inspection and maintenance planning analysis for a tubular joint in a jacket type offshore structure, and Garbatov and Soares [6, 19] applied probabilistic models related to degradation to study risk-based maintenance decisions, and an analysis of the reliability of a bulk carrier hull subjected to the degrading effect of corrosion was presented in [20].

The introduction of risk analysis into the traditional design process cost-effectively established safety objectives. Papanikolaou, et al. [21] proposed risk as a measure of the safety level for the optimisation of the design and Skjong, et al. [22] formalised risk assessment methodology in the design process proposing risk as a design objective among conventional ones, Guia, et al. [23] studied a cost associated with the optimum structural safety level and a risk-based framework for ship and structural design accounting for maintenance planning was proposed in [24].

Nowadays the typical procedure is the substitution of the deteriorated plate with a new one. However, Classification Societies permit a different approach under the mandatory occurrence that structural safety is achieved. A new solution to this problem is the retrofitting of the plate.

Caridis [25] demonstrated the costs associated with the structure renewal or reinforcement and a risk-based framework for the ship and structural design accounting for maintenance planning was proposed in [24, 26, 27].

Chichi and Garbatov [28] studied the regain of the ultimate strength of a non-uniform corroded plate with manhole opening under uniaxial load with the retrofitting process.

In this study is presented a model that relates the retrofitting process or substitution of the plate with risk and cost associated. The philosophy behind the analysis is to furnish to subjects of the maritime industry a tool to quantify the risk for the solution proposed.

Plates are the principal structural components in marine structures. In literature are present studies on the assessment of the ultimate strength of steel plates. Several studies have been performed on the ultimate strength of plates and stiffened plates with an opening.

Shanmugam, et al. [29] studied the variation of ultimate strength in thin perforated plates and the incidence of the different positioning of the opening affecting the ultimate strength. They also analysed the post-buckling behaviour and the ultimate strength of perforated plates under uniaxial or biaxial compression.

Paik, et al. [30] developed formulae for the assessment of plates under the combination of the biaxial compression and edge shear and Kim, et al. [31] proposed a formula for the assessment of the ultimate strength of a perforated plate under axial compression. This study was improved in [32] with experiments, both numerically and in scale, of perforated plates. In this study, it has been proved the influence of different kind of stiffeners on the ultimate strength.

To investigate experimentally and numerically the severe non-uniform corrosion Garbatov, et al. [33] studied the degradation effect on the load carrying capacity of stiffened plates, where different factors leading to a reduction of structural capacity have been investigated, including
the material properties, the degree of degradation, equivalent thickness and testing support conditions.

The influence of large openings on side shell plating demonstrating that the relation between the increase in the number of holes and the diminution of the ultimate strength bending capacity is not linear was analysed in [34-37].

The objective of the study here is to analyse the possibility to recover the strength of the side girder plate of an oil tanker with a retrofit or substitution of the plate. The panel presents a manhole shape opening, and it is subjected to uniaxial compressive load and randomised non-uniform corrosion. For the evaluation of results, a risk assessment is performed, and also a comparison between the most advantageous and the worst solutions is conducted considering the probability of failure and cost.

2. Strength assessment

In the past several corrosion deterioration models, linear and non-linear have been developed. The present study adopts the corrosion deterioration model developed in [38], which was latterly calibrated in [20] and used to develop a formulation in [39] that address the limited corrosion depth measurement data and the current Common Structural Rules [1], CSR corrosion adds in defining the corrosion degradation depth on both sides of a steel plate.

The non-linear corrosion degradation model developed in [20, 39] classify the ship hull plates according to their surrounding environments considering the boundaries between any two spaces that can have similar or different environments and in consequence different corrosion wastage. The thickness reduction due corrosion of any ship plate is equal to the sum of the corrosion wastage on each side [39]:

\[
E[d_{12}(t)] = d_1(t) + d_2(t) = \begin{cases} 
\frac{d_{01}}{2} \left(1 - e^{-(t - \tau_{c1}/\tau)}\right) + d_{02} \left(1 - e^{-(t - \tau_{c2}/\tau)}\right), t > \tau_{c1}, \tau_{c2} \\
\frac{d_{01}}{2} \left(1 - e^{-(t - \tau_{c1}/\tau)}\right) - \tau_{c1} - t < \tau_{c2}, \tau_{c2} \left(1 - e^{-(t - \tau_{c2}/\tau)}\right), t < \tau_{c1}, \tau_{c1} < t < \tau_{c2} \\
0, t < \tau_{c1}, \tau_{c2}
\end{cases}
\]  

(1)

where \(E[d_{12}(t)]\) is the corrosion wastage of both sides of the plate, \(d_1(t)\) is the corrosion wastage of the side 1, \(d_2(t)\) is the corrosion wastage of the side 2, \(t\) is the time, \(d_{01}\) and \(d_{02}\) are the long-term corrosion wastage of the two sides, \(\tau_{c1}\) and \(\tau_{c2}\) are the coating life of the two sides, \(\tau_{c1}\) and \(\tau_{c2}\) are the transition time of the two sides.

The corrosion depth is assumed to be described by the Log-Normal distribution with a mean value of \(E[d_{12}(t)]\) and standard deviation as derived in [20]:

\[
StDev[d_{12}(t)] = \left\{ \begin{array}{ll}
\frac{d_{01}}{a_{12}} \ast Log(t - \bar{\tau}_c - b_{12}) - c_{12}, & t \geq \tau_{c1}, \tau_{c2} \\
0, & t \leq \tau_{c1}, \tau_{c2}
\end{array} \right.
\]

(2)

where \(\bar{\tau}_c\) is the minimum coating life between the two sides of the plate, \(a_{12}, b_{12}\) and \(c_{12}\) are respectively equal to 13.84 years, -9.16 years and 13.22 years.

The procedure developed in [28] is followed to identify the non-uniform corrosion degradation of plates employing the Monte Carlo [40] approach.

The structure analysed here is considered to be a side-girder in the double bottom of an oil tanker. The structure has the following main characteristics: length, \(l=4000\) mm, width, \(b=1400\) mm, initial thickness, \(t_{p,initial}=22\) mm. The structure is represented by a plate with a manhole.
type opening with an extended elliptical opening, $b_{opening} = 600 \text{ mm}$, length of opening, $l_{opening} = 800 \text{ mm}$, and the radius of the opening, $r_{opening} = 300 \text{ mm}$, as shown in Pogreška! Izvor reference nije pronaden..
3. Failure assessment

The ultimate limit state function of the longitudinal girder plate with manhole under uniaxial compression is defined as \[ g(t) = \tilde{x}_u S M_{CR}(t) \tilde{\sigma}_u(t) - \tilde{x}_{SW} \tilde{M}_{SW} - \tilde{x}_w \tilde{x}_s \tilde{M}_w \] \hspace{1cm} (4)

where \( S M_{CR}(t) \) is the midship section modulus, \( \tilde{\sigma}_u(t) \) is the ultimate stress, \( \tilde{M}_{SW} \) is the still water bending moment, \( \tilde{M}_w \) is the wave-induced bending moment, \( \tilde{x}_u \) is the model uncertainty on the ultimate strength, \( \tilde{x}_{SW} \) is the uncertainty in the model of predicting the still water bending moment, \( \tilde{x}_w \) is the uncertainty in the estimation of the wave-induced bending moment due to linear analysis and \( \tilde{x}_s \) takes into account non-linearity and the statistical descriptors of the uncertainty factors are assumed as:

\[ \tilde{x}_u \sim N \{ 1; 0.15 \} \] \hspace{1cm} (5a)

\[ \tilde{x}_{SW} \sim N \{ 1; 0.05 \} \] \hspace{1cm} (5b)

\[ \tilde{x}_w \sim N \{ 0.9; 0.14 \} \] \hspace{1cm} (5c)

\[ \tilde{x}_s \sim N \{ 1.15; 0.03 \} \] \hspace{1cm} (5d)

where \( N \) indicates the Normal distribution function, the first term inside brackets is the mean value, and the second term is the standard deviation. The analysis is performed for a Panamax Tanker with the following main dimensions: the length between perpendicular, \( L=208 \text{ m} \), beam, \( B=32.25 \text{ m} \), depth, \( D=16.125 \text{ m} \), draught, \( d=9.5 \text{ m} \), block coefficient, \(Cb=0.8\), deadweight, \( DW=75,000 \text{ tons} \) and lightweight, \( LW=9,304 \text{ tons} \).

The time-dependent variation of the Section Modulus \( SM(t) \) has been derived taking into account the general corrosion of the structural components of the midship section, plates and stiffeners, accounting for the different environment conditions associated to the location of the plates and the corrosion addiction, \( t_c \), as stipulated by CSR. In the present study, the
environmental coefficients as derived in [39] are employed.

Table 2 – Long-term corrosions, transition times and coating lives considered in the study

<table>
<thead>
<tr>
<th></th>
<th>$d_{\infty 1}$</th>
<th>$d_{\infty 2}$</th>
<th>$\tau_{t1}$</th>
<th>$\tau_{t2}$</th>
<th>$\tau_{c1}$</th>
<th>$\tau_{c2}$</th>
<th>case</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plates</td>
<td>0.78</td>
<td>0.65</td>
<td>13.35</td>
<td>11.97</td>
<td>3.17</td>
<td>3.17</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>0.78</td>
<td>0.97</td>
<td>13.35</td>
<td>15.05</td>
<td>3.17</td>
<td>3.17</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>0.78</td>
<td>0.56</td>
<td>13.35</td>
<td>10.92</td>
<td>3.17</td>
<td>11.49</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>0.93</td>
<td>0.56</td>
<td>14.7</td>
<td>10.92</td>
<td>10.54</td>
<td>11.49</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>0.96</td>
<td>0.93</td>
<td>14.94</td>
<td>14.7</td>
<td>11.49</td>
<td>10.54</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>0.96</td>
<td>0.96</td>
<td>14.94</td>
<td>14.94</td>
<td>11.49</td>
<td>11.49</td>
<td>6</td>
</tr>
<tr>
<td></td>
<td>0.78</td>
<td>0.78</td>
<td>13.35</td>
<td>13.35</td>
<td>3.17</td>
<td>3.17</td>
<td>7</td>
</tr>
<tr>
<td></td>
<td>0.93</td>
<td>0.96</td>
<td>14.7</td>
<td>14.94</td>
<td>10.54</td>
<td>11.49</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>0.78</td>
<td>0.56</td>
<td>13.35</td>
<td>10.92</td>
<td>3.17</td>
<td>11.49</td>
<td>9</td>
</tr>
<tr>
<td></td>
<td>1.18</td>
<td>0.78</td>
<td>16.55</td>
<td>13.35</td>
<td>3.17</td>
<td>11.49</td>
<td>10</td>
</tr>
<tr>
<td>Stiffeners</td>
<td>0.78</td>
<td>0.78</td>
<td>13.35</td>
<td>13.35</td>
<td>3.17</td>
<td>3.17</td>
<td>11</td>
</tr>
<tr>
<td></td>
<td>0.93</td>
<td>0.93</td>
<td>14.7</td>
<td>14.7</td>
<td>10.54</td>
<td>10.54</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>0.96</td>
<td>0.96</td>
<td>14.94</td>
<td>14.94</td>
<td>11.49</td>
<td>11.49</td>
<td>13</td>
</tr>
</tbody>
</table>

The different long-term corrosion wastages, $d_{\infty 1}$ and $d_{\infty 2}$, the coating lives, $\tau_{c1}$ and $\tau_{c2}$, and the transition time, $\tau_{t1}$ and $\tau_{t2}$, for the two sides of the plate are present in Table 2: case 1 is considered for the bottom plating, case 2 for the bilge plating, case 3 for the side shell plating, case 4 for the shell plating within 3 meters below top of tank, case 5 for the weather deck plating/ballast tank, case 6 for the weather deck plating/cargo hold, case 7 for the longitudinal girder, case 8 for the longitudinal bulkhead/cargo-ballast plating (within 3 meters below top of tank), case 9 for the longitudinal bulkhead/cargo-ballast plating (elsewhere), case 10 for the inner bottom plating/bottom of tank. Exclusively for the stiffeners, the following cases are used: case 10 for the ballast tank stiffener (elsewhere), case 11 for the ballast tank stiffener (within 3
meters below the top of tank) and case 12 for the cargo oil tank stiffener.

Fig. 2 presents the section moduli estimated at the level of the deck and bottom and the midship section area as a function of time due to the adopted corrosion degradation of structural components. It can be seen that the midship section modulus at the level of the bottom is more severely reduced by the corrosion degradation, which may be explained by the lower coating protection life and, in consequence, the corrosion wastage initiates earlier.

A general time-dependent relationship for the midship section modulus is derived following the asymmetrical sigmoidal function developed here as:

\[
E[SM(t)] = \begin{cases} 
SM_0 & , t < \tau_{c,\text{min}} \\
SM_0 + \frac{d_{SM}SM_0}{1+\left(\frac{t-\tau_{c,\text{min}}}{\tau_{t,\text{max}}}ight)^b_{SM}m_{SM}} & , t \geq \tau_{c,\text{min}}
\end{cases}
\]

where \(SM_0\) is the intact section modulus, \(\tau_{c,\text{min}}\) is the lowest value of coating life among structural components of the midship section, \(\tau_{t,\text{max}}\) is the highest time of the transition among the structural elements of the midship section, \(d_{SM}\), \(b_{SM}\) and \(m_{SM}\) are coefficients equal to -0.90, 3.68 and -0.7.

The midship section modulus, \(SM(t)\), is considered to be described by the Log-Normal distribution with a mean value of \(E[SM(t)]\) and standard deviation:

\[
\text{StDev}[SM(t)] = SM_0 \cdot \left[a_{SM} \cdot \ln(t) - c_{SM}\right]
\]

where the coefficients \(a_{SM}\) and \(c_{SM}\) are defined respectively as 0.1246 and 0.2273.

\[
SM(t) \sim \text{LogNormal}\{E[SM(t)]; \text{StDev}[SM(t)]\}
\]

The still water bending moment is fitted to the Normal distribution function [43]. It is assumed that the still water bending moment given by the CSR is the maximum value with a probability of exceedance of 5%. The significant variability in the still water bending moment results in a coefficient of variation of 40%, which gives the mean value of the distribution to be 60% of \(M_{SW,CS}\):

\[
M_{SW,CS} \sim N\{0.6M_{SW,CS} ; 0.24M_{SW,CS}\}
\]

If the wave-induced loads can be represented as a stationary Gaussian process (short-term analysis) then the wave-induced bending moment given by the CSR may be modelled as an extreme value following the Gumbel distribution function [44]:

\[
F_w(\omega) = \exp\left\{-N_w\exp\left(-\frac{\omega^2}{2\lambda_0}\right)\right\}
\]

\[
\mu_w = M_{W,CS} = \sqrt{2\lambda_0 \ln(N_w)} + \frac{0.5772}{\sqrt{2\lambda_0 \ln(N_w)}}
\]

\[
\sigma_w = \frac{\pi}{\sqrt{6}} \sqrt{\frac{\lambda_0}{2 \ln(N_w)}}
\]

where \(\mu_w\) is the mean value and \(\sigma_w\) is the standard deviation, \(N_w\) is the number of wave induced bending moment peaks and \(\lambda_0\) is the mean square of the wave induced bending moment.
wave induced bending moment, given by the CSR, is assumed to be the mean value and when \( N_w \) is about 1000 and it is equivalent to a 3 hours storm and gives a coefficient of variation of 9%.

The analysed plate to be retrofitted is a side girder located in the inner bottom of a tanker ship, and according to CSR, the required bending moments and the sectional modulus are:

\[
M_{SW,CSR} = 1521.348 [MNm] \tag{13a}
\]

\[
M_{W,CSR} = 2092.714 [MNm] \tag{13b}
\]

\[
SM_{CSR} = 25.005 [m^3] \tag{13c}
\]

The ultimate strength of the corroded plate \( \sigma_u \) is modelled by the Log-Normal distribution function:

\[
\sigma_u(t) \sim \text{LogNormal}\{E[\sigma_u(t)]; 0.05\} \tag{14}
\]

The mean value of the ultimate strength concerning the time, \( E[\sigma_u(t)] \), is described by an asymmetrical sigmoid function developed here, when the corrosion starts to act on the plate.

\[
E[\sigma_u(t)] = \left\{ \begin{array}{ll}
 c_{AXSG} \cdot \sigma_{YP} & t < \tau_c \\
 a_{AXSG} \cdot \sigma_{YP} + d_{AXSG} \cdot \sigma_{YP} \left[ 1 + \left( \frac{t-\tau_c}{\tau_t} \right) \right] b_{\sigma} & t \geq \tau_c \geq 25 \text{ years} 
\end{array} \right.
\tag{15}
\]

where \( \sigma_{YP} = 355 \text{ MPa} \) is the yield stress of the material, the coating life \( \tau_c = 3.17 \text{ years} \), the transition time \( \tau_t = 13.75 \text{ years} \), the coefficients \( a_{AXSG}, b_{\sigma}, c_{AXSG}, d_{AXSG} \) and \( m_{\sigma} \) are equal respectively to 0.5225, 1.86, 0.06, 0.46, 6.69. The standard deviation has been assumed as 0.05.

![Fig. 3 - Probability of failure of scenario “b”, “c” and “i”](image-url)

The probability of failure along the 25-year-service life of the vessel, using the limit state
function as is defined by Eqn 4 and employing the FORM [45] with the commercial software COMREL [46]. For the different scenarios, the probability of failure as a function of time, conditional to the retrofitting at 10\textsuperscript{th} and 24\textsuperscript{th} year, and replacement at the 14\textsuperscript{th} year is analysed. Fig. 3 presents the probability of failure of the plate retrofitted with two longitudinal stiffeners 50 x 10, flange 100 x 10 and two longitudinal and two transversal stiffeners 100 x 100 x 10 x 10.

It is noticeable that the drop in the probability of failure at year 24\textsuperscript{th} has a greater magnitude and impact on recovering the reliability of the structure than the ones occurring at 10\textsuperscript{th} and 14\textsuperscript{th} year. This can be explained with the fact that the midship section is less corroded and the “impact” of substitution or retrofit of a single plate at year 10\textsuperscript{th} and 14\textsuperscript{th} is not effective in the global scale while it is at the 24\textsuperscript{th} year.

4. Cost-benefit analysis

A cost-benefit analysis has been performed to provide the best solution associated with the containment of costs and safety. The study focuses on defining the optimum safety level combined with the cost of the retrofit of a corroded side girder plate and the reduction of risk. The risk, \( R \) is defined as a product of the probability of failure, \( P_f \), times the associated consequences, \( C \):

\[
R = P_f \ast C
\]  
(16)

The cost related to the structural failure of the ship, \( C_t^{\beta t} \) including the cost of the retrofit process is defined as:

\[
C_t^{\beta t} = C_{T_f}^{\beta t} + C_{me}^{\beta t} + C_{\text{retrofitting}}^{\beta t}
\]  
(17)

where \( C_{T_f}^{\beta t} \) is the cost associated with the structural failure of the ship, \( C_{me}^{\beta t} \) is the cost associated to the structural redesign of the new ship and \( C_{\text{retrofitting}}^{\beta t} \) is the cost associated with the structural retrofitting process.

The cost of the structural failure of the ship includes four major groups:

\[
C_{T_f}^{\beta t} = \sum_{t=1}^{T} P_f(t) [C_n(t) + (C_c + C_d + C_v)] e^{-\gamma t}
\]  
(18)

where \( P_f(t) \) is the probability of failure, \( C_n(t) \) is the cost of the ship as a function of time, \( C_c \) is the cost for the loss of cargo, \( C_d \) is the cost for the accidental oil spilling and cleaning, \( C_v \) is the cost for the loss of human life and \( \gamma \) the discount rate, in this case taken as 5\%.

The cost of the ship is defined as a function of the time and the scrapping value, which has been linearly discounted during the service life of 25 years:

\[
C_n(t) = C_{n_0} - (C_{n_0} - C_{\text{scrap}}) \left( \frac{t}{25} \right)
\]  
(19)

where \( C_{n_0} \) is the initial cost of the ship and \( C_{\text{scrap}} \) is the revenues for the scrapping of the ship. The initial value of the ship is an estimation of the current prices present in a market review in [47] as 38.0 M\€. The revenues from scrapping the ship are defined as:

\[
C_{\text{scrap}} = C_{\text{steel/ton}} LWT
\]  
(20)
where the cost of steel is assumed as \( C_{\text{steel/ton}} = 700 \text{ €/ton} \) and \( LWT \) is the lightweight of the vessel.

The cost due to the loss of cargo take into account as only \( P_{\text{spill}} = 20\% \) of the total cargo carried to be spill caused by the structural failure of the ship, [48]:

\[
C_c = C_{\text{crude/ton}} \times DWT \times P_{\text{spill}}
\]  
(21)

where the cost of a ton of crude oil is assumed as \( C_{\text{crude/ton}} = 62 \text{ €/ton} \), [49], \( P_{\text{spill}} \) is the percentage of oil spilt caused by structural failure and \( DWT \) is the deadweight of the vessel.

A fraction of the total spilt oil due to structural failure it is considered to be 10% of chance that the oil reaches shoreline [48], meaning that there are additional costs associated to it such as cleaning. In this case, the additional costs are estimated employing the CATS criterion:

\[
C_d = P_{\text{spill}} \times P_{\text{sl}} \times \text{CATS} \times DWT
\]  
(22)

where CATS is assumed to be 50,000 €, which is the Cost of Averting a Ton of oil Spilt, \( P_{\text{sl}} = 10\% \) as the probability of the oil spilt reaching the shoreline.

The probability of loss of crew for this study is assumed to be 25%, [50]:

\[
C_v = n_{\text{crew}} \times P_{\text{crew}} \times \text{ICAF}
\]  
(23)

where \( n_{\text{crew}} = 25 \) is the number of the crew members, \( P_{\text{crew}} = 25\% \) is the probability to avert a fatality, ICAF = 3.30 millions of euros is the cost of the occurrence of the fatality.

Due to corrosion degradation, the structural components lose their stiffness and the cost of the steel ship structure as a function of time is defined as:

\[
C_{\text{me}} = (1 - ISM(t)) \times W_{\text{steel}} \times C_{\text{steel}}
\]  
(24)

\[
ISM(t) = \frac{A(t)}{A(0)}
\]  
(25)

where \( ISM(t) \) is the steel reduction due to corrosion measured as a function of time, \( A(t) \) is the midship section area as a function of time, \( t \), \( A(0) \) is the intact midship section area, \( W_{\text{steel}} \) is the weight of steel and \( C_{\text{steel}} \) is the cost of steel per ton.

The current study focuses on the retrofitting process to regain the ultimate strength capacity of the girder plate in the double bottom with a man-hole opening. This process takes into account two different aspects: the retrofitting performed and the strategy to apply it.

The retrofit occurs when the steel plate’s ultimate strength capacity drops below 75% of intact ultimate strength capacity and in the present study occurs at the 10\text{th} and 24\text{th} year, while at the 14\text{th} year, the plate as to be replaced due to reaching the minimum acceptable thickness (3 mm), as shown in Fig. 4.
Fig. 4 - Ultimate strength of retrofitted plate (2 x 50x10 stiffeners, 2x 100x100x10x10, plate substitution) vs corroded plate.

Table 3 shows different solutions adopted in the current study, where cases b, c and d are representing the reinforcement of the corroded plate with 2 longitudinal stiffeners, e, f, and g, the reinforcement is performed by a flange on the opening and i, j, k by 2 longitudinal and 2 transversal stiffeners and the case l a box does the reinforcement.

Table 3 - Retrofitting solutions adopted.

<table>
<thead>
<tr>
<th>Type</th>
<th>N. Scenario</th>
</tr>
</thead>
<tbody>
<tr>
<td>Only plate</td>
<td>Plate</td>
</tr>
<tr>
<td></td>
<td>50x10</td>
</tr>
<tr>
<td>2 Longitudinal stiffeners</td>
<td>100x100x10x10</td>
</tr>
<tr>
<td></td>
<td>300x80x10</td>
</tr>
<tr>
<td>Flange on the opening</td>
<td>flange 100x10</td>
</tr>
<tr>
<td></td>
<td>flange 200x10</td>
</tr>
<tr>
<td></td>
<td>flange 300x17</td>
</tr>
<tr>
<td>2 Longitudinal stiffeners + 2 Transversal stiffeners</td>
<td>50x10+50x10</td>
</tr>
<tr>
<td></td>
<td>100x100x10x10+100x10</td>
</tr>
<tr>
<td></td>
<td>300x80x10x10+300x8</td>
</tr>
<tr>
<td></td>
<td>300x80x10x10+300x20</td>
</tr>
<tr>
<td></td>
<td>box</td>
</tr>
</tbody>
</table>

The cost of the retrofitting is defined as:

\[ C_{\text{retrofitting}}^{\beta_t} = (C_{\text{Capex}} + C_{\text{associated}}) + C_{\text{Strategy}} \]  

(26)

where \( C_{\text{Capex}} \) is the cost of material, manufacturing and installation of the reinforcement on the plate as defined in [28], \( C_{\text{associated}} \) is the cost associated to the access to the location of the retrofitted plate, including cleaning, lighting, opening and closing the tanks and tank testing,
and $C_{\text{Strategy}}$ is the cost associated with the strategy adopted:

$$C_{\text{associated}} = n_s A_S + C_S + C_{ST}$$  \hspace{1cm} (27)

where $n_s$ is the number of accesses to open and reach the working space, $A_S$ is the cost to access the space, $C_S$ is the cost of cleaning, lighting, opening and closing the tanks and $C_{ST}$ the cost to test the water tightness.

In this study, the strategy cost is taken into account only when the entire plate has to be replaced. Four different strategies are taken into account, and they are shown in Fig. 5.

**Fig. 5 – Plate substitution strategies**

Case 1 takes into account that the access to the inner bottom is done from the deck with an opening of 4 x 1 meters, Case 2 from the side with an opening of 2 x 2 meters, Case 3 from the bottom with an opening 4 x 1 meters and Case 4 the access is done without creating openings in the hull. Case 1 and Case 2 also comprehend the necessity to create an opening on the inner bottom of 4 x 1 meters. The strategy cost, $C_{\text{Strategy}}$ is defined as:

$$C_{\text{Strategy}} = n_{1} A_{BT} + n_{2} A_{OT} + C_{BC} + C_{OC} + n_{3} C_{\text{Drydock}} + C_{BT,t} + C_{OT,t} + C_{d,u} + C_{\text{plates}}$$  \hspace{1cm} (28)

where $A_{BT}$ is the cost to access the ballast tank, $A_{OT}$ is the cost to access the oil tank, $C_{BC}$ is the cost of cleaning the ballast tank, $C_{OC}$ is the cost of cleaning the oil tank, $C_{\text{Drydock}}$ is the cost associated to drydock, $C_{BT,t}$ is the cost of testing the ballast tank, $C_{OT,t}$ is the cost of testing the oil tank, $C_{d,u}$ is the cost associated to docking and undocking of the ship, $C_{\text{plates}}$ is the cost of the plates to replace, $n_1$ and $n_2$ are the numbers of accesses to open in the ballast and oil tanks, $n_3$ is the number of days in the drydock. In this case study $n_1 = 6$ days and $n_2 = 6$ days, $n_3 = 4$ days. Those values are susceptible to difference due to different structural arrangement, for the openings, and to dry dock time and in an association to deeper works on the vessel such as multiple repairs or surveys. Table 4 shows the cost for each process related to the chosen strategy.
Retrofitting analysis of tanker ship hull structure subject to corrosion

Chichi D., Garbatov Y.

Table 4 - Cost associated with considered strategies.

<table>
<thead>
<tr>
<th>Process</th>
<th>Deck opening</th>
<th>Side opening</th>
<th>Bottom opening</th>
<th>No opening</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tank access</td>
<td>425</td>
<td>425</td>
<td>425</td>
<td>425 [€]</td>
</tr>
<tr>
<td>Ballast tank access</td>
<td>425</td>
<td>845</td>
<td>425</td>
<td>425 [€]</td>
</tr>
<tr>
<td>Ballast Tank cleaning</td>
<td>3300</td>
<td>4960</td>
<td>3300</td>
<td>3300 [€]</td>
</tr>
<tr>
<td>Oil tank cleaning</td>
<td>23400</td>
<td>23400</td>
<td>93600</td>
<td>93600 [€]</td>
</tr>
<tr>
<td>Dry dock</td>
<td>93600</td>
<td>93600</td>
<td>93600</td>
<td>93600 [€]</td>
</tr>
<tr>
<td>Tank testing</td>
<td>75</td>
<td>115</td>
<td>75</td>
<td>75 [€]</td>
</tr>
<tr>
<td>Oil tank testing</td>
<td>16775</td>
<td>16775</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Docking</td>
<td>4095</td>
<td>4095</td>
<td>4095</td>
<td>4095 [€]</td>
</tr>
<tr>
<td>Undocking</td>
<td>4095</td>
<td>4095</td>
<td>4095</td>
<td>4095 [€]</td>
</tr>
<tr>
<td>Cost of replace study plate</td>
<td>10385</td>
<td>10385</td>
<td>10385</td>
<td>10385 [€]</td>
</tr>
<tr>
<td>Cost of replacing inner bottom plate</td>
<td>6870</td>
<td>6870</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cost of replacing deck plate</td>
<td>6870</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cost of replacing side plate</td>
<td>6400</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cost of replacing bottom plate</td>
<td></td>
<td></td>
<td></td>
<td>10382 [€]</td>
</tr>
</tbody>
</table>

The retrofitting (twice per service life) and plate replacement (once per service life, except scenario one where there are two replacements) cost per solutions adopted in comparison to the different strategies are presented in Fig. 6 and Fig. 7.

![Graph showing total cost for different strategies: bottom opening and no opening](image)

**Fig. 6** - Total cost for different strategies: bottom opening and no opening

It is noticeable that the strategy with no openings in the hull is more economical than the bottom opening.
In this case, the most economical solution is the strategy that provides access from the deck. Overall the most economical solutions are associated with the retrofitting processes conducted without or with a limited number of openings on the hull of the vessel. Table 5 reassumes the increase of cost for every scenario as a function of the most economical one: flange 100 x 10.

**Table 5** – Comparison of costs in comparison of most economical (no opening “e”)

<table>
<thead>
<tr>
<th></th>
<th>a</th>
<th>b</th>
<th>c</th>
<th>d</th>
<th>e</th>
<th>f</th>
<th>g</th>
<th>h</th>
<th>i</th>
<th>j</th>
<th>k</th>
<th>l</th>
</tr>
</thead>
<tbody>
<tr>
<td>Deck op.</td>
<td>175%</td>
<td>44%</td>
<td>44%</td>
<td>44%</td>
<td>44%</td>
<td>44%</td>
<td>44%</td>
<td>44%</td>
<td>44%</td>
<td>44%</td>
<td>44%</td>
<td>44%</td>
</tr>
<tr>
<td>Side op.</td>
<td>178%</td>
<td>45%</td>
<td>46%</td>
<td>46%</td>
<td>45%</td>
<td>45%</td>
<td>45%</td>
<td>45%</td>
<td>46%</td>
<td>46%</td>
<td>46%</td>
<td>46%</td>
</tr>
<tr>
<td>Bottom op.</td>
<td>104%</td>
<td>8%</td>
<td>9%</td>
<td>9%</td>
<td>8%</td>
<td>8%</td>
<td>9%</td>
<td>8%</td>
<td>9%</td>
<td>9%</td>
<td>9%</td>
<td>9%</td>
</tr>
<tr>
<td>No op.</td>
<td>88%</td>
<td>0.1%</td>
<td>0.26%</td>
<td>0.46%</td>
<td>MIN</td>
<td>0.03%</td>
<td>0.13%</td>
<td>0.10%</td>
<td>0.29%</td>
<td>0.50%</td>
<td>1%</td>
<td>0.2%</td>
</tr>
</tbody>
</table>

It can be observed that the strategies no opening and bottom opening have containment of the costs. In particular, the scenario “a” (substitution of the entire plate) is the worst possible solution concerning the real economic aspect of the retrofitting process.

The economic comparison (total cost) associated with the service life of the vessel is presented in Fig. 8.
The total cost includes the mandatory substitution of the plate due to minimum thickness allowed by CSR for the scenarios from “b” to “l” with a total of three operations: the retrofit of the plate at the 10th year, the substitution of the plate at 14th year and retrofit of the new plate at the 24th year. It is noticeable that the scenario “a” is the most economical one. The explanation resides primarily in the difference of some operations: two against the three for the others. Such as for the scenario “g” (flange 300 x 17) and “i” (2 x 100x100x10x10 + 2 x 100x10) a better cleaning and coating protection could reduce the costs and prevent a second retrofitting process.

Fig. 9 presents the total cost associated with the time frame 10th to 14th year, the first retrofitting process to the replacement of the plate.

The total cost has a different distribution with all scenario residing on the same level. In particular, the scenario “i” is the most economical solution regardless of the material used (2 longitudinal and two transversal stiffeners) in comparison to others such as the flange or the
two longitudinal stiffeners. The difference in the cost between the strategies for the scenario “i” is negligible: the most economical one is the one with no opening strategy, the bottom strategy presents a slight increase of costs in comparison of no opening strategy while side opening and deck opening have a moderate increase.

Fig. 10 presents the total costs associated with the time frame 14th to 25th year (substitution of the plate to end of service life).

Also, in this case, the most economical solution is the substitution of the plate (“a”). Among the other scenarios, the “g” and “i” (respectively flange 300x17 and 2x100x100x10x10 + 2x100x10) are the most convenient ones.

The last time frame of interest, from the 24th to the 25th year (second retrofit process to the end of service life), is shown in Fig. 11.
The cost associated with the scenario “a” for this time frame is not comprehensive of the retrofit process. The replacement of the plate happens in the 20th year. The solution more economical remains the “i” (2x100x100x10x10 + 2x100x10).

In the present study, the risk is estimated for every strategy during the service life of the vessel. Fig. 12 presents the total risk for the strategy “no opening” during the 25 years of service (a) and the risk as a function of time (b).

![Graph](image)

**Fig. 12** – Total risk “no opening” strategy (a) and risk as a function of time (b)

The lowest risk during the service life is achieved by the scenario “a”; among the retrofit processes scenarios “g” and “i” have the lowest total risk. From Fig. 12 (b), it can be observed how the risk grows during the time due to the progressing of the corrosion of the entire structure and as well of the considered plate. In particular, while the growth of the risk has a similar pattern from year 15th to 24th, the second retrofit process in year 24 shows different effectiveness for the scenarios considered. All four strategies show the same pattern of risk. The explanation resides in the fact that the total costs for different strategies for every scenario are negligible.
The most substantial contribution to the total cost is given by the fixed costs that are the same for every case. A more detailed cost analysis would make sense of the difference between each strategy for each scenario.

An important aspect is given by the rate of increment of the risk between the year 24 and 25 as shown in Table 6.

<p>| | | | | | | | | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>b</td>
<td>c</td>
<td>d</td>
<td>e</td>
<td>f</td>
<td>g</td>
<td>h</td>
<td>i</td>
<td>j</td>
<td>k</td>
</tr>
<tr>
<td>52.06°</td>
<td>50.88</td>
<td>54.57°</td>
<td>50.21°</td>
<td>47.37°</td>
<td>43.07°</td>
<td>51.98°</td>
<td>44.43°</td>
<td>50.03°</td>
<td>50.09°</td>
</tr>
</tbody>
</table>

As well for the cost comparison of the different retrofit scenarios, the solution “i” and “g” display a rate of an increase in the risk lower than the other cases.

The following scenarios are selected for additional analysis: “a”, ”g”, ”i” and “l”. The selection is made by the minimum and maximum values of the cost and risk analysis. It is important to compare the scenarios “g” and “i” with the case with the substitution of the plate (“a”) due to their close values with this last one. It is essential to verify the most expensive case in comparison with the most economical one.

Fig. 13 a, b, c, shows the comparison between the probability of failure and costs for the selected scenarios for the strategy related to the opening on the deck. Only one strategy is shown because the values among the different strategies have a negligible difference.
It is noticeable that all the scenarios with retrofitting, “g”, “i” and “l” have the same maximum extension of the probability of failure. This is because after the 14\textsuperscript{th} year the original retrofitted plate is replaced with a new only at the 24\textsuperscript{th} year there is a new retrofitting process. The second retrofit can be appreciated with the different “amplitude” of the curves.

The comparison between the probability of failure and total costs between the case “a” and the retrofit scenario selected towards time is shown in Fig. 14 a, b, c.
Fig. 14 - Probability of failure and total costs between the scenario “a” and ”g”, ”i” and “l.”
It is noticeable that while the total cost of the retrofitted scenarios is mostly below the solution “a”, which is explained by the lower amount of the cost associated with the retrofit process. On the other hand, the probability of failure rose suddenly and peaked in the 24th year. It is noticeable that with the retrofit processes “g” and “i”, it drops below the probability of failure of the scenario “a”. This gives an idea of the viability of the retrofit process and profitability at that stage.

5. Conclusion

The works developed a mathematical tool to identify the possibility to recover the structural capacity of a side girder plate of an oil tanker accounting for the probability of failure and cost associated with the retrofit or replacement of the plate.

Four different maintenance actions were considered for the retrofitting process: the replacement of the entire plate, reinforcement by two longitudinal stiffeners, two longitudinal and two transversal stiffeners, a flange on the opening. Twelve scenarios were analysed including four different strategies of accessing the space where the side girder is located to perform the retrofit and replacement are considered: no opening, access from the deck of the vessel, access from the side of the vessel, access from the bottom of the vessel.

The results demonstrated that the more economical and with lower risk solution is the replacement of the entire plate. A better extension of the service life of the retrofitted plate would have been achieved with better coating protection leading to a postpone of the corrosion degradation and reduction of the associated cost during the service life.

The developed mathematical tool is flexible and can be used to identify the most suitable maintenance scenario in recovering the structural capacity of corroded structural components and reducing the associated risk.

6. Acknowledgement

This paper reports a work developed in the project” Ship Lifecycle Software Solutions “, (SHIPLYS), which was partially financed by the European Union through the Contract No 690770 - SHIPLYS - H2020MG-2014-2015.

7. References


Retrofitting analysis of tanker ship hull structure subject to corrosion  
Chichi D., Garbatov Y.


Davide Chichi, davide.chichi@centec.tecnico.ulisboa.pt
Yordan Garbatov, yordan.garbatov@tecnico.ulisboa.pt

Accepted: 23.04.2019.
Centre for Marine Technology and Ocean Engineering (CENTEC), Instituto Superior Técnico, Universidade de Lisboa, Lisbon, Portugal
DESIGN AND MODEL TEST OF STRUCTURAL MONITORING AND ASSESSMENT SYSTEM FOR TRIMARAN

UDC 629.5.022.3:629.5.015.4:629.5.018.4
Original scientific paper

Summary

Due to the unique structural design of multihull, the structural response on trimaran is often more complex than the response of monohull in navigation. In order to ensure the safety of trimaran navigation and predict the potential damage of local structure, the structure monitoring technology is applied. According to the structural characteristics of trimaran, structural monitoring and assessment system for multihull is designed and introduced in detail, and corresponding model test is taken to demonstrate its effectiveness. The self-propelled trimaran model is installed with sensors in different longitudinal positions to monitor the variety of structural responses in irregular waves. And three real-time structural strength assessment methods in the system are used respectively to indicate the structural state about hull longitudinal strength, local yielding strength and fatigue strength. The influence of different wave azimuths and monitoring positions on structural strength assessment is analysed. Finally, these measured data and analyses results will provide technical support for design and installation of the monitoring system on actual trimaran.

Key words: Trimaran; Structure monitoring; structure strength; model experiment; irregular wave; fatigue damage

1. Introduction

With the development of structure monitoring technology, the hull safety monitoring has been paid more and more attention in shipbuilding industry. The real-time strain on hull structure is observed and analysed by the arrangement of sensors in some critical locations. And the corresponding assessment based on measured data is taken to guide the crews to avoid the structural failure accidents in complex sea environment. The application of hull safety monitoring system into ship has been regard as an effective mean to improve the survivability of ship navigation in harsh sea environment.

In fact, many scholars and research institutions had carried out a series of experiments and correlative researches on hull monitoring technology. Lindemann firstly applied the structure monitoring technology into ships in the late 70's in twentieth Century [1]. Although his research had not achieved the desired results with the limit of computer and sensor
technology, it gave a concise way to build hull monitoring system. In twenty-first Century, the computer technology and related electronic data transmission research had made great progress. And the technology of structure monitoring on ships was more feasible. In 1994, the Bureau veritas and Kelvin Hughes Company jointly developed a kind of hull monitoring system under the European project of Brite-EuRam II [2]. The system could monitor the ship structural stress in its whole life, but the algorithm towards structural response in the system is incomplete. It lacks the extraction and processing of high-frequency hull vibration. Then, Hjelme (1997) tried to use two kinds of fibre-optical sensors in model test to catch the structural response of hydroelasticity and bow slamming [3]. In his research, the advantages of fibre-optical Bragg grating strain sensor in the field of hull structure monitoring were illustrated, and the characteristics of the high-frequency structural response were observed. After the Hjelme’s research about application of fibre-optical sensors in structural monitoring, the American Naval Research Laboratory (NRL) combined with Norwegian Defense Research Establishment (NDRE) to study systematically the structural stress monitoring system with fibre optic sensors from 1997 to 2001. The developed monitoring system was applied respectively in ship experiment of Skjold-class fast patrol boat and Oksøy/Alta-class mine counter measure vessels [4-6]. In the ship tests, the bending moment and wave slamming was monitored by the fibre optic sensors installed at the bow and midship section.

In 2003, an American high-speed catamaran called “HSV-2 Swift” was also installed a new-style structural stress monitoring system by the united research group, which included the United States Naval Operations Center, Georgia Institute of Technology, Michigan University and Lehigh University. This real-time monitoring system focused on the wave force and vibration on the aluminium alloy hull structure, and acceleration was firstly included in the real-time monitoring system as an important monitoring parameter [7-8]. Although the ship test had achieved good monitoring results in this stage, there was still a lack of structural evaluation methods to reflect the structural safety status of ships. Base on the theoretical and experimental research on the hull structural monitoring system using fibre optic sensors, Nielsen studied the structural fatigue strength computing method in a container monitoring system. At the same time, the ship motion and environment data such as vertical acceleration, sailing speed, wave height, wind speed and direction, was also recorded in the system [9]. These data laid the foundation on the improvement of container ship design. And it also demonstrates the value of the monitoring system in civilian ships. In order to ensure safe navigation of civilian ships further, Renato developed a simplified assessment method to monitor the longitudinal strength of hull girder using the stress monitoring system [10]. This processing method of monitoring data could be used as a reference for ship emergency. Then, Kokarakis studied a kind of structural corrosion predict method in the monitoring system for bulk carriers. This system could correct and predict the structural corrosion by monitoring the strain [11]. Data communication technology is also very important in the structural monitoring system. It directly affects the efficiency and stability of the monitoring system. Based on CAN communication technology, Yang designed a kind of data transmission framework to make the structural stress monitoring system more convenient and efficient [12].

In addition, He developed a method by combining the multi-channel cross sampling technology and optical fiber sensing technology to monitor the ship deck deformation in real-time. Its feasibility was proved by the experiment and the simulation calculation [13]. Based on research in data communication technology and structural monitoring technology, Jin developed a kind of Container Ship Structural Monitoring and Evaluation System (CSSMAS) and putted it into actual ship experiment [14]. The container ship called “YuFeng” installed some strain sensors and acceleration sensors to monitor local structure condition. Meanwhile, the encountering waves around this container ship were also monitored by the image sensors.
at the fore and aft of the container ship. The results of ship test were satisfactory, and the data was also used to optimize container ship design.

Although the hull structural monitoring technology has been applied widely, there are still many technical difficulties in the field of hull structure monitoring. In fact, most structural monitoring systems are designed by simple and direct requirement scheme. It has a high requirement for computer space, and it is not suitable for long-term and high-speed hull structure monitoring. In order to ensure the monitoring system stability, the number of monitoring sensors in the system is often limited to less than 10, so that the overall longitudinal structural strength monitoring and local structural strength monitoring towards ships cannot be fully considered at the same time. Due to the limitations of the monitoring sensor number, most systems mainly monitor the hull structure at bow and midship, recording and analysing the bow slamming and hull longitudinal structural strength. For some local structural monitoring systems, the monitoring positions are always selected in a small range, and the longitudinal structural strength assessment is often abandoned. So these systems are not easy to expand its functions. Obviously, a multiple composite structure is needed in the design of monitoring system. Parallel computing technology and efficient data processing method should be applied to improve monitoring system stability. In the research on the real-time structural strength evaluation, the existing systems often adopt a simple deterministic evaluation method. Although this deterministic calculation method is fast, the possibility of the sudden wave load changing is neglected. Therefore, a series of structural reliability assessment methods based on real-time monitoring data is needed to ensure the structural assessment by different structural destruction forms. In addition, the most structural monitoring researches mainly focused on traditional monohull and catamaran. There is a lack of the structural monitoring research towards trimaran. As a new-style multihull, trimaran not only has excellent seakeeping performance, but also suffers complex wave load response in irregular waves. The high sailing speed and application of new materials on trimaran increase the structural failure risk in further. Therefore, there is an urgent need to develop a structural monitoring and assessment system for trimaran.

In this paper, Structural Monitoring and Assessment System (SMAS) for trimaran is designed and developed. The multi-level and modular design method is used to make the system run flexibly and expand easily. In order to speed up the system operation further, multi-thread parallel technology is also applied. And the real-time structural strength assessment method, such as simplified yield strength assessment, longitudinal strength assessment, fatigue strength evaluation, is studied to meet the need for overall longitudinal structural strength monitoring and local structural strength monitoring. By constructing the corresponding probability distribution of wave load and structure resistant ability, the structural strength reliability method is studied. In order to improve the utilization of system space, the monitoring system adopts the data processing technology of partitioned storage and cyclic coverage. By combining a variety of technologies, the trimaran structural monitoring system can monitor the hull longitudinal strength and local structural strength simultaneously, and the number of connected monitoring sensors can reach up to 60. In addition, in order to demonstrate the feasibility and effectiveness of this system, the trimaran model equipped with SMAS is tested with different wave azimuths. The real-time fluctuation of strain and sectional monitoring values is recoded in monitoring system. And structural strength assessments are worked in the system to evaluate the structural state in short term and long term. Through comparison and analysis of the assessment results in different wave azimuths, the dangerous positions and its structural damage cause are found. Finally, the measured data and assessment result will become an important foundation on trimaran detailed optimization design.
2. Design of structural monitoring and assessment system

2.1 Overall design of structural monitoring and assessment system

The overall design of structural monitoring and assessment system is a key part in the system design. It can not only affect the system computing efficiency, but also determines the scalability of its functions. In order to make the monitoring system run and extend more efficiently, the multi-level and modular design method is applied in the overall design of trimaran monitoring system, as seen in Figure 1. Based on the structural monitoring purposes, the whole monitoring system is divided into three modules, which are real-time structural monitoring module, structural strength assessment module and structural monitoring database module. Each module owns its sub-modules to implement their specific functions.

![Real-time structural monitoring module]

**Fig. 1 Scheme of monitoring system**

In real-time structural monitoring module, the real-time data of structure response is focused on. The fluctuation of structural response like strain can be recorded and showed by curve graph and histogram. The preliminary analysis and measured data warning is also taken in this module. In preliminary analysis, the statistical characteristic value such as significant amplitude values, maximum and mean value is calculated and guided to help crew understand the basic parameters of structure response more clearly. When the structural responses are violent and beyond to designed alarms threshold, the warning message will appear and point out the warning location. And the out-of-range proportion on the warning threshold is record and store into structural monitoring database. In structural strength assessment module, all programs are worked around calculation and display about the structural strength state. The structural strength assessments in this module include the longitudinal structural strength assessment, fatigue structural strength assessment and simplified yield strength assessment. The characteristics of trimaran structural strength state in different time-domain are recoded by these methods. The short-term structure state is evaluated by longitudinal structural strength assessment. And the instantaneous dangerous structure state is recoded by simplified yield strength assessment. The potential damage of structural strength state in long term is also monitored by the fatigue structural strength assessment. Structural monitoring database is...
an assemblage of measured structural response and strength assessment results. The ship basic information and system operation record is also added in this database. Finally, this database forms a complete logbook about trimaran’s structural state.

Through the actual ship test and preliminary research, it is known that the simple single-threaded calculation method often takes too much time and limits the number of monitoring sensors. In order to make the trimaran monitoring system work more efficiently, multi-threading parallel computing method is adopted. Figure 2 shows the multi-threading scheme of the monitoring system. Five threads are designed for the trimaran monitoring system. They are data collection and display thread, data extraction and storage thread, longitudinal structure strength thread, simplified yield strength thread and fatigue structure strength thread. Each thread extracts and analyses the related data independently, then finishes its own function.

The real-time structural monitoring module is mainly supported by the data collection and display thread. Firstly, this thread receives and classifies the measured data from the strain sensors by using the Transmission Control Protocol (TCP)/IP services. Through combining with the correlation coefficient, the electrical signals are converted into the monitoring variables such as strain, bending moment and torque. And these real-time monitoring variables are shown in histogram or diagram to express the fluctuation of structural response directly. The maximum and mean value is also calculated at same times. Then, the judgement of data warning is taken. If the monitoring indicators appear large values, the data collection and display thread will feed back to data extraction and storage thread. Finally, the data extraction and storage thread will extract corresponding location information and give warning toward crew. In structural strength assessment module, the three kinds of structural strength assessment use their own thread. There is no interference between these threads, so it improves the calculation speed of assessment system. The longitudinal strength thread is based on the strength reliability method to assess hull sectional structure state at different longitudinal positions. The strength thread counts time and calculates the structural failure probability every 30 minutes. The simplified yield strength thread is aimed to evaluate instantaneous structure state. The max strain value in preliminary analysis is transmitted by data collection and display thread to simplified yield strength thread. And the instantaneous yield indictor is calculated and classified into corresponding grades. The fatigue structural strength thread is based on the Palmgren-Miner damage rule, and run itself every 5 minutes. The potential damage of structural fatigue failure is measured by the fatigue cumulative damage. The fatigue damage in 5 min is calculated and added with previous fatigue damage.
to become final fatigue cumulative damage in fatigue structural strength thread. In addition, all assessment results will be written into database by their own threads. In structural monitoring database, the duty of data extraction and storage thread is to write and update the data in structural monitoring database. All query and extraction of data from monitoring database is worked by this thread. The extraction and storage thread also record the crew system operation and abnormal monitoring data for subsequent accident analysis. Through using these threads, the monitoring system can reflect the structure state and deal with some emergency incident quickly. In fact, the max sampling frequency of SMAS can reach 1000Hz. And the fluctuation of sensor sampling frequency, which is from 10Hz to 1000Hz, is also supported automatically by this trimaran monitoring system. Figure 3 shows the designed results of the structural monitoring and assessment system.

![Figure 3](image)

**Fig. 3** Structural monitoring and Assessment System for trimaran

2.2 Structural strength assessment principle in monitoring system

Structural strength assessment based on real-time data has always been a difficult point in the field of structural monitoring. Based on the research of structural real-time monitoring technology, this paper gives some real-time structural strength assessment methods for trimaran structure. These methods include the deterministic assessment method and reliability assessment method. And, considering characteristics of different structural damage form, the system adopts three methods to achieve different structural strength monitoring targets.

2.2.1 Simplified yield strength assessment

In yield strength assessment, a simplified deterministic evaluation method is studied in the paper. This method requirement of data quantity is relatively small, and the computing speed is fast. Therefore, this assessment method is beneficial to reflect instantaneous structure condition. Some critical structures, which suffer the structural response frequently and violently, are suitable to use this assessment method. In simplified yield strength assessment, the instantaneous yield indictor $S(t)$ records the drastic structural state fluctuation and
occurrence time. This indicator is calculated as a ratio between the max value of real-time measured stress $\sigma_m(t)$ and the allowable stress of material $[\sigma_s]$, as shown in Eq. (1).

$$S(t) = \frac{\sigma_m(t)}{[\sigma_s]}$$  \hspace{1cm} (1)

Due to differences in material production, the yield limit of the material is not always invariant constant. It is generally considered that the yield limit of the material fits Gaussian distribution. According to the statistical theory on Gaussian distribution, the probability of yield limit in the area of $(\mu-2\sigma, \mu+2\sigma)$ is 95.44%. Obviously, the possibility of yielding failure on structure will increase greatly, when the real-time measured stress come into this area. Therefore, the allowable stress of material $[\sigma_s]$ is often below to yield limit of material, and is defined as follows.

$$[\sigma_s] = \mu - 2\sigma$$  \hspace{1cm} (2)

where $\sigma$ is standard deviation on material yield limit. And $\mu$ is mean value on material yield limit. Then, the structural safety classification based final instantaneous yield indicator is taken. The specific ranges of different grades are shown in Table 1.

<table>
<thead>
<tr>
<th>Safety grade</th>
<th>Range of instantaneous yield indicator</th>
</tr>
</thead>
<tbody>
<tr>
<td>Class A</td>
<td>0% ~ 1%</td>
</tr>
<tr>
<td>Class B</td>
<td>1% ~ 10%</td>
</tr>
<tr>
<td>Class C</td>
<td>10% ~ 100%</td>
</tr>
<tr>
<td>Class D</td>
<td>&gt;100%</td>
</tr>
</tbody>
</table>

2.2.2 Longitudinal strength assessment

For longitudinal strength assessment, a kind of structural strength reliability method is studied in this paper to evaluate the sectional structure condition. In order to observe the monitoring structural response in real time, the sensors are often installed on hull girder or some longitudinal stiffened panels at different longitudinal positions. The relationship between monitoring structure response and sensor signals is built by some calibration experiments [15]. And the structural section response like bending moment and torque can be monitored. In structural reliability method, the structural response caused by wave impact is representing as $D$. And $C$ is function of the structural resistant ability. When $C$ and $D$ are independent random variables, the structural failure function $M$ can be defined as follows.

$$M = C - D$$  \hspace{1cm} (3)

Obviously, the structure breaks down, when the structure failure function $M$ is less than 0. Therefore, structural failure probability $P_f$ is calculated out by Eq. (4).

$$P_f = P(M < 0) = \int_{-\infty}^{\infty} [1 - F_D(x)]f_C(x)dx$$  \hspace{1cm} (4)

Where $x$ is the value of structural response, $f_C(x)$ is the probability density function on the structural resistant ability, and $F_D(x)$ is the distribution function on extreme value of structural response. According to the statistical data of wave load, the short term distribution on the amplitude of structural response often follows the Rayleigh distribution. So the extreme value distribution of structural response is achieved on the foundation of Rayleigh distribution. The Rayleigh distribution for structural response in short term is shown in Eq. (5).
And the computational method on the corresponding parameter of Rayleigh distribution $k$ is also given in Eq. (6).

$$F_d = 1 - \exp\left(-\frac{x^2}{k^2}\right)$$ \hspace{1cm} (5)

$$k^2 = 2\sigma_d^2$$ \hspace{1cm} (6)

Where $\sigma_d$ is the standard deviation of real-time structure response, and it is statistically calculated every 30 minutes. The cyclic number of real-time data $n$ is also counted. According to the principle of sequence statistics, the distribution of the extreme value on structural response can be obtained as follows.

$$F_D = \left[1 - \exp\left(-\frac{x^2}{k^2}\right)\right]^n$$ \hspace{1cm} (7)

The ability of structural resistance is considered generally to obey Gaussian distribution. Therefore, the probability density function on the structural resistant ability $f_c(x)$ is calculated by Eq. (8).

$$f_c = \frac{1}{\sqrt{2\pi}\sigma_c} \exp\left(-\frac{(x - \mu_c)^2}{2\sigma_c^2}\right)$$ \hspace{1cm} (8)

$$\mu_c = \mu_i - d_i$$ \hspace{1cm} (9)

Where $\sigma_c$ is the standard deviation of the probability density function on the structural resistant ability; $\mu_c$ is mean value of the probability density function; $d_i$ is the fluctuation of monitoring data caused by other factors such as sensor temperature, changing of ship loading status and the initial stress in material. The mean value $\mu_c$ can be obtain by subtracting $d_i$ from section ultimate strength $\mu_i$ as Eq. (9).

**Fig. 4** Results of designed structure strength

In fact, the trimaran ultimate strength at different cross-sections needs to consider many factors such as structural failure forms and material yield characteristics, so the accuracy of ultimate strength calculation is difficult to guarantee. And the Rule for the Classification of Trimaran is issued by the Lloyd’s Register (LR) in 2006, it is unique complete rule for trimaran design [16]. Most trimarans are designed and built on the basis of this rule. In LR’s trimaran rule, the calculation formula of hull structure strength is given. And the requirements of hull structural strength in each cross-section must be satisfied. Therefore, the ultimate
fatigue at cross-section in this paper is simplified and calculated by LR’s rule. Figure 4 show the designed structural ultimate strength in different locations. Then, the mean value of structural resistance can be obtained by eliminating non-wave disturbance. And the coefficient of variation on structural resistance distribution is assumed as 8%, according to previous research on hull ultimate strength [17]. The structural response ability function at different cross-sections is established. Finally, the longitudinal strength assessment in monitoring system works successfully. Figure 5 shows the process of longitudinal strength assessment in monitoring system.

![Longitudinal Strength Assessment Diagram](image)

**Fig. 5** Process of longitudinal strength evaluation

2.2.3 Fatigue strength assessment

The structural fatigue failure is easy to occur, when the ship suffers the high-frequency wave load. It is also a kind of potential structural failure form that is often overlooked. Nowadays many ship accidents are found to be caused by structural fatigue. Therefore, the structural fatigue monitoring is indispensable. Based on the Palmgren-Miner damage rule and the rain flow counting method, a kind of real-time fatigue strength assessment is studied.

In fact, the fluctuation of stress is always irregular and random, when ship sails in the complex irregular wave condition. Therefore, the rain flow counting method is needed to transform the random stress into a series of variable amplitude stresses. And the characteristic parameter of these variable amplitude stresses, which includes the cyclic number, amplitude and mean value, is also calculated out by the rain flow counting method. Because the S-N curve method is based on symmetrical stress cycle condition, these measured asymmetric stresses need to be corrected in further. So the Gerber correction method, which is applied widely in ship engineering, is used. The equation of the Gerber correction is as follows.

$$\frac{S_u}{S'_r} + \frac{S_m^2}{S'_r} = 1 \tag{10}$$

Where $S_u$ is the amplitude of measured stress; $S_r$ is the corrected amplitude of stress under symmetrical cycle; $S_m$ is mean value of measured stress; And $S_r$ is ultimate strength stress, it is depended on material yield limit. The corrected amplitude of stress under symmetrical cycle $S'_r$ is calculated by solving the Eq.(10). Then, the maximum cyclic number of structural fatigue limit under the corrected amplitude stress $N$ can be calculated by the S-N curve method as follows.

$$\log(N) = \log(K) - m\log(S'_r) \tag{11}$$
Where $K$ is the $S$-$N$ curve parameter. And $m$ is the inverse slope of the $S$-$N$ curve. These parameters of $S$-$N$ curve are determined by the material fatigue test [18] and guide for fatigue strength of hull structures in China Classification Society [19]. Then, the ratio between the measured cyclic number $n_i$ and the cyclic number of the corresponding fatigue limit $N_i$ is calculated on each series of variable amplitude stresses. And the final fatigue cumulative damage $D_j$ is achieved by summing these ratios, as shown as Eq. (12).

$$D_j = \sum_{i=1}^{j} \frac{n_i}{N_i}$$  \hspace{1cm} (12)

Thereinto, $j$ is the number of the series of variable amplitude stresses; $n_i$ is the measured cyclic number on the $i$th series of variable amplitude stresses, it is obtained by rain flow counting method; $N_i$ is the cyclic number of the corresponding fatigue limit on the $i$th stress level. The whole calculation process of fatigue strength assessment is shown at Fig. 6.

![Fig. 6 Process of fatigue strength assessment](image)

**2.3 Establishment of structural monitoring database**

To store monitoring data more conveniently, based on the Structured Query Language (SQL) server, a kind of data extraction and storage method is developed to establish a trimaran structural monitoring database. In order to record the trimaran structural information as much detail as possible, seven tables are designed in this monitoring database, as seen in Figure 7. Each table consists of time data and corresponding monitoring data.

![Fig. 7 Structure diagram of monitoring database](image)
The data extraction and storage thread plays a critical role in the data extraction and storage method for trimaran structural monitoring. It will be able to write the data into the corresponding table, and also extract the data from these tables when the monitoring data is inquired. In long-term structural monitoring, the huge quantity of monitoring data like real-time measured data is high possible to lower the system querying speed. Therefore, in order to avoid this influence, the partition function is used in each table. The storage space of table is divided into 48 regions. All region addresses are recorded by the partition function, so the system can find the corresponding storage location quickly. When all region storage is full, the thread will create some specified documents to output these data one region by one region. At same time, the method of cyclic covering region is also done to keep the number of regions unchanged. After the data output, the early stage region is regard as a new empty region to restart storage for the latest monitoring data. Certainly, the partition function still mark the outputted data addresses and can help the thread to find corresponding data in stored documents. Through system running test in long-term (six months), it is verified that the design of data storage improves the efficiency on data querying, and make the system speed of data extraction less than 15s averagely. Figure 8 gives some results of the structural monitoring database design in SMAS.

![Real-time monitoring data record](image1)

![Fatigue strength assessment record](image2)

**Fig. 8** Design of structural monitoring database

3. **Design of trimaran structural monitoring model**

In order to observe the characteristics of structural response in irregular waves and demonstrate the feasibility on structural monitoring system, a structural monitoring model is designed with the scale ratio of 1 to 25 by the model similarity theory [20-22]. The trimaran model had a series of segmented shells, which would offer similar wave force and impact to simulate the wave influence toward full-scale trimaran. And the disturbance of surround flow field caused by propeller is also considered by the installation of self-propulsion equipment in this trimaran model.

<table>
<thead>
<tr>
<th>Description</th>
<th>Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall length (m)</td>
<td>5.64</td>
</tr>
<tr>
<td>Waterline length (m)</td>
<td>5.44</td>
</tr>
<tr>
<td>Moulded breadth (m)</td>
<td>1.06</td>
</tr>
<tr>
<td>Depth (m)</td>
<td>0.47</td>
</tr>
<tr>
<td>Draft (m)</td>
<td>0.2</td>
</tr>
<tr>
<td>Displacement (t)</td>
<td>0.255</td>
</tr>
</tbody>
</table>

**Table 2** Principal dimension of the trimaran model
In addition, there are two kinds of steel backbones systems in the model. The longitudinal steel backbone system includes ten steel girders with different thickness to meet the stiffness distribution along the ship length. And transverse steel backbone system has two girders to ensure the transverse stiffness of cross-bridge structure. The proper placement of iron is taken to satisfy the similarity on mass distribution between the model and full-scale ship. Finally, the principal dimension of the trimaran model is designed and shown in table 2. And Figure 9 also shows the detailed design of trimaran model.

In Figure 9, it is observed that the model is divided into seven segments along the ship length. In order to measure the wave-induced hydroelastic vibration on the trimaran model, the interval in each segment maintain at 1cm. And the thin latex is used to keep the watertight in each interval. The positions of these intervals are introduced at table 3. And FP denotes fore perpendicular.

<table>
<thead>
<tr>
<th>Division</th>
<th>Location from FP(m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>M1</td>
<td>1.02</td>
</tr>
<tr>
<td>M2</td>
<td>1.56</td>
</tr>
<tr>
<td>M3</td>
<td>2.04</td>
</tr>
<tr>
<td>M4</td>
<td>2.72</td>
</tr>
<tr>
<td>M5</td>
<td>3.19</td>
</tr>
<tr>
<td>M6</td>
<td>4.28</td>
</tr>
</tbody>
</table>

In fact, these interval positions are also the main structural monitoring points in the model test. The strain sensors are installed on the longitudinal circular hollow girders at each interval to monitoring the structural strain in model test. And the method of Wheatstone bridge was applied on the arrangement of the strain gauges to measure vertical bending moment (VBM), horizontal bending moment (HBM) and torque. The specific monitoring sensor layout and corresponding circuit diagram on the three structural deformations are shown respectively in Figure 10(a)-(c). In order to avoid the possible wire drag and water injection in model test, the silicon rubber is used around these monitoring sensors. And the installation process of these strain sensors is also seen in Figure 10(d). At same time, the relationship between the monitoring value and the output voltage of strain sensors is obtained by the static calibration experiment. Figure 11 shows the process and the results of static calibration experiment at bow.
Design and Model Test of Structure Monitoring and Assessment System for Trimaran

Haoyun Tang, Huilong Ren, Qi Zhong

(a) Sensor arrangement for VBM

(b) Sensor arrangement for HBM

(c) Sensor arrangement for torque

(d) Arrangement of sensors

Fig. 10 Arrangement and installation of strain gauges

In Figure 11 (a), it can be seen clearly that the measured steel girder in trimaran model was fixed with orthogonal metal frame at one end. At another end of steel girder, some weights were putted to form bending deformation or torsional deformation on the measured girder. The theoretical bending moment and torque at the monitoring position with different weights is calculated, and the output voltage of the monitoring sensors is also observed. By using linear fitting method, the corresponding correlation coefficients on the relationship between the sensor’s output voltage and monitoring values like bending moment or torque are found out.

Fig. 11 Static calibration experiment

Figure 11 (b) shows the result of static calibration experiment at M1. The similar method is used to obtain corresponding correlation coefficient at other monitoring positions. And the correlation coefficient on surface strains also follows the same process in static calibration experiment.
4. Model test and monitoring results analysis

The model test was taken in the comprehensive tank of Harbin Engineering University. And the ISSC dual parameter spectrum was used to guide the hydraulically driven wave maker for simulation of irregular waves. In model test, the typical irregular wave environment for full-scale trimaran, whose statistical significant wave height was 4m and zero-crossing period was 6.7s, was mainly considered. By changing the model orientation, the measurement of model structural response in different wave azimuths was achieved. In addition, the model self-propulsion equipment was controlled to make the relative position invariant with trailer. Each irregular wave case lasted 2 hours to give enough time for structural strength assessment in SMAS. Figure 12 shows the process of model test.

![Fig. 12 Process of trimaran model test](image)

4.1 Model test results analysis

4.1.1 Preliminary analysis of structural responses

When the trimaran model suffers wave impact in the irregular waves, the real-time structural deformation at different sections is recorded by the monitoring system. The real-time monitoring data is converted by the similitude rule and then is taken into the statistical analyses in trimaran monitoring system.

![Fig. 13 Time histories of structural response in bow sea](image)
Figure 13 (a) shows the vertical bending moment along the ship length in bow sea condition. Obviously, the real-time VBM fluctuations at these three positions are similar. And the time of max peak appearing in these positions is not far. Also, it is not hard to find that the VBM data at M1 and M6 contains more high-frequency fluctuation, although the mean amplitude of VBM at M4 is biggest in these monitoring positions. In fact, the wave slamming at bow and stern often causes the structural high-frequency vibration. And the self-propeller equipment exacerbates the high-frequency vibration at stern. In Figure 13 (b), the three kinds of structure deformations at M3 are also shown. It is found that the structural response of horizontal bending deformation and torsional deformation fluctuate more frequently than that of the vertical bending deformation in oblique waves. In addition, the structural response amplitude of VBM amidships is still bigger than amplitude of horizontal bending moment (HBM) and torque in bow sea case.

In fact, the measured structural responses are often random and irregular, when trimaran is sailing in irregular waves. Therefore, the statistical characteristic value is always used as an indicator to judge the severity of structural response in the whole irregular wave case. The autocorrelation function based on spectral analyses was conducted to obtain the significant amplitude values in monitoring system. And the results of significant amplitude values with different wave azimuths and deformation forms are shown in Figure 14. The wave azimuth of heading wave condition is defined as 180 deg. And the wave azimuth in following wave is 0 deg. Similarly, the wave azimuth case at 135 deg is representative of the bow sea condition. And the wave azimuth of quartering sea case is 45 deg.

![VBM in different wave azimuths](image1)
![HBM in different wave azimuths](image2)
![Torque in different wave azimuths](image3)

**Fig. 14** Structural responses with different wave azimuths

Obviously, the VBM reaches minimum in beam sea condition, as seen in Figure 14(a). And the vertical bending deformation in bow sea is more severe than the deformation in quartering sea. The severest vertical bending deformation is found at M5, it is the bow end of the cross-bridge. In fact, the flow field interference of side hull, especially wave slamming at side hulls, aggravates the vertical bending deformation at M5. And the position of max vertical bending deformation is moved a little toward stern. In Figure 14(b), it is observed easily that the trimaran in bow sea suffers more horizontal bending deformation among these irregular wave cases. The position of max HBM is not stable and changes with different wave azimuths. In bow sea condition, the HBM get the max peak at M5 like vertical bending moment. And the max HBM position is replaced by M3 in quartering sea. The arrangement of
side hulls at stern improves the structural strength and weakens the structural response caused by wave impact from stern direction, in some extent. The statistical characteristic value of torque in different wave azimuths is also shown in Figure 14(c). Through observation, it is found that the trimaran will suffer more torsional deformation when ship is sailing in beam sea. And the position of M5 and M6, whose location contains the side hulls, get more violent torque in irregular waves. It is explained that the unique structural features of side hulls make wave force asymmetric along the width, and causes more violent torsional deformation in oblique irregular waves, especially in beam sea.

4.1.2 Analysis of longitudinal structural strength assessment results

The preliminary analysis mainly focuses on the structural response caused by wave impact, so it just can indicate the hull structure condition roughly. And the varied structural resistance ability is also another indispensable factor to determine structural damage risk. In fact, it is high possible to appear this phenomenon that the structure break causes by low structure resistance, although the structure suffers small wave load. In order to understand real-time hull structural condition more exactly, the differences of the longitudinal structural resistance ability are needed to be considered. The longitudinal structural strength assessment combines the wave load and structural resistance ability to evaluate the hull structural condition, and it is regarded as a more comprehensive mean to monitor the real-time hull structural condition.

The trimaran monitoring system takes the longitudinal strength thread to evaluate the structural longitudinal strength along ship length every 30 minutes. Figure 15 shows the results of this longitudinal strength assessment in 2 hours. Because the horizontal bending and torsional deformation in heading sea and following sea is too low to motivate severe structural damage risk, Figure 15 (d) only shows the assessment results of VBM in non-oblique wave cases. Although there is a high structural resistance amidships, the structural failure probability at M5 is still highest in the longitudinal strength assessment of VBM. Regardless of the different wave azimuths, the dangerous section on vertical bending deformation is invariable and consistent with the peak location of load statistical characteristic value. It is explained that the violent wave load response is still the main factor to cause the high failure probability at the most dangerous position of vertical bending deformation. But the position of second severest wave load response (M4) doesn’t always get the second highest failure probability with the comparison of VBM’s assessment results in different wave azimuths. In fact, the position M3 has relative lower structural resistance and is evaluated as second dangerous position in bow sea and following sea, as shown in Figure 15(a) and Figure 15(d). Therefore, the influence of low structural resistance toward the failure probability is distinct at the position with mild wave load response. In addition, the highest failure probability at M5 is happened in heading sea condition, so the heading sea condition is regarded as the most dangerous structural case for vertical bending deformation in irregular waves.

For horizontal bending deformation, the highest failure probability is found at M3 in different oblique wave cases. It is also the comprehensive result of wave load response and structural resistance. Although the position M3 is not always suffered severest wave load response, the relative lower horizontal bending structural resistance make this section more perishable at horizontal. In bow and quartering sea, the position of weak horizontal bending resistance (M2) often achieves high hull structural failure probability. And the violent wave load response at M5 also makes the structure amidships more fragile in beam sea. By observing the failure probability of horizontal bending moment in Figure 17, it is found that the highest failure probability of horizontal bending deformation is 1.86%. And it is happened in bow sea. Therefore, the bow sea condition is regarded as the most concerned case for horizontal bending deformation.
Fig. 15 Results of longitudinal strength assessment in different wave azimuths

The hull structural strength assessment of torsional deformation is also important in oblique irregular waves. Owing to the structure of side hulls and cross bridge, the trimaran gets more wave load along ship width and suffers severe torsional deformation in oblique wave cases. Although the structure at stern suffers relatively severe deformation, M3 is still the most dangerous position on torsional deformation by observing the results of structural strength assessment in monitoring system. Obviously, the common influence between wave load response and structural resistant strength makes the dangerous position move toward bow. In addition, the high failure probability at M2 also indicates the importance for structure design on torsion resistance near bow. The highest failure probability of torque is 3.47E-09, and it is found in beam sea. So, the most dangerous case for torsional deformation is beam sea. In oblique waves, the peak of failure probability on HBM and torque is often at the same time period. It suggests that the wave load response on torsional and horizontal deformation is associated lightly in time domain. By comparison of the failure probability on these three destruction forms, it is also found that the failure probability of HBM is higher than the failure probability of other deformation in oblique wave cases, although the significant amplitude value of HBM is lower than that of VBM in bow sea and quartering sea. And the failure probability of torque is relatively low among most irregular wave cases. Thus, it is
known that the structural horizontal deformation is dangerous for the present trimaran design in oblique waves, while the torsional deformation on structure is relatively safe.

4.1.3 Analysis of simplified yield strength assessment results

The strain on steel girders is also monitored by the structural monitoring system for the simplified yield strength assessment and fatigue strength assessment. And Figure 16 shows the time histories of strain amidships in quartering sea.

![Vertical deformation and Transverse deformation](image)

**Fig. 16** Time histories of strain in quartering sea

Obviously, the similar irregular oscillation on strain is recorded in vertical deformation and transverse deformation. At the same monitoring position, the peak appearing time of strain on these two deformations is close. The real-time strain is converted to stress in monitoring system. And the simplified yield strength assessment is taken to evaluate the instantaneous structural state. Figure 17 shows final results of simplified yield strength assessment in different wave azimuths.

![Simplified yield strength evaluation](image)

**Fig. 17** Simplified yield strength assessment results

In fact, the simplified yield strength assessment results both on vertical and transverse deformation fall into the region of Class A and Class B. In Figure 17(a), it is not hard to found that the vertical instantaneous yield indictor at M5 is always highest in different wave azimuths. And the max value of simplified yield strength assessment result is 6.51%. The bow is least possible to have yield deformation in irregular waves, especially in beam sea. The
instantaneous yield indicator at M1 falls into Class A, only in beam sea case. For structural transverse states, the instantaneous yield indicator of all positions in heading sea and following sea is in Class A, as seen in Figure 17(b). Non-oblique irregular wave case is too hard to motivate yield destruction on transverse. The max value of instantaneous yield indicator on transverse is 3.28% at M3, and it is caught in bow sea. But this transverse dangerous position is varied and changed as M5 in beam sea. By comparison in assessment results of vertical and transverse deformation, the transverse instantaneous yield indicator is still smaller than the vertical indicator. The instantaneous structural break should be paid more attention on vertical deformation.

4.1.4 Analysis of fatigue strength assessment results

Fatigue failure is a kind of imperceptible structural failure form. By the potential damages accumulation in long term, the flaw initiates and propagates on the surface of critical structure. With the development of flaw, the hull structure will lose its own structural resistance and break down by small wave impact in final. Therefore, in order to find the structural fatigue failure early, the critical structure of trimaran needs to be monitored in long term. And potential damages should be predicted and warned toward crew. Based on real-time monitored strain, the fatigue failure damage is calculated by the fatigue strength assessment thread in monitoring system every 5 minutes. The process of fatigue damage accumulation at vertical and transverse surface of longitudinal steel backbone is recorded in 2 hours. And these results of fatigue strength assessment on vertical and transverse are shown in Figure 18 and Figure 19, respectively.

In Figure 18, it is found easily that all time histories of fatigue accumulative damage in different wave azimuths are not linear relationship. It is obvious that the randomness of wave impact in irregular waves brings the inhomogeneous increasing on fatigue accumulative damage. Although the increased amplitude of fatigue accumulative damage in different monitoring positions is diverse, the increasing trend on fatigue accumulative damage is consistent roughly in time domain. In quartering sea, the fatigue accumulative damage at all positions increases quickly at the time period from 60 min to 65 min, as seen in Fig.18 (d). The similar phenomenon can be also found in other irregular waves cases. By comparison of vertical fatigue accumulative damage in different monitoring positions, it is found that the vertical fatigue accumulative damage at M4 increases most fast in beam sea, and this damage fastest increasing position is replaced by M5 in other azimuth cases. In fact, the position M5 always obtain higher amplitude of strain on vertical deformation than other positions in irregular waves. Therefore, the vertical fatigue accumulative damage at M5 enhances acutely in most irregular wave cases. The position M4 suffers more cycle number of vertical strain than the cycle number at M5 in beam sea, and strain amplitude gap between M4 and M5 is not far. Thus, the M4 finally become the dangerous position of fatigue strength in beam sea. This difference of strain cycle number is also explained by the phenomenon that the trimaran cross-bridge structure bring some structural vibration into the main hull, when the side hull blocks the wave spreading in beam sea.

Figure 18 (e) shows the final fatigue accumulative damage at four dangerous positions. Obviously, the max value of fatigue accumulative damage is happened at M5 in heading sea. So the position at the start end of side hulls in heading sea should be considered carefully in the design of structural fatigue resistance. Although the amplitude of strain on vertical deformation at M3 is higher than the strain amplitude at M2, the bow slamming motivate more higher frequency vertical vibration and make M2 suffers fatigue damage more violently than M3 by comparing the fatigue damage in Figure 18(a)-(d). Thus, the influence of high
frequency vibration cause by wave slamming is also important in fatigue strength assessment and cannot be ignored.

![Fatigue damage graphs](image)

(a) Fatigue damage in heading sea  
(b) Fatigue damage in bow sea  
(c) Fatigue damage in beam sea  
(d) Fatigue damage in quartering sea  
(e) Final fatigue damage results

**Fig. 18** Vertical fatigue strength assessment

Because of weak transverse vibration in head sea and following sea, the structural fatigue breaks on transverse is too hard to form. Therefore, Figure 19(d) only shows the results of transverse fatigue strength assessment in oblique waves.
Through observation of the transverse fatigue cumulative damage at different monitoring locations, it is found that the gap of transverse fatigue damage among M3, M4 and M5, is not far. And the highest fatigue damage appears at these three positions alternately in quartering sea, bow sea and beam sea. Obviously, the most dangerous position on transverse fatigue breaks is unfixed, and the high fatigue damage position is sensitive with wave azimuths. Even in same wave condition, the high fatigue damage position has high possible to change at different times. In the beam sea case, the M3 replace M5 as highest fatigue damage position at 70 min, but the situation turn back at 120 min. This replacement of max dangerous position is observed clearly in Fig.19 (b). In quartering sea, the similar fluctuation is also found by comparing the fatigue damage of M3 and M4 in Figure19 (c). The bow sea condition is considered as the most dangerous cases for transverse fatigue damage by the final fatigue damage statistic results in Figure 19 (d). And the transverse position at M4 should be focused as the important location to avoid the transverse structural fatigue breaks.

5. Conclusions

In this paper, a kind of structural monitoring and assessment system for trimaran is designed and installed in a self-propelled model to study the fluctuation characteristics of trimaran structural state in irregular waves. The trimaran model is equipped with some strain

---

**Fig. 19 Transverse fatigue strength assessment**

(a) Fatigue damage in bow sea  
(b) Fatigue damage in beam sea  
(c) Fatigue damage in quartering sea  
(d) Final fatigue damage results
sensors along the ship length and tested in different wave azimuths. Meanwhile, three real-time structural strength assessment methods are adopted in the monitoring system to evaluate the trimaran structural state at different monitoring locations and irregular wave cases. The final results of these structural strength assessments in the trimaran monitoring system is compared and analysed. And the conclusions are made as follows:

(1) The structural monitoring and assessment system for trimaran is proved to achieve well the purpose of trimaran structure safety monitoring by a series of model tests. The real-time strain and sectional indicator is monitored and stored in the system database successfully. And the simplified yield strength assessment in the system can record instantaneous structural state effectively. The overall and local structural condition of trimaran is also evaluated autonomously in the system module of longitudinal and fatigue strength assessments. And all these monitoring and assessment data by this monitoring system will give the foundation of the actual trimaran structural monitoring.

(2) The location at bow end of side hull is regarded as the most dangerous position for vertical bending deformation. Although the distribution of longitudinal structural resistance is considered in longitudinal structural strength assessment, the violent wave load response is still found as the main factor to determine this dangerous position of vertical bending deformation. But, for horizontal bending and torsional deformation, the structural resistance has played an unignored role on selecting the dangerous position. In longitudinal structural strength assessment of horizontal bending moment and torque, the trimaran arrangement of side hull at stern improves the stern structural resistance and decreases structural failure possibility at stern. Therefore, the dangerous position of horizontal bending and torsional deformation is near to the trimaran bow relatively.

(3) In simplified yield strength assessment of the monitoring system, the instantaneous yield indictors in both vertical and transverse deformation fall into Class A and B. It is suggest that this trimaran structural instantaneous condition is relatively safe. The peak of vertical yield indictor is always observed at the bow end of side hull and unvaried at different environments, while the peak location of transverse yield indictor is changed with different wave azimuths. Obviously, the structural yielding failure on the local structure of trimaran is sensitive to wave azimuths. Thus, the probability distribution of wave direction encountered by trimaran should be considered in the specific selection of structural yield strength monitoring points for the full-scale trimaran structural monitoring.

(4) In irregular waves, the increasing of fatigue accumulative damage in fatigue strength assessment are not linear, but the increasing trend of fatigue damage at different monitoring positions is consistent roughly on time domain. The fastest vertical fatigue accumulative damage is also found at bow end of side hull. And the dangerous position of horizontal fatigue damage in oblique waves is varied in the midship region. For the horizontal structural fatigue failure, the bow sea case should be taken more attention during the trimaran voyage.

In present research, the main monitoring subjects focus on the trimaran longitudinal positions. The cross-bridge structural monitoring and wave impact will be considered in further research. And the inverse finite element method will be adapted to predict of the local structural stress [23-24]. The full-scale or large-scale trimaran test in actual sea environment will be also taken to make the monitoring technology more credible.

ACKNOWLEDGMENTS

Special thanks to all members in ship structural mechanics laboratory of Harbin Engineering University and Dalian Maritime University for the preparation of this trimaran test. This research was supported by the National Natural Science Foundations of China
(No.51679049 and No.51709030) and the Fundamental Research Funds for the Central Universities (3132019134). The authors express their gratitude to these foundations.

REFERENCES

https://doi.org/10.1016/j.oceaneng.2016.04.025

https://doi.org/10.1016/j.oceaneng.2015.11.032
The classical method (namely inclining experiment) has been used to estimate the vertical center of gravity (VCG or more often KG) of ships for many years. This method is based on the assumption that the metacenter is unchanged in the calculation of KG when the vessel is heeled. However, ships built today have knuckles, chines, dead-rise which may give rise to excessive change in the water-plane. The location of the metacenter is changed on these vessels when heeled. Therefore, determination of the vertical center of gravity may be somewhat erroneous. In this study, three different methods based on the assumption of unchanged metacenter have been examined. Employing the Graphical method, Polar method and Generalized method, the KG values of all vessels can be calculated without any dependence on the metacenter. The three methods mentioned in this study were studied and applied on ten different ships. Furthermore, the results from the classical method compared with those obtained from the recently developed methods. Based on this comparison, it is observed that the new methods developed have similar results to the classical method. Therefore, these methods may be a good alternative to the inclining experiment in the future. Moreover, uncertainty analyses have been performed for the results obtained from the classical method just to realize if there are any critical GM values within the margin of error.

Keywords: Vertical center of gravity; Metacenter; Graphical method; Generalized method; Polar Method; Uncertainty analysis

1. Introduction

For the time being, the inclining experiment is the principal method for determining center of gravity of a ship. After a vessel is launched, an inclining experiment is performed to calculate especially lightship characteristics of the vessel, KG in particular. Likewise, this test should be repeated for existing ships following any major over-haul that will alter ship’s weight considerably.
The basis of the inclining experiment calculation, which is also known as the classical method, dates back to the end of 17th century. Firstly, Hoste, a professor of mathematics, suggested the concept of inclining experiment in 1697 [1]. Furthermore, a practical method for the implementation of inclining experiment was defined by Bouguer in 1746 [2]. Using this classical method, GM₀ of ships are to be obtained directly.

The classical method contains some assumptions intrinsically. Therefore, the validity of the classical method has been under discussion for many years. For small angles of heel, the immersed and emerged volumes are equal to each other when heeled. It is assumed that the metacenter is unchanged. The position of the metacenter depends on the unchanged water-plane area, equation (1).

\[ KM \quad KB + BM = KB + \frac{I_{WP}}{\text{V}} \]

The classical method is limited to small angles of heel. Especially, ships built in recent years have chines, knuckles and deadrise that can cause certain deviations in the metacenter. Alternative methods in calculating KG that has a great importance on ship stability have been emerged and improved in recent years.

One of the alternative methods of calculating KG of ships is the Graphical method proposed by O.O. Kanifolskyi and M.M. Konotopets in 2016 [3]. The Generalized method was proposed by R.J. Dunworth in 2013 and improved by R.J. Dunworth and A.C. Smith [5-8]. Another alternative method, the Polar method was proposed quite recently and refined by K. B. Karolius and D. Vassalos [4, 11, 12]. These alternative methods do not rely on the metacenter.

Since KG of ships is one of the most important governing parameter in stability and in turn safety, determination of it accurately is of great importance. Criticism on the inclining test has been around for so many years. Thus, pursuit for a more reliable method has gained momentum in recent years. In this study, KG of 10 vessels of various types are calculated by using three above-mentioned alternative methods and the results are compared with those from the classical method in order to lay out the differences if any. Additionally, uncertainty analyses have been employed on all 10 vessels to pinpoint the possible sources of error result from the classical method.

2. The Inclining Experiment (Classical Method)

The inclining experiment is carried out to find the vertical center of gravity of ships. The test is based on the measurement of angle of inclinations caused by moving the weights transversely placed on the deck. In some cases, ballast water may be used instead of solid weights as an alternative. In this case, selected tanks should be symmetrical (PS/SB) and trim of the ship should not change.

The inclining experiment is mandatory by the IMO’s 2008 IS Code for every commercial ship above 24 m in length and all passenger ships.

2.1 Application

During the test, the vessel should float freely in water and mooring lines should be slack. The ship is then heeled by movements of test weights. These deflections are recorded by various means such as U-tubes, inclinometers or pendulums. If the inclining experiment is carried out using solid test weights, 8 shifts should be conducted as a standard required by 2008 IS Code. A sample sequence of the shifts are shown in Table 1.
Table 1 Weight shift sequence

<table>
<thead>
<tr>
<th>Shift</th>
<th>Shift of weights</th>
<th>PS</th>
<th>SB</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>w2, w4, w6</td>
<td>w1, w3, w5</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>w4, w6</td>
<td>w1, w2, w3, w5</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>w1, w2, w3, w4, w5, w6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>w6</td>
<td>w1, w2, w3, w4, w5</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>w2, w4, w6</td>
<td>w1, w3, w5</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>w1, w2, w3, w4, w6</td>
<td>w5</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>w1, w2, w3, w4, w5, w6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>w1, w2, w4, w6</td>
<td>w3, w5</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>w2, w4, w6</td>
<td>w1, w3, w5</td>
<td></td>
</tr>
</tbody>
</table>

2.2 Determination of KG

Since GM of a ship is obtained by the classical method directly, KG is calculated using equation (2).

\[ KG = KM - GM \]  
\[ (2) \]

According to equation (1), the metacenter remains constant. KM in equation (2) is acquired from the hydrostatics at the test loading condition.

Ideally, prior to the inclining experiment, the ship should be in equilibrium with the weights placed symmetrically on deck without any heel and trim. When a weight is moved to a distance \( d \), the center of gravity shifts from \( G \) to \( G_1 \), Fig. 1. Then \( \tan \phi \) is given by:

\[ \tan \phi = \frac{G G_1}{G M} \]  
\[ (3) \]

Re-arranging this equation, GM is obtained.

\[ GM = \frac{G G_1}{\tan(\phi)} \]  
\[ (4) \]

Fig. 1 Shift of center of gravity
When a weight is moved, the center of gravity is shifted accordingly. In this case, \( GG_1 \) is expressed by equation (5).

\[
GG_1 = \frac{w.d}{\Delta}
\]  

(5)

Re-arranging equation (4), the final equation is obtained for GM.

\[
GM = \frac{w.d}{\Delta \tan(\phi)}
\]  

(6)

In this expression, \( \tan\phi \) is calculated according to equation (7) for each pendulum and \( \tan\phi \) values calculated for each movement are averaged to minimize the error.

\[
\tan(\phi) = \frac{\text{pen.deflection}}{\text{pen.length}}
\]  

(7)

tan\( \phi \) and moment values for all weight shift cases are calculated by the above equations. Then, \( \frac{w.d}{\Delta} \) is plotted against the corresponding \( \tan\phi \) for each weight shift and a linear line passing through these points is fitted according to the least squares method. The slope of the linear line gives GM. Alternatively, \( \frac{w.d}{\Delta} \) is plotted against \( \tan\phi \) for each weight shift, as shown in Fig. 2, and GM is calculated according to equation (8).

\[
GM = \frac{1}{\Delta R_{\text{slope}}}
\]  

(8)

![Fig. 2 Tan\( \phi \)-moment linear line](image)

In case of slack liquid in tanks, it is necessary to introduce a free surface correction. Free surface moment (FSM) is calculated as in equation (9).

\[
FSM = \frac{1}{12.\Delta} \frac{h^3}{4h}
\]  

(9)

Finally, KG is obtained as below.

\[
KG = KM - GM - FSM
\]  

(10)

In addition, LCG and TCG are calculated as below.
Another Blow on the Torn Down Wall-The Inclining Experiment

Selahattin Ozsayan, Metin Taylan

\[ LCG = LCB - \frac{Trim.MCT.100}{\Delta} \]  

\[ TCG = TCB_{\phi=0} + \tan(\phi).GM \]  

3. New Alternative Methods

3.1 Generalized method (Dunworth method)

Generalized method was first proposed by R.J. Dunworth and A.C. Smith [3]. After each weight shift, the vessel reaches an equilibrium. Therefore, the vessel’s righting and heeling moments must be equal.

\[ H_{moment} = R_{moment} \]  

\[ \Delta.HZ = B_{M}.GZ \]  

Since displacement and buoyancy forces are equal, equation (14) reduces to equation (15).

\[ HZ = GZ \]  

Heeling arm (HZ) is obtained as below.

\[ HZ = \frac{w.d.\cos(\phi)}{\Delta} \]  

The trigonometric relationships shown in Fig. 3 are used to obtain the righting arm of the ship.

\[ GZ = KN - KG_{1}\sin(\phi) - TCG_{1}\cos(\phi) \]  

When this equation is re-arranged using equation (15) and equation (16), KG is calculated as follows.

\[ KG_{1}\sin(\phi) = KN - HZ - TCG_{1}\cos(\phi) \]
Equation (18) reduces to equation (20) yielding \( TCG_1 \) when \( \phi = 0 \).

\[
TCG_1 = KN_0 - HZ_0
\]  

(20)

\( KN_0 \) is obtained from the hydrostatics. Dunworth proposed that \( HZ_0 \) is calculated by plotting \( HZ \) against heel angle (\( \phi \)) and a third-order polynomial passing through these points is fitted in accordance with the least squares method. When \( \phi = 0 \), \( HZ_0 \) equals the intercept point on the y-axis of the trend line, Fig. 4 [7].

3.2 Graphical method

Graphical method, which calculates \( KG \) directly, was proposed by Kanifolskyi and Konotopets in [3]. \( KN \) used in the calculation of \( KG \) are acquired from the hydrostatics. Kanifolskyi and Konotopets summarized the calculation of \( KG \) in 5 steps as shown in Fig. 5. The first step is to draw \( KN \) from the keel (K). The second step is to draw a perpendicular to \( KN \). In the third step, \( HZ \) is calculated according to equation (16). In the fourth step, calculated \( HZ \) is placed in the correct position. That is, \( HZ \) arm is parallel to \( KN \) and intercepts the centerline. In the final step, \( KG \) is read off from the graph. These 5 steps must be repeated for all shifts of weights. In addition, the mean \( KG \) is calculated according to equation (21).

\[
KG = \frac{1}{n} \sum_{i=1}^{n} KG_i
\]  

(21)

As can be seen in Fig. 5, trigonometric relationships are driven from the sketch. \( KG \) is calculated according to the following sequence of equations:

\[
KG = \frac{KN - HZ}{\sin(\phi)}
\]  

(22)

\[
KG \cdot \sin(\phi) = KN - HZ
\]  

(23)
Another Blow on the Torn Down Wall-The Inclining Experiment  
Selahattin Ozsayan, Metin Taylan

\[ KG \cdot \sin(\varphi) = KN - \frac{w_d \cdot \cos(\varphi)}{\Delta} \]  \hspace{1cm} (24)

![Graphical method steps](image1)

**Fig. 5** Graphical method steps [4]

3.3 Polar method

Polar method is also a new method proposed by K. Karolius and D. Vassalos [4,11,12]. In this method, a line parallel to BM radius is considered. This line is shifted up to distance HZ, and represented in polar coordinates. In the initial condition, KG and TCG are located on this line. Moreover, the location of KG and TCG does not changed when each weight is shifted. Initial KG0 and TCG0 are fixed on this line. However, TCG is shifted up a distance G0Gi when each weight is shifted, Fig. 6.

![The parameters of Polar method](image2)

**Fig. 6** The parameters of Polar method [4]

The equation of the straight line is given below.
\[ z = x \cos(\phi) + y \sin(\phi) \]  

Equation (25) is rearranged to get equation (26). When equation (26) and equation (27) are combined, equation (28) is attained.

\[ z = TCG \cos(\phi) + KG \sin(\phi) \]  

\[ z = KN - HZ \]  

\[ KN - HZ = TCG \cos(\phi) + KG \sin(\phi) \]  

As mentioned above, \( KG_i \) and \( TCG_i \) for each shift \( i \) must be equal to \( TCG_0 \) and \( KG_0 \) respectively.

\[ TCG_0 = TCG_i \]  

\[ KG_0 = KG_i \]  

When equation (28) is reorganized using equations (29) and (30), \( KG \) and \( TCG \) are found as follows:

\[ KG = \frac{(KN_i - HZ_i) \cos(\phi_0) - (KN_0 - HZ_0) \cos(\phi_i)}{\cos(\phi_0) \sin(\phi_i) - \sin(\phi_0) \cos(\phi_i)} \]  

\[ TCG = \frac{(KN_i - HZ_i) \sin(\phi_0) - (KN_0 - HZ_0) \sin(\phi_i)}{\cos(\phi_i) \sin(\phi_0) - \sin(\phi_i) \cos(\phi_0)} \]  

Equation (29) and (30) are general forms which may be simplified according to state of the vessel during the inclining experiment. For example, if heeling arm in the initial position is zero, \( HZ_0 \) vanishes. Furthermore, if the initial heel angle \( (\phi_0) \) is zero and ship is completely symmetrical, \( KN_0 \) is also equal zero. Therefore, equations (29) and (30) are simplified as below.

\[ KG = \frac{(KN_i - HZ_i)}{\sin(\phi_i)} \]  

\[ TCG = 0 \]  

The above equations are the same equations seen in Graphical method. Therefore, the Graphical method is valid only for completely symmetrical vessels which have a limited heel angle in the initial condition. The \( KN \) used in the Polar method are to be obtained from a 3D stability software model of the ship. But they would still introduce additional errors. On the other hand, \( HZ_0 \) is calculated as described in Section 3.1.

4. Sample Ships

In this study, 10 different ships having different sizes, hull forms and types with approved inclining experiment results have been used in order to observe the effect of various parameters on the results by these different methods.

This set of sample ships is comprised of two chemical tankers, two asphalt tankers, a service boat, a tug, a Ro-Ro, a container ship, a research vessel and a fast boat. The axis convention applied throughout this study is given in Table 2. The general characteristics of these vessels are supplied in Table 3.
5. Adoption of New Methods

The calculation of KG is performed according to the methods described in Section 3 on the sample vessels. The details of the calculations are given for research vessel only as an example in the following subsections respectively. The same procedures have been applied to the other ships for all methods but only the final results were revealed for space limitation.

5.1 The graphical method

The graphical method KG calculations are shown below.

<table>
<thead>
<tr>
<th>Shift</th>
<th>tanφ</th>
<th>φ</th>
<th>Σφ</th>
<th>φ</th>
<th>HZ</th>
<th>KN</th>
<th>Sinφ</th>
<th>KG.Sinφ</th>
</tr>
</thead>
<tbody>
<tr>
<td>mean</td>
<td>degree</td>
<td>degree</td>
<td>radian</td>
<td>m</td>
<td>m</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>-0.0175</td>
<td>0</td>
<td>-0.0754</td>
<td>-0.0754</td>
</tr>
<tr>
<td>1</td>
<td>0.0164</td>
<td>0.9423</td>
<td>-0.0577</td>
<td>-0.0010</td>
<td>0.0068</td>
<td>-0.0043</td>
<td>-0.0010</td>
<td>-0.0112</td>
</tr>
<tr>
<td>2</td>
<td>0.0304</td>
<td>1.7428</td>
<td>0.7428</td>
<td>0.0130</td>
<td>0.0137</td>
<td>0.0560</td>
<td>0.0130</td>
<td>0.0423</td>
</tr>
<tr>
<td>3</td>
<td>-0.0164</td>
<td>-0.9423</td>
<td>-1.9423</td>
<td>-0.0339</td>
<td>-0.0069</td>
<td>-0.1465</td>
<td>-0.0339</td>
<td>-0.1396</td>
</tr>
<tr>
<td>4</td>
<td>-0.0006</td>
<td>-0.0335</td>
<td>-1.0335</td>
<td>-0.0180</td>
<td>0.0000</td>
<td>0.0779</td>
<td>0.0180</td>
<td>0.0779</td>
</tr>
<tr>
<td>5</td>
<td>-0.0149</td>
<td>-0.8553</td>
<td>-1.8553</td>
<td>-0.0324</td>
<td>-0.0068</td>
<td>-0.1399</td>
<td>-0.0324</td>
<td>-0.1331</td>
</tr>
<tr>
<td>6</td>
<td>-0.0201</td>
<td>-1.6663</td>
<td>-2.6663</td>
<td>-0.0465</td>
<td>-0.0137</td>
<td>0.0465</td>
<td>-0.0137</td>
<td>-0.1874</td>
</tr>
<tr>
<td>7</td>
<td>0.0153</td>
<td>0.8753</td>
<td>-1.1247</td>
<td>-0.0022</td>
<td>0.0069</td>
<td>-0.0094</td>
<td>-0.0022</td>
<td>-0.0163</td>
</tr>
<tr>
<td>8</td>
<td>0.0000</td>
<td>0.0000</td>
<td>-1.0000</td>
<td>-0.0175</td>
<td>0.0000</td>
<td>-0.0754</td>
<td>-0.0175</td>
<td>-0.0754</td>
</tr>
</tbody>
</table>

tanφ values given in Table 4 were found from the ratio of pendulum deviations to the pendulum length from the inclining experiment report. Since three pendulums were used in the inclining experiment, readings from these three pendulums were averaged. In the fourth column, heel angle of the third column and initial heel angle are collected. In sixth column, HZ
is calculated according to equation (16). KN in Table 4 is obtained from ship’s stability booklet. Finally, KG.sinφ is calculated according to equation (23).

Results that are obtained from Table 4 is plotted in Fig. 8. Slope of the linear line provides KG. Free surface correction, weights to be added and removed for calculations of lightship KG are shown in Table 5.

![Fig. 7 Research vessel Graphical method, KG.sinφ vs sinφ]

**Table 5 Research vessel-Graphical method KG value**

<table>
<thead>
<tr>
<th></th>
<th>Weight (t)</th>
<th>KG (m)</th>
<th>FSM (ton.m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship as inclined</td>
<td>1167.602</td>
<td>3.8706</td>
<td>104.025</td>
</tr>
<tr>
<td>FS corrections</td>
<td>-0.089</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fluid KG</td>
<td>1167.602</td>
<td>3.782</td>
<td></td>
</tr>
<tr>
<td>Total items to remove</td>
<td>-295.85</td>
<td>2.051</td>
<td></td>
</tr>
<tr>
<td>Lightship</td>
<td>871.75</td>
<td>4.369</td>
<td></td>
</tr>
</tbody>
</table>

5.2 The Generalized method

**Table 6 Research vessel-Generalized method procedure**

<table>
<thead>
<tr>
<th>Shift</th>
<th>tanφ</th>
<th>φ</th>
<th>Σφ</th>
<th>φ</th>
<th>HZ</th>
<th>KN</th>
<th>Sinφ</th>
<th>KG.Sinφ</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>mean</td>
<td>degree</td>
<td>degree</td>
<td>radian</td>
<td>m</td>
<td>m</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.0000</td>
<td>0.0000</td>
<td>-1.0000</td>
<td>-0.0175</td>
<td>0.0000</td>
<td>-0.0754</td>
<td>-0.0175</td>
<td>-0.0678</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0.0164</td>
<td>0.9423</td>
<td>-0.0577</td>
<td>-0.0010</td>
<td>0.0068</td>
<td>-0.0043</td>
<td>-0.0010</td>
<td>-0.0036</td>
</tr>
<tr>
<td>2</td>
<td>0.0304</td>
<td>1.7428</td>
<td>0.7428</td>
<td>0.0130</td>
<td>0.0137</td>
<td>0.0560</td>
<td>0.0130</td>
<td>0.0499</td>
</tr>
<tr>
<td>3</td>
<td>-0.0164</td>
<td>-0.9423</td>
<td>-1.9423</td>
<td>-0.0339</td>
<td>-0.0069</td>
<td>-0.1465</td>
<td>-0.0339</td>
<td>-0.1320</td>
</tr>
<tr>
<td>4</td>
<td>-0.0006</td>
<td>-0.0335</td>
<td>-1.0335</td>
<td>-0.0180</td>
<td>0.0000</td>
<td>-0.0779</td>
<td>-0.0180</td>
<td>-0.0703</td>
</tr>
<tr>
<td>5</td>
<td>-0.0149</td>
<td>-0.8553</td>
<td>-1.8553</td>
<td>-0.0324</td>
<td>-0.0068</td>
<td>-0.1399</td>
<td>-0.0324</td>
<td>-0.1255</td>
</tr>
<tr>
<td>6</td>
<td>-0.0291</td>
<td>-1.6663</td>
<td>-2.6663</td>
<td>-0.0465</td>
<td>-0.0137</td>
<td>-0.2011</td>
<td>-0.0465</td>
<td>-0.1798</td>
</tr>
<tr>
<td>7</td>
<td>0.0153</td>
<td>0.8753</td>
<td>-1.1247</td>
<td>-0.0022</td>
<td>0.0069</td>
<td>-0.0094</td>
<td>-0.0022</td>
<td>-0.0087</td>
</tr>
<tr>
<td>8</td>
<td>0.0000</td>
<td>0.0000</td>
<td>-1.0000</td>
<td>-0.0175</td>
<td>0.0000</td>
<td>-0.0754</td>
<td>-0.0175</td>
<td>-0.0678</td>
</tr>
</tbody>
</table>
The generalized method utilizes the spreadsheet shown in Table 6. The steps in the table are the same as the graphical method except the last column. In the last column, KG.sin\(\phi\) is calculated according to equation (18). Fig. 8 is used to obtain HZ in this equation. Using a third order polynomial approximation, HZ\(_0\) is found to be -0.0023 m.

Graphical representation of the results in Table 6 is depicted in Fig. 9. The slope of the linear line gives KG. Free surface correction, weights to be added and removed for the lightship KG are shown in Table 7.

\[
y = 0.0002x^3 + 0.0005x^2 + 0.0078x + 0.0076
\]
\[
R^2 = 0.9993
\]

Fig. 8 Research vessel-Generalized method, HZ vs \(\phi\)

\[
y = 3.8707x - 0.0002
\]
\[
R^2 = 1
\]

Fig. 9 Research vessel-Generalized method, KG.sin\(\phi\) vs sin\(\phi\)
Table 7 Research vessel-Generalized method, KG value

<table>
<thead>
<tr>
<th></th>
<th>Weight (t)</th>
<th>KG (m)</th>
<th>FSM (ton.m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship as inclined</td>
<td>1167.602</td>
<td>3.8707</td>
<td>104.025</td>
</tr>
<tr>
<td>FS corrections</td>
<td></td>
<td>-0.089</td>
<td></td>
</tr>
<tr>
<td>Fluid KG</td>
<td>1167.602</td>
<td>3.782</td>
<td></td>
</tr>
<tr>
<td>Total items to remove</td>
<td>-295.85</td>
<td>2.051</td>
<td></td>
</tr>
<tr>
<td>Lightship</td>
<td>871.75</td>
<td>4.369</td>
<td></td>
</tr>
</tbody>
</table>

5.3 Polar method

The procedure to calculate KG in Polar method is shown in Table 8. The steps up to last two columns are similar to the graphical and generalized methods. Unlike other two methods, \( \sin(\phi_i - \phi_0) \) is used. KG.sin(\( \phi_i - \phi_0 \)) in the last column is calculated according to equation (31). HZ\(_0\) in equation (31) is the same as the Generalized method and equal to -0.023m.

Table 8 Research vessel-Polar method procedure

<table>
<thead>
<tr>
<th>Shift</th>
<th>tan( \phi )</th>
<th>( \phi )</th>
<th>( \Sigma \phi )</th>
<th>( \phi )</th>
<th>HZ</th>
<th>KN</th>
<th>Sin( \phi )</th>
<th>sin(( \phi_i - \phi_0 ))</th>
<th>KG.sin(( \phi_i - \phi_0 ))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.0000</td>
<td>0.0000</td>
<td>-1.0000</td>
<td>-0.0175</td>
<td>0.0000</td>
<td>-0.0754</td>
<td>-0.0175</td>
<td>0.0000</td>
<td>-0.0678</td>
</tr>
<tr>
<td>1</td>
<td>0.0164</td>
<td>0.9423</td>
<td>-0.0577</td>
<td>-0.0010</td>
<td>0.0068</td>
<td>-0.0043</td>
<td>-0.0010</td>
<td>0.0164</td>
<td>-0.0036</td>
</tr>
<tr>
<td>2</td>
<td>0.0304</td>
<td>1.7428</td>
<td>0.7428</td>
<td>0.0130</td>
<td>0.0137</td>
<td>0.0560</td>
<td>0.0130</td>
<td>0.0304</td>
<td>0.0499</td>
</tr>
<tr>
<td>3</td>
<td>-0.0164</td>
<td>-0.9423</td>
<td>-1.9423</td>
<td>-0.0339</td>
<td>-0.069</td>
<td>-0.1465</td>
<td>-0.0339</td>
<td>-0.0164</td>
<td>-0.1320</td>
</tr>
<tr>
<td>4</td>
<td>-0.0006</td>
<td>-0.0335</td>
<td>-1.0335</td>
<td>-0.0180</td>
<td>0.0000</td>
<td>-0.0779</td>
<td>-0.0180</td>
<td>-0.0006</td>
<td>-0.0703</td>
</tr>
<tr>
<td>5</td>
<td>-0.0149</td>
<td>-0.8553</td>
<td>-1.8553</td>
<td>-0.0324</td>
<td>-0.0068</td>
<td>-0.1399</td>
<td>-0.0324</td>
<td>-0.0149</td>
<td>-0.1255</td>
</tr>
<tr>
<td>6</td>
<td>-0.0291</td>
<td>1.6663</td>
<td>-2.6663</td>
<td>-0.0465</td>
<td>-0.0137</td>
<td>-0.2011</td>
<td>-0.0465</td>
<td>-0.0291</td>
<td>-0.1797</td>
</tr>
<tr>
<td>7</td>
<td>0.0153</td>
<td>0.8753</td>
<td>-0.1247</td>
<td>-0.0022</td>
<td>0.0069</td>
<td>-0.0094</td>
<td>-0.0022</td>
<td>0.0153</td>
<td>-0.0087</td>
</tr>
<tr>
<td>8</td>
<td>0.0000</td>
<td>0.0000</td>
<td>-1.0000</td>
<td>-0.0175</td>
<td>0.0000</td>
<td>-0.0754</td>
<td>-0.0175</td>
<td>0.0000</td>
<td>-0.0678</td>
</tr>
</tbody>
</table>

After KG.sin(\( \phi_i - \phi_0 \)) is calculated, results are plotted in Fig.10. Slope of linear line gives KG. Finally, free surface correction, weights to be added and removed for the lightship KG are shown in Table 9.

Table 9 Research vessel-Polar method KG value

<table>
<thead>
<tr>
<th></th>
<th>Weight (t)</th>
<th>KG (m)</th>
<th>FSM (ton.m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship as inclined</td>
<td>1167.602</td>
<td>3.8692</td>
<td>104.025</td>
</tr>
<tr>
<td>FS corrections</td>
<td></td>
<td>-0.089</td>
<td></td>
</tr>
<tr>
<td>Fluid KG</td>
<td>1167.602</td>
<td>3.780</td>
<td></td>
</tr>
<tr>
<td>Total items to remove</td>
<td>-295.85</td>
<td>2.051</td>
<td></td>
</tr>
<tr>
<td>Lightship</td>
<td>871.75</td>
<td>4.367</td>
<td></td>
</tr>
</tbody>
</table>
6. Comparison and Analysis of the Results

In this section, KG of 10 different vessels obtained from four different methods are presented in Table 10. KG obtained according to the classical method is taken as a reference. The items shown as difference are the difference between the KG values obtained from the alternative methods and by the classical method.

When differences are examined, the highest difference was 8.4% between Graphical and Generalized methods in research vessel. In polar method, this difference is 8.3%. In chemical tanker (1), the difference in three methods is obtained as -6.4%. For tugboat, similar to chemical tanker (1) in the Graphical method and Generalized methods, however, a positive 6.4% difference was calculated. This difference is 6.3% in polar method. For asphalt tanker (1), in the polar and graphical method, the difference is in the order of -5.1% and in the Generalized method in the order of -5.0%. In the case of fast boat, the difference is calculated as -5.1% in the polar method and -5.0% in the graphical and Generalized methods. The difference for the container ship is 4.5% in all methods. Asphalt tanker (2) shows a difference of -3.8% in all methods. In Ro-Ro vessel and service boat, the difference is -1.8% in all methods evaluated except for polar method. This difference is -1.9% in polar method. Finally, the closest result is calculated with 0.2% difference in all methods for chemical tanker (2).

Small differences between 3 methods indicate that these methods can be applied to different ships having different hull geometries, sizes and characteristics with a good level of confidence. Graphical representation of comparison of all three methods is shown in Fig. 11.
Table 10 Comparison of results by methods

<table>
<thead>
<tr>
<th>Sample Ships</th>
<th>Classical</th>
<th></th>
<th></th>
<th>Graphical</th>
<th></th>
<th></th>
<th>Polar</th>
<th></th>
<th></th>
<th>Generalized</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>KG (m)</td>
<td>KG (m)</td>
<td>Diff. (mm)</td>
<td>Diff. (%)</td>
<td>KG (m)</td>
<td>KG (mm)</td>
<td>Diff. (%)</td>
<td>KG (m)</td>
<td>KG (mm)</td>
<td>Diff. (%)</td>
<td>KG (m)</td>
<td>KG (mm)</td>
</tr>
<tr>
<td>Chemical Tanker (1)</td>
<td>7.449</td>
<td>6.969</td>
<td>-480</td>
<td>-6.4</td>
<td>6.969</td>
<td>-480</td>
<td>-6.4</td>
<td>6.969</td>
<td>-480</td>
<td>-6.4</td>
<td>6.969</td>
<td>-480</td>
</tr>
<tr>
<td>Asphalt Tanker (1)</td>
<td>8.466</td>
<td>8.034</td>
<td>-431</td>
<td>-5.1</td>
<td>8.038</td>
<td>-428</td>
<td>-5.1</td>
<td>8.038</td>
<td>-427</td>
<td>-5.0</td>
<td>8.038</td>
<td>-427</td>
</tr>
<tr>
<td>Service Boat</td>
<td>1.663</td>
<td>1.632</td>
<td>-31</td>
<td>-1.8</td>
<td>1.632</td>
<td>-31</td>
<td>-1.9</td>
<td>1.633</td>
<td>-30</td>
<td>-1.8</td>
<td>1.633</td>
<td>-30</td>
</tr>
<tr>
<td>Asphalt Tanker (2)</td>
<td>8.501</td>
<td>8.177</td>
<td>-324</td>
<td>-3.8</td>
<td>8.176</td>
<td>-325</td>
<td>-3.8</td>
<td>8.177</td>
<td>-323</td>
<td>-3.8</td>
<td>8.177</td>
<td>-323</td>
</tr>
<tr>
<td>Chemical Tanker (2)</td>
<td>8.153</td>
<td>8.170</td>
<td>17</td>
<td>0.2</td>
<td>8.169</td>
<td>16</td>
<td>0.2</td>
<td>8.170</td>
<td>17</td>
<td>0.2</td>
<td>8.170</td>
<td>17</td>
</tr>
<tr>
<td>Fast boat</td>
<td>1.294</td>
<td>1.228</td>
<td>-66</td>
<td>-5.1</td>
<td>1.228</td>
<td>-66</td>
<td>-5.0</td>
<td>1.228</td>
<td>-66</td>
<td>-5.1</td>
<td>1.228</td>
<td>-66</td>
</tr>
</tbody>
</table>

7. Uncertainty Analysis

There are a number of uncertainties in the inclining experiment conducted for the determination of the vertical center of gravity (KG) of ships. Since it involves certain measurements and human interference, this experiment more likely contains some errors. These uncertainties and errors start with draft readings before and after the test and continue with the
assumptions made during the calculations. The following may be shown as sources of error; wind, current, wave, human error, errors of measurement devices etc. All these errors may accumulate from the beginning to the end of the inclining experiment and may result in significant nonconformities in KG and GM.

It is assumed that metacenter is fixed when the vessel is heeled. However, most of the ships that are built today having knuckles, chines and dead-rise which result in deviations in the water-plane area. This in turn causes errors in the lightship KG and GM.

On the contrary, three new methods discussed in this study are not based on the assumption that the metacenter is fixed. Therefore, the uncertainties caused by the change of metacenter are eliminated.

7.1 Uncertainty analysis procedure and results

In this section, a range of source of uncertainties and errors such as draft reading, displacement, heel angle, wind, wave etc. are examined. The effect of these uncertainties has been applied to all ships used in the study [10]. Uncertainties in inclining experiment have been investigated by many other researchers such as Wilczynski et al [13] and Woodward et al [14]. They examined the uncertainty in GM which results from the bias and accuracy errors pertinent to each measured and calculated variable.

As a procedure, the uncertainty in the slope of the best fit linear line is determined first. Then, the uncertainty in displacement is calculated according to equation (8). Uncertainties of vessel draft, calculated molded volume, density, molded vs. as-built volume are reflected to the uncertainty of the displacement. The uncertainty in GM (U_{GM}) is found after the uncertainty of displacement (U_\Delta) and slope of line (U_{slope}) are determined.

\[
\left( \frac{U_{GM}}{GM} \right) = \sqrt{\left( \frac{U_{slope}}{slope} \right)^2 + \left( \frac{U_\Delta}{\Delta} \right)^2}
\]

Or, in shorthand;

\[
U_{GM} = \sqrt{U_{slope}^2 + U_\Delta^2}
\]

Since KG is composed of various hydrostatic parameters, the uncertainty in KG (as inclined) is given as;

\[
U_{KG} = \sqrt{U_{KB}^2 + U_{BM}^2 + U_{GM}^2}
\]

BM on the other hand is the ratio of moment of inertia and volume having the uncertainty of;

\[
U_{BM} = \sqrt{U_i^2 + U_v^2}
\]

Similarly, uncertainty of each parameter can be determined using its constituents in the same manner as explained above.

Next, errors from KB, BM and experimental weights are calculated. Finally, Uncertainty of lightship KG is determined. The details of uncertainty analysis applied to chemical tanker (1) are given in Table 11 below.
Table 11 Result of the uncertainty analysis for research vessel

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Measured/calculated value</th>
<th>Uncertainty</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Slope (tangent/moment)</td>
<td>265.550</td>
<td>0.316</td>
<td>l/t.m</td>
</tr>
<tr>
<td>Molded vs. as-built volume (v)</td>
<td>1143.587</td>
<td>0.043</td>
<td>m³</td>
</tr>
<tr>
<td>Vessel drafts</td>
<td>3.948</td>
<td>0.019</td>
<td>m</td>
</tr>
<tr>
<td>Calculated molded volume (m)</td>
<td>1143.587</td>
<td>22.872</td>
<td>m³</td>
</tr>
<tr>
<td>Displacement volume (V)</td>
<td>1143.587</td>
<td>23.612</td>
<td>m³</td>
</tr>
<tr>
<td>Density</td>
<td>1.021</td>
<td>0.002</td>
<td>t/m³</td>
</tr>
<tr>
<td>Displacement (∆)</td>
<td>1167.602</td>
<td>24.171</td>
<td>t</td>
</tr>
<tr>
<td>As- inclined GM</td>
<td>0.445</td>
<td>0.009</td>
<td>m</td>
</tr>
<tr>
<td>As- inclined KG</td>
<td>3.529</td>
<td>0.076</td>
<td>m</td>
</tr>
<tr>
<td>Lightship KG</td>
<td>4.031</td>
<td>0.183</td>
<td>m</td>
</tr>
</tbody>
</table>

Note: All uncertainties are at 95% confidence level.

GM= 0.445±0.009 m
KG= 4.031±0.183 m

The same procedure has been applied to all sample vessels and tabulated results are supplied in Table 12 below. Uncertainties in GM and KG are also depicted graphically in Fig.12 and Fig.13 respectively.

Table 12 Comparison of uncertainty analyses for sample ships

<table>
<thead>
<tr>
<th>Sample Ships</th>
<th>GM (m)</th>
<th>error (± m)</th>
<th>KG (m)</th>
<th>error (± m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical tanker-1</td>
<td>4.206</td>
<td>0.09</td>
<td>7.449</td>
<td>0.47</td>
</tr>
<tr>
<td>Chemical tanker-2</td>
<td>2.971</td>
<td>0.06</td>
<td>8.153</td>
<td>0.34</td>
</tr>
<tr>
<td>Asphalt tanker-1</td>
<td>2.30</td>
<td>0.05</td>
<td>8.466</td>
<td>0.31</td>
</tr>
<tr>
<td>Asphalt tanker-2</td>
<td>7.441</td>
<td>0.16</td>
<td>8.501</td>
<td>0.31</td>
</tr>
<tr>
<td>Service boat</td>
<td>1.266</td>
<td>0.04</td>
<td>1.663</td>
<td>0.09</td>
</tr>
<tr>
<td>Tug</td>
<td>1.998</td>
<td>0.04</td>
<td>3.425</td>
<td>0.11</td>
</tr>
<tr>
<td>Ro-Ro</td>
<td>1.256</td>
<td>0.03</td>
<td>9.413</td>
<td>0.29</td>
</tr>
<tr>
<td>Research vessel</td>
<td>0.445</td>
<td>0.01</td>
<td>4.031</td>
<td>0.18</td>
</tr>
<tr>
<td>Fast boat</td>
<td>0.828</td>
<td>0.04</td>
<td>1.294</td>
<td>0.14</td>
</tr>
<tr>
<td>Container</td>
<td>2.44</td>
<td>0.05</td>
<td>9.926</td>
<td>0.35</td>
</tr>
</tbody>
</table>

Uncertainty analysis provides a possible margin of error which is inherent to inclining experiment. Table 12 depicts the errors in GM and KG for the sample vessels. This may be important especially for ships whose margin of stability is critical.

Wilczynski [13] also provided a sample calculation on the uncertainty of GM of an OSV having length 44.5 m. and GM = 6.593 m. from the test. He obtained ±0.122 m. uncertainty in GM which is quite comparable with the uncertainties in chemical tanker (1) and asphalt tanker (1). Woodward et al [14] compared the uncertainties of four ships (with a fixed reference GM=0.15 m.): buoy tender 0.15±0.15 m., superyacht 0.15±0.033m., supply ship 0.15±0.047m.,
containership 0.15±0.029m. and ropax 0.15±0.077m. These results are also in line with the results obtained for the sample vessels in Table 12.

![Uncertainties in GM](image1)

**Fig. 12 Uncertainty in GM**

![Uncertainties in KG](image2)

**Fig. 13 Uncertainty in KG**

8. **Conclusions**

In this study, Graphical method, Generalized method and Polar method developed in recent years have been applied to ten different sample ships to determine the lightship KG values and the pertinent calculations are presented in details.

When the results of the methods are evaluated, the three anew proposed methods give very similar results for KG. Having compared the classical method results with the newly developed three methods, the biggest difference is seen as 8.4% in the research vessel. This
discrepancy may be attributed to the fact that the research vessel has such a hull form which plays an important role on the water-plane area. The closest results are attained within the range of 0.2% for chemical tanker (2) having a high Cb.

The generalized method reveals slightly different results since it does not take into account the initial heel angle ($\phi_0$). Among the three methods discussed, the most diverse results are obtained from the graphical method. This may be because the graphical method does not take into account the transverse center of gravity (TCG) and the initial heel angle ($\phi_0$).

The outcome of the analysis on the vessels such as asphalt tankers, chemical tanker (2), Ro-Ro and container ship reveals that there is a difference of less than 5% in KG. However, there is a difference in the order of 6-8% in KG calculated with the newly developed methods. An uncertainty analysis has been carried out on the results from the inclining experiment in order to observe whether they fall within the above-mentioned margins. This fact is really important for the vessels having small GM values barely complying with the regulations.

It is the authors’ belief that the difference between the results of the inclining experiment and the results of the other methods taken into consideration emanating from the assumptions made during the conventional inclining experiment in addition to the inherent errors from various sources such as draft, pendulum readings etc.

As a conclusion, certain amount of error is inevitable especially in determining KG from the inclining experiment. Therefore, in order to minimize the error, these proposed methods may be considered as an alternative since they do not depend on the uncertainties confronted in the test. Furthermore, these methods are much easier to apply to any ship regardless of hull form, size or type. However, it is not easy to draw definitive generalized conclusions from this comparative analysis.

Nevertheless, the readers should be bear in mind that the comparative analysis carried out in this study relies on the fact that the results from the inclining experiment are taken as a reference since the absolute error free GM is not known. Obviously, the main objective of this paper is not to determine the exact value of GM but to compare results of various methods using different test vessels. Although, it is mandatory to perform inclining experiment for ships today, it seems feasible that one of these methods may be considered by IMO in the future to supersede the current inclining experiment after further research and benchmark studies.

REFERENCES


Another Blow on the Torn Down Wall-The Inclining Experiment  
Selahattin Ozsayan, Metin Taylan


Submitted: 18.11.2018. Selahattin Ozsayan (e-mail: ozsayan@itu.edu.tr)  
Istanbul Technical University, Department of Naval Architecture and Marine Engineering, Istanbul, Turkey

Accepted: 03.05.2019. Metin Taylan (e-mail: taylan@itu.edu.tr)  
Istanbul Technical University, Department of Naval Architecture and Marine Engineering, Istanbul, Turkey

153
Piston ring material in a two-stroke engine which sustains wear due to catalyst fines

UDC 621.436.1:62-242:620.178.16
Original scientific paper

Summary

This research paper discusses preventive protection of the marine engine against abrasive effects of the catalyst fines in fuel. The relevant facts were gathered while analysing the performance of the slow-speed Wärtsilä RTA engine on an Aframax tanker. The analysis comprised fuel properties, regular and extraordinary examinations of the engine, including the control and replacement of piston rings and the measurement of the liner wear rate. The research particularly focused on the top compression ring as it withstands highest loads and fastest wear. On the other side, the research involved a chemical analysis of the piston rings with the purpose of establishing their chemical properties, structure, coating thickness and micro-hardness. In addition to a number of preventive measures that can be taken on board before injecting the fuel into the combustion chamber, the selection of a piston ring’s material quality affects its wear rate to a large extent. The wear rate also depends on the catalyst fines in fuel. In practice, the piston ring of GGV type proved to be resistant to abrasive wear and its chemical properties were established through laboratory material testing.

Key words: piston ring materials; abrasive wear; catalyst fines; marine diesel engine

1. Introduction

Large marine engines use heavy fuel oil due to its low price. As a complex compound of various hydrocarbons, crude oil has a high content of carbons (83-87%). In petroleum refining, the process of cracking is used to break up heavy hydrocarbon molecules into lighter molecules containing less carbon atoms. The most efficient process is achieved by means of catalysts, i.e. solid compounds containing aluminium or silicon. These are very hard fines that cause abrasion of the engine liners and piston rings. The amount of catalytic fines in fuels is reduced in refineries and their content in fuel is prescribed.

There are a number of harmful effects of catalyst fines in the engine cylinders. Abrasion occurs on the cylinder liner walls if the impurities are thicker than the lubricating oil film. Impurities may get stuck between the piston ring and its groove and they may cut into a liner wall as well. Repair costs may be high and cause the ship’s delay. For example, according to the report made on board a tanker [1], the repair costs that included spare parts, manhours,
engagement of the technical superintendent and additional expenses excluding the charter-related losses, amounted to 500,000 $. Another example [2] of damage caused by catalytic fines refers to the cost of replacing the liner on the nine-liner two-stroke main engine, amounting to 420,000 $. The overall costs arising from the same cause in Sulzer 8RTA84T and MAN B&G 6S50MC engines are presented in Table 1 as the third example.

Table 1. Costs resulting from damage of marine engines due to catalyst fines [3]

<table>
<thead>
<tr>
<th>Ship type</th>
<th>Tonnage</th>
<th>Engine type</th>
<th>Bore</th>
<th>Stroke</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tanker</td>
<td>302,986 DWT</td>
<td>Sulzer 8RTA84T</td>
<td>840 mm</td>
<td>3150 mm</td>
<td>900,000 $</td>
</tr>
<tr>
<td>Bulk carrier</td>
<td>54,000 DWT</td>
<td>MAN B&amp;W 6S50MC-C</td>
<td>500 mm</td>
<td>2000 mm</td>
<td>1,500,000 $</td>
</tr>
</tbody>
</table>

According to the reports presented in the relevant literature [4], the rate of wear of the piston rings and the engine liners of the MAN 6S80ME-C was two times higher than expected, thus increasing maintenance and other related costs by 183,000 $. It should be noted that, due to harmful effects of catalytic fines, it may take just 100 running hours or slightly more than 4 days to reach the upper limit of the wear on all liners in a two-stroke slow-speed engine [5].

As the new international regulations on reduction of maximum allowed sulphur content in fuel are coming into force in 2020, thereby increasing the need for fuel cracking, it is assumed that the amount of fine catalyst residues acting as fuel impurities will be even larger.

Catalytic fines are easily capable of scratching and becoming embedded in steel surfaces. The abrasive property of the aluminium-silicon (Al+Si) fines results from their extreme hardness, measuring up to 8 on the Mohs scale (where diamond is 10) [6]. Marine residual fuels (RM) must meet ISO standard 8217: 2017 (Table 2). According to this standard, fuels that are commonly used in marine engines are allowed to contain maximum 60 mg/kg of catalytic impurities [7]. Engine makers prescribe the allowed maximum of catalytic impurities of 15 mg/kg before injecting the fuel into the engine cylinder.

Refineries are capable of producing fuels with 15 mg/kg of catalytic impurities, but this would make the fuel price considerably higher, so this is not practiced in real life [8].

Table 2. Limits of fuel contents to meet ISO standard 8217: 2017 [9]

<table>
<thead>
<tr>
<th>Limits</th>
<th>Parameters (units)</th>
<th>RMA 10</th>
<th>RMB 30</th>
<th>RMD 80</th>
<th>RME 180</th>
<th>RMG 380</th>
<th>500</th>
<th>700</th>
<th>RMK 380</th>
<th>500</th>
<th>700</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max.</td>
<td>Aluminium + Silicon (mg/kg)</td>
<td>25</td>
<td>40</td>
<td>50</td>
<td>60</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Along with the efficient analyses of marine fuels, there are a number of other procedures and guidelines aimed at reducing fuel impurities, e.g. fuel bunker cleaning, equipment maintenance, crew training, high position of fuel intake from the tank, efficient fuel separation and filtering [10,6]. In addition, some companies perform fuel sampling immediately before and after the fuel separator in order to monitor its efficiency: according to the need or recommendation, another separator is cut in or the engine is switched from heavy fuel oil to diesel oil, as a prevention against the increased maintenance costs due to the wear of the piston rings and cylinder liners.

Another essential requirement for the long durability of the liners and rings is the efficient lubricating system, with the use of adequate lube oil and maintaining the engine
loads within permitted limits. The purposes of lubricants are to protect against wear, reduce friction, separate moving parts, transfer heat (from combustion chamber and piston to liner) and power, reduce noise and vibration, seal for gases, prevent corrosion and carry away contaminants and debris [11]. To increase efficiency [12] to reduce pollution [13] and exhaust emissions [14] in marine two-stroke low-speed diesel engines, it is very important that, among other numerous engine components, that piston rings and cylinder liners have high-quality coatings, i.e. surfaces that generate ultra-low wear and friction between surfaces [15].

The ways of prolonging the durability of the liners and rings, i.e. the causes of shortening their life cycles, depends considerably on the material the piston rings are made of and its ability to withstand the effect of impurities that may appear in the combustion chamber through the scavenge air or fuel.

This paper presents an analysis of the effects of the piston ring material on its wear rate. Given the fact that the chemical composition is not known because the manufacturer cannot present its exact chemical composition and properties, laboratory testing of the ring materials was carried out for the purpose of the research.

2. Piston rings and their function

Piston rings in internal combustion engines have multiple functions. Apart from separating the combustion chamber from the scavenge air space, they ensure the heat transfer from the piston to the cylinder liner and prevent excessive oil to enter the combustion chamber from the crankcase. It is essential for the rings to ensure uniform film of lubricating oil across the cylinder walls. Piston rings include compression and oil control rings, their configuration being presented in Figure 1.

![Figure 1. Design and configuration of the piston rings](image)

2.1 Analysis of the recommended configuration of the piston rings and operation parameters in Wärtsilä RTA 58 marine engine

In this type of engine, the recommended configuration features the top compression piston ring of *Running in coated* (RC) type or the *Angle cut profiled* (KOP) type, its height being 16 mm [16,17]. The lower rings – not being that exposed to high pressure and temperature during the process of combustion – are of the K0 type and *Angle cut* type.
Operation parameters during the process of testing and the data referring to lubricating oil are presented in Tables 3 and 4. The six-cylinder RTA 58 engine used heavy fuel oil (HFO) type IFO350 whose viscosity at 50°C was 352.6 cSt, density 973.5 kg/m³ at 15°C, sulphur content 1.8% and amount of catalytic impurities 20 mg/kg [18].

Table 3. Information on the tested marine engine [19].

<table>
<thead>
<tr>
<th>Maker</th>
<th>Engine type</th>
<th>Nominal / Service power (kW)</th>
<th>Nominal / Service speed (min⁻¹)</th>
<th>Mean indicated pressure (bar)</th>
<th>Stroke (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wärtsilä</td>
<td>6RTA58-T</td>
<td>12000/10800</td>
<td>103 (92)</td>
<td>18.3</td>
<td>2416</td>
</tr>
</tbody>
</table>

Table 4. Information on lubricating oil [19].

<table>
<thead>
<tr>
<th>Cylinder oil</th>
<th>Consumption of cylinder oil (g/kWh)</th>
<th>Crankcase oil</th>
<th>Consumption of crankcase oil (l/day)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Talusia Universal/HR70</td>
<td>1.4</td>
<td>Elf Atlanta Marine 3005</td>
<td>100</td>
</tr>
</tbody>
</table>

During this engine type testing, the analysis involved two types of piston rings, specifically two types of the top rings that bear highest loads, with the code marks G17 SCP1RC16 and GGV SCP1CC16. There were engine problems that occurred while using the G17 RC piston ring due to the piston ring fracture (Figure 2) and the cylinder liner wear (Figure 3). The piston rings were replaced with the new chromium-plated rings of GGV CC type. The liner was replaced as well.

Figure 2. Damaged piston rings

Figure 3 presents the results of the worn-out cylinder liner testing. The liner’s cross-section, with the reference points A – L as points of obligatory measurement, is on the left side. The inserted table shows the distance of the measurement points from the top of the liner. The second and third columns feature the measured values (in yellow colour), whereas the deviations from the boundary values are entered into the last two columns. The measurements revealed that the high rate of wear was observed only at the upper part of the liner, as shown in the diagram on the right side, which conforms to the manufacturer’s guidelines in the manual WinGD for G17 SCP1RC16 and GGV SCP1CC16 piston rings [20].
Piston Ring Material in a Two-Stroke Engine
Which Sustains Wear Due to Catalytic Impurities

Miroslav Vukičević,
Nikola Račić, Špiro Ivošević

A suspicion of whether sudden liner and piston ring wear are caused by cat fines can be examined by taking a replica test of the liner surface (done together with Wartsila Switzerland Ltd. Technical services). The surface where was done replica test is shown in Figure 4 and microscope result in Figure 5.

![Figure 3. Measured wear of the cylinder liner](image1.png)

![Figure 4. Replica location done at the cylinder liner](image2.png)

<table>
<thead>
<tr>
<th>Distance from top</th>
<th>F/A</th>
<th>F/S</th>
<th>Wear (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>150</td>
<td>0.05</td>
<td>0.02</td>
</tr>
<tr>
<td>B</td>
<td>195</td>
<td>2.00</td>
<td>1.00</td>
</tr>
<tr>
<td>C</td>
<td>238</td>
<td>3.38</td>
<td>3.94</td>
</tr>
<tr>
<td>D</td>
<td>295</td>
<td>4.55</td>
<td>3.50</td>
</tr>
<tr>
<td>E</td>
<td>390</td>
<td>3.05</td>
<td>2.00</td>
</tr>
<tr>
<td>F</td>
<td>875</td>
<td>2.27</td>
<td>2.22</td>
</tr>
<tr>
<td>G</td>
<td>1424</td>
<td>0.50</td>
<td>0.51</td>
</tr>
<tr>
<td>H</td>
<td>1970</td>
<td>0.50</td>
<td>0.51</td>
</tr>
<tr>
<td>I</td>
<td>2275</td>
<td>0.50</td>
<td>0.51</td>
</tr>
<tr>
<td>K</td>
<td>2380</td>
<td>0.50</td>
<td>0.51</td>
</tr>
</tbody>
</table>

Max. wear: longitudinal: mm/1000h transverse: mm/1000h
3. **Methodology applied in analysing the chemical composition of the base and layers of the piston rings**

The analysis of the chemical composition of the piston rings’ base and layers included the methods shown in Figure 6. The samples were gathered by the author during the onboard service in the capacity of the chief engineer.
3.1 Chemical analysis of the base and layer material of the rings GGV and G17 by using XRF and SEM techniques

Modern piston rings commonly consist of the base and the layer of coating that is added to improve the ring’s performance. In order to determine their chemical composition, the X-ray fluorescence spectrometry (XRF) analysis was applied. This is a non-destructive technique that has been widely used in scanning electron microscopy instrumentation for elemental analysis of specimens. Obtained results are presented in Table 5.

<table>
<thead>
<tr>
<th>Name of specimen</th>
<th>% C-carbon</th>
<th>% Si-silicon</th>
<th>% Mn-manganese</th>
<th>% S-sulphur</th>
<th>% Cr-chromium</th>
<th>% Ni-nickel</th>
<th>% Cu-copper</th>
<th>% Mo-molybdenum</th>
<th>% V-vanadium</th>
</tr>
</thead>
<tbody>
<tr>
<td>Top ring GGV</td>
<td>3.29</td>
<td>2.0</td>
<td>0.80</td>
<td>0.005</td>
<td>0.11</td>
<td>&lt;0.1</td>
<td>0.80</td>
<td>0.48</td>
<td></td>
</tr>
<tr>
<td>Top ring G17</td>
<td>2.84</td>
<td>1.3</td>
<td>0.80</td>
<td>0.005</td>
<td>0.14</td>
<td>0.40</td>
<td>1.0</td>
<td>0.69</td>
<td>0.17</td>
</tr>
</tbody>
</table>

Iron (ferrous) makes the base of the analysed samples. The data presented in Table 5 show that the specimen of the GGV piston ring has the higher content of carbon (3.29%) and silicon (2%), while there is much less nickel (<0.1%), copper (0.8%) and molybdenum (0.48%). Hence, according to its chemical composition, the base of the piston ring is grey iron. It is actually cast iron whose structure lies in the metal matrix and consists of graphite lamellae. The improvement of wear-resistant properties of a cast iron piston ring-cylinder liner pair of an internal combustion engine can exert a significant influence on maintaining service properties, extending service life, and reducing the repair costs [22,23,24].

Grey iron has found a wide application primarily owing to the affordable property-value ratio. It has a few particular properties that arise from the presence and form of the graphite lamellae.
lamellae within its structure. It can be easily cast, even into complicated forms, and it has good machinability properties. Grey iron has exceptional features for the application in the conditions requiring the smothering vibrations or heat shocks that are present while the engine is running. As the strength and conductivity of this material are limited, they can be improved by adequate heat treatment. This process reduces the size of the lamellae and defines the metal structure.

Parameters which have to be considered for ranking material of the piston ring and cylinder liner in conditions during combustion are: material microstructure, surface topography, the lubricant and the different operating conditions ranging from boundary to hydrodynamic lubrication [25].

When comparing the measured results with the effects of alloyed elements on the properties of steel [26], it can be noted that the specimen Top ring GGV has increased effects of strength, hardness and elasticity when the high contents of carbon and silicon are present. The analysis of the base material’s chemical composition indicates that both samples have almost identical composition, with a difference in the percentage content of carbon (C) and silicon (Si); however, both samples fall into the category of grey iron, as can be noted in Figure 7, points A and B.

![Figure 7](image.png)

**Figure 7.** Content of carbon and silicon in specific types of cast iron- Maurer chart [27]

By using high definition SEM (*scanning electron microscope*), type FEI Sirion 400 NC, the image is obtained by regular movement of the beam of electrons across the surface of the sample. This tool is ideal for examining the structure of materials. When complemented by an energetic dispersive spectrometer EDS Oxford INCA 350, it is possible to establish the chemical composition of the sample material on the selected and observed micro surface that is presented as *spectrum* in the obtained images.
Figure 8. Analysis of elements in the observed areas of the piston ring in layers Spectrum 1 and 2 and base materials Spectrum 3 and 4 [21].

Figure 8 shows the areas selected for chemical analysis, where it is taken into account to select all visible coatings in the piston ring layer. The produced results, presented in Table 6, indicate that the piston ring base is essentially made of iron (96.4% and 68.2%). The neighbouring layer is mostly molybdenum (96%) while the last layer coating the ring is mostly nickel (92.3%). Hence this is the so-called nickel-plated ring.

Table 6. Percentage chemical content of the layer of the top ring G17 [21]

<table>
<thead>
<tr>
<th>Spectrum</th>
<th>In stats.</th>
<th>C</th>
<th>O</th>
<th>Si</th>
<th>Fe</th>
<th>Ni</th>
<th>Mo</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spectrum 1</td>
<td>Yes</td>
<td>5.73</td>
<td>1.93</td>
<td></td>
<td>92.33</td>
<td></td>
<td>100.00</td>
<td></td>
</tr>
<tr>
<td>Spectrum 2</td>
<td>Yes</td>
<td>3.94</td>
<td></td>
<td>96.06</td>
<td></td>
<td>100.00</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Spectrum 3</td>
<td>Yes</td>
<td>2.04</td>
<td>1.51</td>
<td>96.45</td>
<td></td>
<td>100.00</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Spectrum 4</td>
<td>Yes</td>
<td>30.86</td>
<td>0.93</td>
<td>68.21</td>
<td></td>
<td>100.00</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 9. Polished section of composite coating Cr-Al2O3 by piston ring producer Goetz [21].
Figure 9 shows the enlarged areas selected for the analysis of the GGV ring. The data presented in Table 7 indicate that there is only one chromium layer as the ring’s plating (96.4% and 98.9%), whereas the base of this ring is also essentially made of iron (96.4% and 81.2%). Therefore, this ring can be called chromium-plated. The small particles are alumina particles ($O_2=1.95$ Table 7). Between the galvanic deposited Cr-layer and the cast iron surface is a Cu interlayer.

<table>
<thead>
<tr>
<th>Spectrum</th>
<th>In stats.</th>
<th>C</th>
<th>O</th>
<th>Si</th>
<th>P</th>
<th>Cr</th>
<th>Fe</th>
<th>Mo</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spectrum 1</td>
<td>Yes</td>
<td>1.67</td>
<td>1.95</td>
<td></td>
<td></td>
<td>96.38</td>
<td></td>
<td></td>
<td>100.00</td>
</tr>
<tr>
<td>Spectrum 2</td>
<td>Yes</td>
<td>1.12</td>
<td></td>
<td></td>
<td></td>
<td>98.88</td>
<td></td>
<td></td>
<td>100.00</td>
</tr>
<tr>
<td>Spectrum 3</td>
<td>Yes</td>
<td>1.87</td>
<td></td>
<td>1.71</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>96.42</td>
</tr>
<tr>
<td>Spectrum 4</td>
<td>Yes</td>
<td>82.24</td>
<td>8.74</td>
<td>9.02</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>100.00</td>
</tr>
<tr>
<td>Spectrum 5</td>
<td>Yes</td>
<td>3.72</td>
<td>1.49</td>
<td>1.12</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>81.22</td>
</tr>
</tbody>
</table>

3.2 Analysis of the thickness of the piston ring layer

This segment of the research focuses on the layers of the GGV and G17 piston rings. When observing the plating of the top ring GGV, it can be noticed that this ring has a thick layer of chromium amounting to 400 µm, while the micro-structure of the base material features simple graphites (Figure 10).

In contrast to the GGV ring, the G17 Top ring has a completely different structure of the surface layer. In Figures 11 and 12, two different platings can be noticed, having the maximum thickness of 95.6 µm and 110 µm respectively. The prominent feature of this ring is the overall thickness of its layer (205 µm, both platings). It can be noted that this value is double in the chromium-plated ring.
3.3 Analysis of the micro-hardness of the piston ring base and layers

Measurements of the hardness of both specimens were performed with the aid of Zwick 3212 tester for Vickers hardness (HV0.2). As the sample of the Top ring G17 has rough surface, micro-hardness of the transversal section of coating (Figure 13) could not be measured. Attempts were made by applying a lower load of a HV 0.2, i.e. 1.961 N and a higher load of HV 5, i.e. 49.03 N in order to obtain the pyramid indentation, but the indentation was not visible through the electronic microscope due to the specimen’s rough surface. Glazing the surface would wear out the coating, so that the measurement would not be satisfactory.

The points of the micro-hardness measurement are shown in Figure 13. The procedure established that the Top ring G17 had a greater average hardness of the base material, amounting to 306.5 (HV 0.2). However, its hardness at the cross section coating is only 385.5 (HV 0.2), which indicates that the surface layer of the ring is just a bit harder than its base material.

On the other hand, the chromium-plated Top ring GGV has 217% greater hardness of the coating, amounting to 838.17 (HV 0.2), as presented in Table 8. This is essential in terms of abrasive wearing because the surface layer (coating) makes a physical contact with the cylinder liner, lube oil film and catalyst fines.
Traditionally, it is considered that wear resistance of the coatings just depends on the hardness but nevertheless, elastic modulus also plays a vital contribution in the wear characteristics. [28]. The modulus of elasticity of the G17 piston ring is 110-140 GPa [29], whereas the modulus of elasticity at the complex chromium alloyed grey iron is 154-196 GPa according to DIN 1695 [27].

4. Suggested procedures for further research and improvement of quality of the piston rings on the market

For the purpose of further analysis, it would be beneficial to compare the measured values of the worn-out piston rings with the equivalent data obtained from the rings that are not worn-out, prior to examining the cylinder liner. When taking samples from the cylinder liner, the latter becomes damaged and therefore useless. As there is typically only one spare liner on board, sampling becomes difficult. Furthermore, performing a complex simulation of exposing both types of piston rings to abrasion would be helpful in future analyses, providing that the prevailing operating conditions in the main engine can be simulated. One of the specific features of such a simulation would be the use of both types of fuel, i.e. marine diesel oil (MDO) and heavy fuel oil (HFO), as the durability of the piston rings is considerably affected by the quality of used fuel. The expected durability of the piston ring corresponds to the time between overhaul and is much longer when the engine runs on cleaner MDO, as shown in Table 9.

<table>
<thead>
<tr>
<th>Engine parts</th>
<th>Time between overhaul (h)</th>
<th>Expected durability (h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quality of fuel</td>
<td>HFO</td>
<td>MDO</td>
</tr>
<tr>
<td>- piston rings</td>
<td>12-20,000</td>
<td>20-24,000</td>
</tr>
<tr>
<td>- cylinder liner</td>
<td>12-20,000</td>
<td>20-24,000</td>
</tr>
</tbody>
</table>
Apart from the different quality of fuel used by the engine in operation, the simulation of abrasion effects will also depend on the use of adequate lubricating oils having different recommended base number (TBN). Simulation and accurate recommendations referring to the above parameters are essential as today’s diagnostic parameters indicate problems only when the piston ring wear is greater than 75%.

Companies should introduce the “cat fine risk surveys”, i.e. reports including the state of the engine fuel system components, analysis of the specific features of the marine power system, classification of risks based on the long-term experience of the crew members and superintendents, and recommendations on optimal protection of the engine against damage caused by catalyst fines. This sort of report would be also suitable for education and training of the engineer and their crew who would be able to take better care of the engine. Besides, if damages caused by catalyst fines do occur, the well-trained engineer will be of vital importance for gathering evidence and producing damage reports that can be subsequently used in the process of damage compensation or possible international arbitration. A continued understanding of the piston-cylinder-contact assembly only helps engineers, scientists and any other stakeholder to improve on the piston ring and cylinder liner interaction [31].

One of the critical measures for improving the piston ring quality on the market should be the marking of rings in line with the Technical Code requirements, in order to enable the control quality and reduce the possibility of using poor quality or poorly reconditioned rings [30]. It is necessary to enforce standardisation that would bind all ring manufacturers in terms of the quality of material and production, but also in terms of providing the specification of materials the rings are made of.

5. Conclusion

Catalyst fines are always found in heavy fuel oil. Their presence has to be carefully monitored (used Veritas Petroleum Services BV - VPS laboratory for fuel testing) and their amount should be reduced to a minimum. After settling in fuel tanks, separation and filtering, catalytic fines enter the combustion chamber through the fuel injector. Being highly abrasive, they can create huge problems, damage to the main engine and, consequently, high direct and indirect costs.

This research focused on the analysis of the running in coated (RC) and chromium coated (CC) top compression rings in Wärtsilä RTA two-stroke slow-speed diesel engine on an Aframax tanker. Examination and measurement carried out on board enabled the insights into the state of the piston rings, while the fuel analysis revealed the amount of impurities. During the tests the RC piston rings were damaged or broken, while the wear of the cylinder liner was faster. The analyses of chemical composition established that the grey iron CC ring had a coating of chromium whose thickness amounted to around 400µm, whereas its base material contained more carbon and silicon than the RC piston ring. The latter had a coating consisting of two layers containing nickel and molybdenum, having the overall thickness of 200µm. In addition, the modulus of elasticity of the chromium-plated grey iron ring was higher than in the RC ring. Micro-hardness measurement revealed that the CC ring had a surface layer that was 217% harder than the surface layer of the RC ring.

The results obtained by the analyses and measurements indicated that the chromium-plated piston ring had higher hardness and was more efficient in withstanding abrasive effects of catalytic fines, while the higher modulus of elasticity of its base material enabled a better resistance to breaking.
Acknowledgement

All laboratory measurements in this research were performed at the Faculty of Mechanical Engineering of the University of Maribor (Fakulteta za strojništvo Univerze v Mariboru) through the bilateral project **BI-ME/18-20-024 (2019-2020)**. Practical testing of the samples required for the analysis was performed on board an Aframax tanker. We would like to thank our Slovenian colleagues for the use of laboratory facilities and consultations.

REFERENCES


[16] Lalić, B., Komar, I., Dobrota, D.: *Structural modifications for improving the tribological properties of the cylinder unit in two-stroke slow speed marine diesel engines*, Transactions on Maritime Science (ToMS), Vol. 01, No. 02, 2012. [https://doi.org/10.7225/toms.v01.n02.004](https://doi.org/10.7225/toms.v01.n02.004)


Piston Ring Material in a Two-Stroke Engine Which Sustains Wear Due to Catalytic Impurities

Miroslav Vukićević, Nikola Račić, Špiro Ivošević


[21] Results of the measurements performed at the University of Maribor, the Faculty of Mechanical Engineering (Fakulteta za strojništvo Univerze v Mariboru), through the bilateral project BI-ME/18-20-024 (2019-2020).


Submitted: 21.03.2019. Miroslav Vukicevic, miroslav.v@ucg.ac.me
vukicevic.miroslav@gmail.com

Accepted: 13.05.2019. University of Montenegro, Faculty of Maritime Studies, Kotor, Montenegro
Nikola Racic, University of Split, Faculty of Maritime Studies, Split, Croatia
Spyro Ivošević, University of Montenegro, Faculty of Maritime Studies, Kotor, Montenegro

169
A LID APPROACH FOR PREDICTING WAVE INDUCED MOTIONS OF TRIMARAN IN REGULAR WAVES

UDC 629.5.022.3:629.5.017
Professional paper

Summary

The wave induced motions of a trimaran sailing in regular head waves were predicted by using the three dimensional boundary integral method. Large wave elevation on the free surface in between the trimaran’s main-hull and the outriggers occurred at some specific frequencies in the numerical model. The large wave elevation also induced large heave and pitch motions of trimaran. However, the large wave elevation and corresponding large motions were not observed during towing tank tests. A lid approach was introduced in this paper by placing lid on the free surface in between hulls to suppress the unrealistically large wave elevation and to correctly predict the induced motions of trimaran. The feasibility and practicability of lid approach were validated against experimental results.

Key words: Trimaran; lid approach; ship motions; wave elevation; regular waves

1. Introduction

Naval architects have been pursuing one permanent goal – achieving maximum speed with minimum energy. Many distinct hull forms have been created and innovated over several decades. Among them is the trimaran hull, which held the attention of ship designers and researchers immediately after the concept was proposed. The main-hull of a trimaran could be constructed much slenderer than a regular mono-hull vessel to reduce drag. Meanwhile the trimaran could exhibit excellent seakeeping performance with outriggers. RV Triton, launched in May 2000 and delivered in August 2000, is the world’s first large motor powered trimaran, with a displacement of 1200 t, a length of 90 m and a beam of 22.5 m [1]. The successful trials program proved that the design was able to satisfy any operational requirement in the exactly same way as a mono-hull vessel. Significant drag reduction leading to high-speed has been well demonstrated through this program. Hitherto, multiple types of trimaran have been enlisted in various fields, like Benchijigua Express and Independence class LCS.

A number of research papers focusing on seakeeping performance of trimaran have been published by various researchers. Bingham et al. [2] computed motions (heave and pitch) and loads of a trimaran in regular waves using frequency-domain boundary integral method. They used two methods, pulsating and translating source and purely pulsating source distributions on
mean wetted surface. The results obtained from the two approaches show that the latter is surprisingly more suitable for fast vessels. Model test is an efficient method to research wave induced motions [3] and consequently many researchers employ this method to study the motions of trimaran. Hebblewhite, Sahoo and Doctors [4] investigated the effects of longitudinal stagger of outriggers on heave and pitch motions of a typical trimaran hull. A round-bilge high-speed hull form was constructed and four different longitudinal stagger positions were tested in the experiments. Hebblewhite, Sahoo and Doctors [4] also provided numerical results obtained from code HYDROS. The paper stated that the variation in the radius of gyration could have a significant effect on the heave and pitch motions. McDonald et al. [5] compared trimaran small waterplane area center hull, monohull and trimaran hullforms from the angle of design. Pavkov and Morabito [6] conducted a series of tests of two different trimaran models to determine the effect of finite water depth on the resistance, heave and trim.

In this paper, the three-dimensional boundary integral method (BIM) has been utilized to analyze wave induced motions of a trimaran hull in regular head waves. In the numerical model, large wave amplitude of the free-surface in between the main-hull and outriggers were observed in some specific narrow frequency bands. The large wave elevation also resulted in unrealistically large hull motion in heave and pitch. However, this phenomenon of large wave elevation and large hull motion at specific frequencies were not observed during towing tank experiments. The unrealistic phenomenon has also been reported in literatures with regard to numerical simulations of multiple floating bodies and catamarans ([7], [8]). Researchers also provided some practical methods to solve this issue, such as directly introducing artificial viscosity to the free surface [8] and adding artificial damping with a rigid lid [9] or with a flexible lid [10]. However, most researches focused on the multiple floating structures with zero velocity. This was different from the situation happening to the trimaran because the large wave elevation only happened when the velocity was larger than zero according to our research.

This paper introduces the lid approach to the prediction of wave induced motions of trimaran. A rigid lid with artificial damping was laid on the free surface in between the main-hull and outriggers to suppress the unrealistic wave elevation. The results showed that the lid approach successfully eliminated the negative effect of large wave elevation around the trimaran. The lid approach removed the peak of Response Amplitude Operator (RAO) of hull motion at frequencies where the large wave elevation occurred. Furthermore, this approach hardly influenced the RAO value at another frequencies. To validate the lid approach, a series of tests were also carried out in the towing tank of Dalian University of Technology. The agreements of numerical prediction and experimental data indicated that the lid approach could be a satisfying practical solution.

2. The trimaran model

The trimaran model was composed of one central main-hull and two outriggers standing respectively on each side of the main-hull. All these three hulls were rigidly connected together as an entire floating structure with two wooden beams. As a consequence, the trimaran model exactly held six degrees of freedom though it was composed of three separate hulls. The main-hull was of hard chine type, with a transom stem designed as a high-speed displacement hull form. The outriggers, which were approximately a quarter in length of the main-hull, had similar hull lines to those of main-hull. Nevertheless, the main dimensions of outrigger were not on the same scale to those of main-hull. The hull lines and the main dimensions of the trimaran are shown in Fig. 1 and Table 1, respectively.
Table 1 Main dimension

<table>
<thead>
<tr>
<th>Item</th>
<th>Main-hull’s</th>
<th>Outrigger’s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Waterline length (m)</td>
<td>4.0</td>
<td>1.0</td>
</tr>
<tr>
<td>Waterline beam (m)</td>
<td>0.3584</td>
<td>0.085</td>
</tr>
<tr>
<td>Draft (m)</td>
<td>0.17</td>
<td>0.10</td>
</tr>
<tr>
<td>Displacement mass (kg)</td>
<td>129.07</td>
<td>4.45</td>
</tr>
<tr>
<td>Wetted surface area (m²)</td>
<td>1.899</td>
<td>0.201</td>
</tr>
<tr>
<td>Block coefficient C_B</td>
<td>0.530</td>
<td>0.524</td>
</tr>
<tr>
<td>Prismatic coefficient C_p</td>
<td>0.769</td>
<td>0.746</td>
</tr>
</tbody>
</table>

Fig. 1 Trimaran bodyline

Different schemes were designed in accordance with changes of longitudinal and/or transverse positions of the outriggers. The positions of the outriggers and the coordinate system are displayed in Fig. 2. Here, $O$, intersection of central longitudinal section and midship section on waterplane, represents original point; $a$ represents the distance from midship section of the main-hull to that of the outriggers, positive towards the bow; $b$ denotes the distance from central longitudinal section of main-hull to that of outriggers. As shown in Table 2, four schemes are used as sample to illustrate the phenomenon of unrealistic wave elevation: Scheme 4-13, Scheme 4-07, Scheme 5-13 and Scheme 5-07. Scheme 4-13 and Scheme 4-07 are selected to validate the lid approach’s feasibility.

Table 2 Location of outrigger

<table>
<thead>
<tr>
<th>Scheme</th>
<th>$a$ (m)</th>
<th>$b$ (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4-13</td>
<td>-1.3</td>
<td>0.4</td>
</tr>
<tr>
<td>4-07</td>
<td>-0.7</td>
<td>0.4</td>
</tr>
<tr>
<td>5-13</td>
<td>-1.3</td>
<td>0.5</td>
</tr>
<tr>
<td>5-07</td>
<td>-0.7</td>
<td>0.5</td>
</tr>
</tbody>
</table>
All calculations were carried out in deep water condition in head seas with three speeds: 1.464 m/s, 3.091 m/s and 4.392 m/s. The corresponding Froude Numbers are 0.23, 0.49, and 0.70. The wave frequency ranges from 0.4 rad/s to 6 rad/s.

### Table 3 Wave lengths and corresponding wave frequencies

<table>
<thead>
<tr>
<th>Item</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wave length (m)</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>11</td>
</tr>
<tr>
<td></td>
<td>9</td>
</tr>
<tr>
<td></td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>7</td>
</tr>
<tr>
<td></td>
<td>6</td>
</tr>
<tr>
<td></td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>2</td>
</tr>
<tr>
<td>Wave frequencies (rad/s)</td>
<td>2.3</td>
</tr>
<tr>
<td></td>
<td>2.4</td>
</tr>
<tr>
<td></td>
<td>2.6</td>
</tr>
<tr>
<td></td>
<td>2.8</td>
</tr>
<tr>
<td></td>
<td>3.0</td>
</tr>
<tr>
<td></td>
<td>3.2</td>
</tr>
<tr>
<td></td>
<td>3.5</td>
</tr>
<tr>
<td></td>
<td>3.9</td>
</tr>
<tr>
<td></td>
<td>4.5</td>
</tr>
<tr>
<td></td>
<td>5.5</td>
</tr>
</tbody>
</table>

The research also applied towing tank test results to validate the computational results. The wave direction and travelling speeds were the same with numerical prediction but the tests were carried out with respect to wave length. In order to conveniently compare, the wave lengths employed in the tests are translated into circular frequencies, which are presented in Table 3.

### 3. Computational methodology and numerical model

The Cartesian coordinate system $X = (x, y, z)$ is utilized by considering the plane $z = 0$ the undisturbed free surface of water and the space $z < 0$ the fluid domain. The fluid is assumed to be incompressible, inviscid, and the flow is irrotational. As a result, the velocity vector $V$ can be represented by the gradient of the potential $\phi$

$$V = \nabla \phi$$

Therefore, the governing partial differential equation can be described

$$\frac{\partial^2 \phi}{\partial x^2} + \frac{\partial^2 \phi}{\partial y^2} + \frac{\partial^2 \phi}{\partial z^2} = 0$$

A trimaran exhibits 6 degrees of freedom so it could be assumed as a rigid body by means of a mass-spring system. As a consequence, a trimaran’s motions induced by waves in the frequency domain can be described with the following equation:

$$\sum_{j=1}^{6} (M_{kj} + a_{kj})\ddot{x}_j + b_{kj}\dot{x}_j + c_{kj}x_j = F_k \quad k = 1, 2, \ldots, 6$$

where:

- $k, j =$ suffixes describing hydrodynamic property in $k$ mode as a result of motion in $j$ mode
- $M_{kj} =$ mass of trimaran

$$\ddot{x}_j + \frac{b_{kj}}{M_{kj}}\dot{x}_j + \frac{c_{kj}}{M_{kj}}x_j = \frac{F_k}{M_{kj}}$$

$$\frac{\partial^2 \phi}{\partial x^2} + \frac{\partial^2 \phi}{\partial y^2} + \frac{\partial^2 \phi}{\partial z^2} = 0$$

$$V = \nabla \phi$$

Therefore, the governing partial differential equation can be described

$$\frac{\partial^2 \phi}{\partial x^2} + \frac{\partial^2 \phi}{\partial y^2} + \frac{\partial^2 \phi}{\partial z^2} = 0$$

A trimaran exhibits 6 degrees of freedom so it could be assumed as a rigid body by means of a mass-spring system. As a consequence, a trimaran’s motions induced by waves in the frequency domain can be described with the following equation:

$$\sum_{j=1}^{6} (M_{kj} + a_{kj})\ddot{x}_j + b_{kj}\dot{x}_j + c_{kj}x_j = F_k \quad k = 1, 2, \ldots, 6$$
A lid approach for predicting wave induced motions of trimaran in regular waves

Zongyu Jiang, Yun Gao, Jie Liu

\[ a_{k,j} = \text{added mass matrix} \]
\[ b_{k,j} = \text{damping coefficient matrix} \]
\[ c_{k,j} = \text{hydrostatic restoring matrix} \]
\[ F_k = \text{external force in mode } k \]

According to the potential flow theory, the added mass and damping coefficient can be described

\[ a_{k,j} = \text{Re}[\rho \int_S \phi_j n_k dS] \quad (4) \]
\[ b_{k,j} = \text{Im}[\rho \int_S \phi_j n_k dS] \omega \quad (5) \]

where
\[ S = \text{body surface} \]
\[ \rho = \text{density of water} \]
\[ \omega = \text{wave frequency} \]

The boundary integral method (BIM) was employed to predict wave induced motions of the trimaran. The fundamentals of this method lie in the Green’s theorem, in which velocity potential at any point of the fluid domain is represented by distributions of singularities on boundary surfaces ([11], [12], [13]). Consequently, the wetted surface of body should be represented by a number of panels. Figure 3 shows the distribution of panels of Scheme 4-07 as an example. The mesh convergence tests were performed to establish a proper mesh for the calculation by using the Scheme 4-13. During the tests, the body surface was respectively represented with 229, 980, 1738 and 4848 meshes. The results, as shown in Fig. 4, indicated that the calculation converged. When the body surface was built with 1738 meshes, the calculation reaches a fairly good precision. Thus, the model with 1738 meshes was employed for the calculation.
The boundary-integral equations of wave problems suffer an irregular frequency issue. The irregular frequencies correspond to the eigen frequencies of the homogeneous Dirichlet problem arising from the existence of fluid and free surface inside the body [14]. Consequently, numerical solutions of the integral equations in the vicinity of these irregular frequencies are incorrect. In order to eliminate irregular frequencies, the technique proposed by Ohmatsu [15] is employed. By imposing an extra boundary condition on the interior of the free surface, the Green’s integral equation is extended to the interior of the free surface ([16], [17], [18]). The encounter frequency method was used to perform the calculation. This method is widely used for calculating hydrodynamics of body with forward speed and it can provide reasonable results ([19], [20]). The relationship between the encounter frequency and wave frequency follows

$$\omega_e = \omega - U k \cos \theta$$

where
- $\omega_e$ = encounter frequency
- $\omega$ = wave frequency
- $U$ = ship velocity
- $k$ = wave number
- $\theta$ = the direction of wave propagation relative to the x-axis
4. The phenomenon of unrealistic wave elevation and lid approach

4.1 The unrealistic wave elevation

In the process of calculating the trimaran’s motions in waves, a very interesting phenomenon was discovered, which was somewhat like the irregular frequency issue, even though extension for source distribution in the interior of the free surface of the hull is imposed to eliminate irregular frequencies. This phenomenon occurred at some specific frequencies and there also existed abnormal resonance peak in the computed RAO curve in the vicinity. It only happened in numerical simulation and had never been observed during tests. One numerical example and its comparison to the experimental result are presented in Fig. 5 where the RAOs are plotted with line representing numerical data and dark points representing experimental data. It was apparent that the computed result deviate significantly from the experimental data at nondimensional frequency 2.4 (wave frequency 3.8 rad/s) and its neighbourhood. Further investigation revealed that, at the specific frequencies, wave amplitudes of the free surface in between the main-hull and the outriggers were unrealistically large. The abnormal elevation of the free surface in between the main-hull and the outrigger is shown in Fig. 6. The peak locates at nondimensional frequency 2.4 as well. The added mass and radiation damping coefficient are shown in Fig. 7. There existed a dent on the curve of added mass and the curve of radiation damping precipitates in the vicinity of nondimensional frequency 2.4. According to equation (3), the small value of added mass and radiation damping could lead to large motions. Therefore, the large heave motion of hull was induced by the unrealistic large wave elevation in the gap between hulls.

Fig. 6 RAO of free surface elevation in between hulls, 4-13, Fn=0.49 (x=-0.9 m, y=0.27 m)
The occurring frequency of large wave elevation was connected to forward speed and configuration of gaps in between the main-hull and the outriggers. Firstly, the phenomenon occurred at a certain encounter frequency when the configuration of gap was fixed. Examples of occurring frequencies and their corresponding encounter frequencies are shown in Table 4. Secondly, the encounter frequency of resonance was inversely proportional to the gap width. The encounter frequency of Scheme 4-07 and Scheme 4-13 is approximately 8.2 rad/s and the encounter frequency of Scheme 5-07 and Scheme 5-13 was approximately 7.8 rad/s.

Table 4 Occurring frequencies and corresponding encounter frequencies

<table>
<thead>
<tr>
<th>Scheme</th>
<th>Froude number</th>
<th>Resonance freq. (rad/s)</th>
<th>Encounter freq. (rad/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4-13</td>
<td>0.23</td>
<td>4.8</td>
<td>8.2</td>
</tr>
<tr>
<td></td>
<td>0.49</td>
<td>3.8</td>
<td>8.4</td>
</tr>
<tr>
<td></td>
<td>0.7</td>
<td>3.3</td>
<td>8.2</td>
</tr>
<tr>
<td></td>
<td>0.23</td>
<td>4.7</td>
<td>8.1</td>
</tr>
<tr>
<td>4-07</td>
<td>0.49</td>
<td>3.7</td>
<td>8.2</td>
</tr>
<tr>
<td></td>
<td>0.7</td>
<td>3.3</td>
<td>8.2</td>
</tr>
<tr>
<td></td>
<td>0.23</td>
<td>4.6</td>
<td>7.8</td>
</tr>
<tr>
<td>5-13</td>
<td>0.49</td>
<td>3.6</td>
<td>7.7</td>
</tr>
<tr>
<td></td>
<td>0.7</td>
<td>3.2</td>
<td>7.8</td>
</tr>
<tr>
<td></td>
<td>0.23</td>
<td>4.6</td>
<td>7.8</td>
</tr>
<tr>
<td>5-07</td>
<td>0.49</td>
<td>3.6</td>
<td>7.7</td>
</tr>
<tr>
<td></td>
<td>0.7</td>
<td>3.2</td>
<td>7.8</td>
</tr>
</tbody>
</table>

Molin [21] investigated the piston mode motion of water occurs in the moonpool. According to his theory, the natural frequency of piston mode motion can be described

$$\omega_n = \sqrt{\frac{g}{h(1+C)}}$$

(7)

with

$$C = \frac{1}{2\pi \ b l h} \left[ b^2 l \arg \sinh \frac{1}{b} + b l^2 \arg \sinh \frac{b}{l} + \frac{1}{3} (b^4 + l^4) - \frac{1}{3} (b^2 + l^2)^{3/2} \right]$$

(8)
A lid approach for predicting wave induced motions of trimaran in regular waves

Zongyu Jiang, Yun Gao, Jie Liu

where

\( g \) = gravity acceleration
\( h \) = draft of moonpool
\( b \) = width of moonpool
\( l \) = length of moonpool

While considering the occurring frequency as the natural frequency of piston mode and using this equation to calculate the occurring frequency, \( h \) was the draft of hard chine of outrigger, \( b \) was the gap width at the midship section of outrigger and \( l \) was equal to the length of outrigger. Consequently, the gap configuration of Scheme 4-13 was similar to that of Scheme 4-07 and the condition between Scheme 5-13 and 5-07 was alike. According to Molin’s equation, the resonant frequency of Scheme 4-13 and 4-07 is 8.2 rad/s and the resonant frequency of Scheme 5-13 and 5-07 is 7.4 rad/s. These frequencies slightly deviated from those listed in Table 4. Molin’s theory is based on an assumption that the moonpool is rectangular. However, the gap between hulls is not strictly rectangular because of the hull line. As a result, the natural frequency derived from Molin’s equation slightly deviated from the numerical calculation.

4.2 The lid approach

In order to suppress the large wave elevation, a lid was utilized to replace the two physical free surfaces in between the main-hull and the outriggers. One exemplary outline of the lid approach was given in Fig. 8. Only a replacement of sufficient part of resonant free surface was capable to achieve satisfactory results so the lid was arbitrarily rectangular. This lid was independent of the trimaran so the lid and trimaran composed a multi-body system which involves twelve degrees of freedom due to rigid-body assumption. The motion equation for multi-body system in regular waves ([22], [23]) can be described as:

\[
\sum_{\alpha=1}^{M} \sum_{\beta=1}^{n} \left( \delta_{\alpha \beta} M \dot{x}_{\alpha} + a_{\alpha \beta} x_{\alpha} \right) \ddot{x}_{\beta} + b_{\alpha \beta} \dot{x}_{\beta} + \delta_{\alpha \beta} c_{\alpha \beta} \dot{x}_{\beta} = F_{k} \quad n = 1, 2, \ldots, M \quad k = 1, 2, \ldots, 6
\]

where:

\( M \) = the number of bodies, obviously, \( M = 2 \) in this case
\( \delta_{\alpha \beta} \) = the Kroneker delta function

If the lid was free to move without any external restraints, it could not provide any assistance to reduce the large wave elevation. Thereby, a fictitious linear damping coefficient \( \mu_{p} \) was artificially added on the lid to suppress its motion induced by waves. An appropriate value of linear damping coefficient could be evaluated from experimental data. Tan et al. [24] also provided an equation to estimate the linear damping coefficient
\[ \mu_p = \zeta \frac{\omega_n^2 \eta_A}{2g} \]  

where:
\( \zeta \) = the non-dimensional energy loss coefficient
\( \eta_A \) = the response amplitude of wave elevation
\( \omega_n \) = the occurring frequency of large wave elevation

By using the lid approach, the impact of large elevation in the gap could be successfully suppressed, as shown in Fig. 9, where the prediction agreed with the experiment. Figure 10 presents the added mass and radiation damping coefficient calculated by the model with and without lid. The two added mass curves matched well except in the resonant region, where the added mass calculated from the model with lid was greater and the dent was filled. Furthermore, the radiation damping was approximately three times larger than that from the model without lid at the occurring frequency. The larger added mass and radiation damping could be a factor that reduces the heave motion at the occurring frequency. It should be noted that the radiation damping from lid model was apparently smaller before the occurring frequency. That might be caused by the lid because the lid suppressed the waves induced by the motions of hull which was directly related to the radiation damping.

![Fig. 9 Heave RAO deriving from the lid approach, 4-13, Fn=0.49](image)

![Fig. 10 Added mass and radiation damping for heave deriving from the lid approach, 4-13, Fn=0.49](image)
4.3 The validation of lid approach

A series of calculations for Scheme 4-07 and Scheme 4-13 were carried out and they were compared with the experimental data to verify the feasibility of the lid approach.

The RAOs of heave and pitch are nondimensionalized with respect to wave amplitude and wave number

Heave RAO: \[ H_z(\omega) = \frac{\zeta_a}{\tilde{z}_a} \] (11)

Pitch RAO: \[ H_\theta(\omega) = \frac{\theta_a}{k\tilde{\zeta}_a} \] (12)

where:
\[ \omega = \text{circular frequency of incoming wave in } \text{rad/s} \]
\[ \zeta_a = \text{amplitude of heave motion in } m \]
\[ \theta_a = \text{amplitude of pitch motion in } \text{rad} \]
\[ \tilde{\zeta}_a = \text{amplitude of incoming wave in } m \]
\[ k = \text{wave number in } m^{-1} \]

Figure 11 and 12 illustrate the heave and pitch RAOs of Scheme 4-07 and 4-13 obtained from calculation and experiment, respectively. As shown in Fig. 11 and 12, most RAO curves calculated by the model without lid had two peaks. One reflected the trait of wave induced motions of the hull, which agreed with the tests. The other was induced by the unrealistic wave elevation in the gap, which deviated from the tests. In this peak region, the value of RAO obtained from the model without lid was irrationally larger than those obtained from the tests while the RAO obtained from the model with lid satisfactorily agreed with the RAO from tests. This demonstrated that the lid approach was able to practically deal with the issue with successful suppression of unrealistic wave elevation in between the main-hull and the outriggers.
Fig. 11 Heave RAO and pitch RAO of Scheme 4-07. Dot: test; Solid line: numerical model without lid; Dashed line: numerical model with lid.
A lid approach for predicting wave induced motions of trimaran in regular waves

Zongyu Jiang, Yun Gao, Jie Liu

Fig. 12 Heave RAO and pitch RAO of Scheme 4-13. Dot: test; Solid line: numerical model without lid; Dashed line: numerical model with lid.

5. Conclusion

The wave induced motions of the trimaran were predicted by using the boundary integral method (BIM). The predicted wave elevation in between the trimaran’s main-hull and outriggers were unrealistically large. However, the large wave elevation was not discovered during tests. The large wave elevation could induce large heave and pitch motions of the trimaran, which were also larger than the experimental results. The study revealed the equation raised by Molin could be used to predict the occurring frequency of the large wave elevation.

A practical approach was employed to suppress the large wave elevation by laying a lid on the free surface in between hulls. By using the lid approach, the large wave elevation was successfully suppressed. Meanwhile, this approach also suppressed the large motions induced
by the large wave elevation. According to comparison with the test results, it was verified that the lid approach was able to practically predict motions of the trimaran sailing in regular waves.

REFERENCES


A lid approach for predicting wave induced motions of trimaran in regular waves

Zongyu Jiang, Yun Gao, Jie Liu


Accepted: 16.05.2019.

Zongyu Jiang 1,*, jzongyu@hotmail.com
Yun Gao 2,3, dutgaoyun@163.com
Jie Liu 4, Lucy.Liu@mathworks.cn
1 Wood Group, Shanghai 201206, China
2 State Key Laboratory of Oil and Gas Reservoir Geology and Exploration, Southwest Petroleum University, Chengdu 610500, China
3 Department of Mechanical Engineering, The University of Tokyo, Tokyo 113-8656, Japan
4 MathWorks, Shanghai 200122, China
* Corresponding author (Zongyu Jiang): jzongyu@hotmail.com
DETERMINATION OF AN OPTIMAL FLEET FOR A CNG TRANSPORTATION SCENARIO IN THE MEDITERRANEAN SEA (str.1-23)
Francesco Mauro, Luca Braidotti, Giorgio Trincas
Original scientific paper

EFFECT OF BREAKING WAVE SHAPE ON IMPACT LOAD ON A MONOPILE STRUCTURE (str.25-42)
Duje Veić, Wojciech Sulisz, Rohan Soman
Original scientific paper

SIMPLIFIED AND ADVANCED APPROACHES FOR EVACUATION ANALYSIS OF PASSENGER SHIPS IN THE EARLY STAGE OF DESIGN (str.43-59)
Carlo Nasso, Serena Bertagna, Francesco Mauro, Alberto Marinò, Vittorio Bucci
Professional paper

ENVIRONMENTAL AND COST-EFFECTIVENESS COMPARISON OF DUAL FUEL PROPULSION OPTIONS FOR EMISSIONS REDUCTION ONBOARD LNG CARRIERS (str.61-77)
Nader R. Ammar
Review article

EVALUATION MODEL OF MARINE POLLUTION BY WASTEWATER FROM CRUISE SHIPS (str.79-92)
Tina Perić, Vice Mihanović, Nikola Račić
Original scientific paper

STUDY OF CONTINUOUS ICEBREAKING PROCESS WITH COHESIVE ELEMENT METHOD (str.93-114)
Feng Wang, Li Zhou, Zao-Jian Zou, Ming Song, Yang Wang, Yi Liu
Original scientific paper

VEHICLE SECURING SAFETY ASSESSMENTS OF A KOREAN COASTAL CAR FERRY ACCORDING TO ACCELERATION PREDICTION APPROACHES (str.115-131)
Joonmo Choung, Se-Min Jeong
Professional paper

VALIDATION OF OPTIMALLY DESIGNED STATOR-PROPELLER SYSTEM BY EFD AND CFD (str.133-151)
Yong Jin Shin, Moon Chan Kim, Jin Gu Kang, Hyeon Ung Kim, I Rok Shin
Original scientific paper

A DECISION-SUPPORT TOOL FOR DEMOLITION SALE OF A VESSEL (str.153-173)
Basak Akdemir, Ahmet Beskese
Review article
DETERMINATION OF AN OPTIMAL FLEET FOR A CNG TRANSPORTATION SCENARIO IN THE MEDITERRANEAN SEA

UDC 629.5(05): 629.56: 629.5.545: 629.5.01
Original scientific paper

Summary

For the natural gas transportation, several technologies can be applied, having different effectiveness and costs depending on the analysed case. The Mediterranean Sea is presenting a typical scenario where compressed natural gas (CNG) transportation is particularly attractive compared to liquefied natural gas (LNG) and pipelines, not only for stranded gas shipping but also for transportation cases where CNG is usually representing the most economically convenient solution. Approaching the design of a CNG ship is not an easy task, since the pressure vessel (PV) technology is strongly influencing the ship layout and hull form. Here an enhanced conceptual design method is adopted; taking into account the economic-financial issues together with logistics, in order to determine the best fleet composition selecting the best ships for the selected scenario. The ships composing the fleet are supposed to load/offload the natural gas on buoys; hence, dynamic positioning (DP) will also be considered as an attribute in the evaluation of alternative designs. As final outcome of the enhanced concept design process it will be possible to speed up drawing of the preliminary lines plan and general arrangement plan of the sister ships composing the fleet.

Key words: CNG transportation; Conceptual design; Optimal fleet composition; Dynamic positioning; Shipping tariff

1. Introduction

Nowadays hydrocarbons are still the main source for energy production around the world, covering almost the 80% of the total energy production [1]. Natural gas, among all available hydrocarbons, is for sure the most environmental friendly solution for energy production and it is believed that it will be continuously more employed for production with an estimated annual rate of 2.2% from 2015 to 2050. Already in 2025, the natural gas production will grow up to 151 trillion cubic feet (tcf), being almost 70% higher than global production in 2001 [2]. The growing necessity to access the gas resources requires more flexibility for the exploitation of the considered resource [3]. In particular, especially for maritime countries, this flexibility implies to dedicate particular effort also on the gas transportation from off-shore fields to on-shore installations [4]. Several technologies can be adopted for natural gas marine transportation. Present means of transporting natural gas to the markets consists primarily of
pipelines and liquefied natural gas (LNG), the former accounting for 71% of all internationally traded gas volumes with the latter making for the rest.

Pipelines are usually the most efficient way to supply natural gas to a final on-shore user. However, its adoption for offshore purposes is limited by the huge installation costs and durable investments, making the offshore pipelines costs up to ten times higher than on-shore ones. The final costs are then influenced by the distance between the gas source and the end user, the water depth and the seabed orography. On the contrary, whichever marine gas transportation mode avoids these kind of restrictions, removing the physical tie between producer and buyer, resulting in a more flexible solution even for the spot market [5].

However, due to the continuous research on the pressure vessels (PV) materials [6], the compressed natural gas (CNG) transport is starting to be really attractive [7,8] and economically competitive also on medium-long distances and large gas volumes, where, traditionally, LNG has always been the most economical solution. CNG is also attractive because of the absence of costly infrastructure facilities such as liquefaction and regasification plants. In fact, in terms of infrastructures, a CNG supply chain only requires submerged turret loading (STL) systems and single point mooring (SPM) discharging systems, which are less expensive than LNG facilities. For such a reason, being CNG competitive also for stranded gas transportation, it can be stated that, in the near future, CNG technology will be the most flexible solution for natural gas transportation.

Marine CNG might be studied as ideal solution for gas trading in several areas around the globe [9,10], where trade links are all within a transport range from 800 up to 2500 nm, which can be considered as the economic competitive range for fleets composed by CNG ships [11,12]. Then, to efficiently minimise the shipping tariff for the specific trade scenario, the definition and selection of an optimal fleet for CNG transportation becomes of primary importance [13,14,15]. An alternative approach was proposed [16], where the best CNG ships of different size and hull form characteristics can be identified, and then, according to the specific boundaries and constraints, the best solution is identified in terms of economic effectiveness.

Whatever approach is used to handle the CNG marine transportation, the success of the gas value chain is given by the adoption of a feasible and sufficiently accurate concept design process. It is well known that most of the total ship lifetime cost is driven by the decisions taken in the concept design phase [17]. Therefore, putting even more effort in this phase will be extremely favourable for the final economic effectiveness of the gas transportation. For this purpose, an enhanced concept design process [18] is adopted throughout this work, to study the optimal fleet composition for CNG transportation. In the specific case of a CNG ship [19], this kind of approach is mandatory since, due to the continuous developments in the storage technologies and pressure vessels’ materials (see Table 1), there is the necessity to design never-built ships, without having any comparable reference.

By means of an accurate concept design, different concurrent solutions for several subsystems can be easily compared [20], identifying the most competitive one.

Table 1 Pressure vessels types and characteristics

<table>
<thead>
<tr>
<th>PV (250 bar)</th>
<th>D_pv (m)</th>
<th>W_pv (t/m)</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type I</td>
<td>1.00</td>
<td>0.86</td>
<td>Full steel</td>
</tr>
<tr>
<td>Type II</td>
<td>0.95</td>
<td>0.71</td>
<td>Full steel</td>
</tr>
<tr>
<td>Type III</td>
<td>2.35</td>
<td>0.81</td>
<td>Steel liner wrapped with composites</td>
</tr>
<tr>
<td>Type IV</td>
<td>2.34</td>
<td>0.72</td>
<td>Full composites</td>
</tr>
</tbody>
</table>
Here, a possible scenario of natural gas transportation in the Mediterranean Sea is presented, assuming an economic/financial scenario referred to the gas shipping between Zohr field (Egypt) and Brindisi (Southern Italy) with Type III PVs. Since the ships’ loading phase is performed through an STL system, also dynamic positioning (DP) has been considered during the concept design phase, evaluating the station-keeping ability in specific conditions and using it as a constraint in the selection procedure.

2. Design Methodology

With the aim of selecting the “best possible” fleet capable to transport gas in a given operative scenario, the Multiple Attribute Decision Making (MADM) approach has been adopted as concept design methodology of CNG ships. With this approach, a huge number of different concept solutions can be generated and then the “best possible” concept design is selected through a dedicated evaluation process. The outcome of the process has a strong connection to the operative scenario and can be assumed as basis for the next design phases.

In the present study, MADM has been used to build two databases of “best possible” CNG ships characterized by different gas capacities (from 50 to 900 mmsfc, in steps of 50 mmsfc) and different propulsion system. Type III PVs of 2.35 m diameter at 250 bar have been chosen as gas containment system for all the ships. Then, in the so called external task, the best fleet composition is assessed in order to minimise the gas shipping tariff. Concurrently, the corresponding number of sister ships, their capacity and service speed are determined according to logistics constraints. Furthermore, the internal cargo layout is perfected by defining the sectional area curve at both design draught and tank-top height. With this approach, the number and length of pressure vessels are more accurately assessed and a concept lines plan and general arrangement are generated [18] in order to speed up the transition to preliminary design. The whole concept design process is provided in Figure 1.
2.1 Generation of ship database

In the frame of the MADM approach, each design can be represented by a point in the design space spanned by the design variables as well as by a point in the attribute space spanned by the design attributes [21]. The design generation process maps the variables space to the attributes space by means of a Mathematical Design Model (MDM). A number of modules, which are related to functional decomposition of the ship, where detailed requirements and functions are grouped together, composes the MDM. The MDM is driven by an adaptive Monte Carlo sampling which generates a large number of different designs (i.e. sets of variables’ values) within a bounded design space. Through the MDM, the ship properties (attributes) corresponding to each design are assessed from variables and parameters.

Then, the selection process starts. The feasible design sub-space in the attributes space is defined by the application of constraints on design properties. Then, the feasible designs are filtered for dominance, i.e. filtering the Pareto set through metrics of attributes values’ distance from the identified ideal point. The preferred non-dominated solutions are then identified via a fuzzy ELECTRE method [22]. In order to better explore the attribute space, further solutions are generated in mini-cubes around the preferred non-dominated designs through fractional factorial design (FFD). The best possible design identified for a selected capacity is added to the database.

It is worth noticing that, in MADM, the selection of variables, parameters, attributes, and constrains, as well as the definition of design space should be carefully carried out by the design team. In the present study, the design model is structured around the following free variables:

- Length between perpendiculars ($L_{BP}$);
- Beam ($B$);
- Draft ($T$);
- midship section coefficient ($C_M$);
- vertical prismatic coefficient ($C_{VP}$).

These variables are considered sufficient to define the ship accurately and uniquely at the concept design stage, since other hull form characteristics are derived by means of regressions on specific ship database from the above mentioned variables. Furthermore, some technical constants, the so-called parameters, are determined outside the model and consequently remain fixed in running generation of all the alternative designs. They can be classified as shape and size parameters, topological parameters and positioning parameters. As the MDM has to represent the ship in a simple but meaningful manner, a Design of Experiments (DoE) was performed to reduce the number of significant design variables by running different sample runs [13]. In order to restrain the model from shifting to infeasible regions of the design space, design variables are not completely free since, in addition to lower and upper bounds, they may be constrained by dependencies with each other.

Design attributes are principal elements for decision making defining the performances and characteristics of the generated ships. Based on the attained level of the attributes, designs will be accepted or rejected from further analysis. Number of attributes may be as large as needed. Attribute values, which usually have different units of measurement, are normalized via fuzzy sets before entering the dominance analysis. The upper and lower limits of satisfaction for each attribute are provided by the design team. Most of the attributes are calculated via metamodels [23]. Search and optimisation algorithms were used to find the metamodels of ship response functions which are represented by multi-linear regression equations on the predictors. The main metamodels included in the MDM in order to assess attribute values refer to:

- identification of main dimensions and geometrical coefficients viable to install the pressure vessels adequate to transport the required volume of CNG;
selection of length of pressure vessels with given internal and external diameter under the following crisp constraints for the ship: compliance with intact and damage stability rules, roll period greater than 13 seconds to avoid resonance between roll and wave periods, and avoidance of coupling lateral and vertical motions;
- assessment of midship section structure and lightship weight breakdown;
- power prediction in calm water and in a seaway;
- analysis of intact and damage stability for compliance with IMO rules;
- electric power balance;
- seakeeping assessment for added resistance, motions and accelerations;
- powering and gas consumption prediction at different speeds in a round voyage (cycle);
- total time to perform a cycle;
- one-dimensional vibration calculations to avoid risk of resonance between main modes of hull vibration and propellers speed;
- round-trip modelling;
- DP during STL/SPM connection/disconnection and simultaneous power required for gas compression during loading/offloading operations.

To prevent the design from attaining some unwanted characteristics, all intermediate and final results are subject to constraints which are linear and nonlinear equations of the equality and inequality type. Some relationships between geometrical variables and parameters are used as min-max constraints. In principle, constraints are used for hard type of decisions to distinguish between feasible and unfeasible designs, verifying that the solution remains within allowable design space. Attributes, which are design performance measures, are in fact constraints since their values may be constrained; hence, they may be considered as soft (fuzzy) constraints. In particular, a minimum DP capability was set, requiring ships to withstand at least to head seas (15 deg heading) assuming a wind speed of 20 kn.

2.2 Hub-and-Spoke scheme

It is well known that two major distribution patterns, namely hub-and-spoke and milk-run, are envisaged for CNG marine transportation. In this paper, only the first pattern is considered.

Altogether, four types can be distinguished for the hub-and-spoke scheme:
- Continuous-Continuous (CC): no storage facility is available;
- Continuous-Interrupted (CI): storage facility at receiving point to accelerate the offloading phase;
- Interrupted-Continuous (IC): storage facility at the origin point to speed up the loading phase;
- Interrupted-Interrupted (II): storage facilities at both source and destination points.

Table 2 summarizes the minimum ship capacity and storing capacity required for a given transportation scenario, where:
- $G_n$ is the minimum ship capacity;
- $n$ is the number of ships entering the fleet;
- $Q_{in}$ is the loading rate;
Table 2 Hub-and-spoke schemes

<table>
<thead>
<tr>
<th>Distribution Pattern</th>
<th>Minimum ship capacity</th>
<th>Minimum storage capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Continuous-Continuous (CC)</td>
<td>( G_{n}^{CC} = \frac{T_{rt}}{n-1} \frac{1}{Q_{off} - Q_{on}} )</td>
<td>no storage facility</td>
</tr>
<tr>
<td>Continuous-Interrupted (CI)</td>
<td>( G_{n}^{CI} = \frac{Q_{u}T_{rt}}{n} - \frac{Q_{u} - Q_{off}}{Q_{on}} )</td>
<td>( S_{n,spoke} = G_{n}^{CI} \left( 1 - \frac{Q_{u}}{Q_{on}} \right) )</td>
</tr>
<tr>
<td>Interrupted-Continuous (IC)</td>
<td>( G_{n}^{IC} = \frac{T_{n}^{IC}Q_{h}}{n} )</td>
<td>( S_{n,hub} = G_{n}^{IC} \left( 1 - \frac{Q_{h}}{Q_{on}} \right) )</td>
</tr>
<tr>
<td>Interrupted-Interrupted (II)</td>
<td>( G_{n}^{II} = \frac{T_{n}^{II}Q_{h}}{n} )</td>
<td>( S_{n,hub} = G_{n}^{II} \left( 1 - \frac{Q_{h}}{Q_{on}} \right) )</td>
</tr>
</tbody>
</table>

- \( Q_{off} \) is the offloading rate;
- \( Q_{h} \) is the consumption rate;
- \( Q_{u} \) is the daily supply rate of gas at hub;
- \( T_{rt} = 4t_{c} + 2L/V \) is the time spent to complete a gas distribution cycle (total round-trip time), being \( t_{c} \) the time needed for connecting/disconnecting the ship to/from the buoys, \( L \) the distance from the source site (hub) to the destination site (spoke) and back after gas loading, and \( V \) the unknown service speed;
- \( T_{n}^{IC} = G_{n}^{CC} / Q_{on}^{IC} + G_{n}^{CC} / Q_{off} \) is the actual cycle time if there is a storage at the destination site;
- \( T_{n}^{II} = T_{n}^{IC} (Q_{on}^{II} - Q_{h}) \) is the actual cycle time if there is a storage at both the origin and destination site.

The loading and offloading rates depend on the facilities available on-sites (e.g. gas grid, compressors, storage, etc.). It has been demonstrated [13] that the overall capacity of a CNG fleet, \( G_{f} = nG_{n} \), which substantially has to satisfy shipping of the natural gas available at the origin, is bounded by a lower and an upper limit, respectively, as:

\[
\frac{n}{n-1}T_{rt}Q_{u} \leq G_{f} \leq \frac{n}{n-2}T_{rt}Q_{u}
\]  \( (1) \)

depending on the gas consumption rate, \( Q_{u} \), which can be actually absorbed by the destination terminal. It results that CNG fleets require two ships \( (n = 2) \) at least to ensure continuity of gas delivery. Of course, that holds also in the CC scheme where the offloading rate has to equal the daily consumption rate as per Table 2. Intermittent supply and/or delivery (CI, IC, II schemes) results in less ships composing the fleet, or at least in a fleet with the same number of ships but of reduced size, with corresponding lower Capex and Opex and lower shipping tariff.

2.3 Selection of the optimal CNG fleet

In last decades, there has been a steady increase in the number of optimisation studies carried out for new buildings. However, in the CNG business, as a rule, the approach has not only to be concerned with optimising one single ship, but must be aimed at formulating
optimum fleet for a particular transport of natural gas. Different feasible fleets may well have the same annual cargo capacity, while being made up of ships of varying sizes and different service speeds. Enlarging what stated by Lamb [24], the fleet success substantially depends on its economic success. In evaluating different feasible fleets for CNG marine transport, the optimal (preferred) fleet will be that one which requires the minimum tariff to transport an energy unit (USD/mmBtu).

The optimal CNG fleet composition is strictly connected to the operative scenario, techno-economic characteristics of feasible ships, costs and feasibility of infrastructures, the financial parameters (loans, interest rates, period of reimbursement, depreciation allowances, etc.). Scope of the implemented decision-making process is to establish simultaneously the number of sister ships entering the fleet, as well as to simultaneously identify their capacity and economic service speed, in order to transport a specified volume of gas per year from loading terminal (both onshore and offshore) to destination terminals while achieving the expected rate of return. The definition of the shipping scenario is completed by the distance to the market, the stand-by time, the connection/disconnection time on/from the buoys, the loading and offloading rates, and the possible storage facilities at both and destination sites. Several studies [4,7,13] have highlighted that all these parameters have a strong influence on the best fleet composition and must be taken into account simultaneously.

The main logistic constraint in optimising the CNG fleet composition is to ensure that the delivery site is always supplied with the required volume of gas by a ship during the time other CNG ships are on a round-trip from the destination site to the source and back. The loading and offloading rates play a relevant role in the optimal fleet selection. The identification of the best rates depends on many factors of both technical and economic nature. These rates are strictly correlated to the number of ships in the fleet and to ship capacity. For instance, in the interrupted-continuous scheme the loading rate is solved through the following equation

\[
Q_{on}^{IC} = \frac{G_n^{CC} Q_{on}}{nG_n^{CC} - G_n^{CC} Q_{off} - Q_{on} T_{rt}}
\]

which could be solved by means of a constrained multiobjective optimisation method.

On the contrary, the authors’ strategy has been to establish a set of possible values for loading and offloading rates and to determine the sub-optimal fleet for each couple \((Q_{on}, Q_{off})\) by identifying the feasible sister ship previously stored in a database. Then, the optimal fleet is selected as the sum of ships extracted from the database, requiring the minimum total tariff, e.g. the shipping tariff plus the extra cost incurred by the infrastructure facilities.

Even in the simplest hub-and-spoke distribution pattern, e.g. continuous-continuous service, avoiding concurrent consideration of logistics and conceptual designing of ships with simultaneous merging of technical and economic properties can lead to wrong decision making. Availability of a database of CNG ships with different capacity and evaluated in techno-economic terms at different service speeds is a must in searching for the optimal CNG fleet while complying with the aforementioned logistic constraint.

2.3.1 Cashflow model

Especially in concept design, which impacts on total building cost for 70% at least, cost estimation of alternative technical solutions is a decision making tool of paramount importance. The engineering economics constitutes a basic element of the shipping strategy since optimisation of a fleet composition cannot be a mere technical problem. Economic prognoses constitute the input from which any selection method concludes to an optimal decision making.
In order to evaluate a shipping tariff, a built-in economic model provides monetary value of each alternative CNG project from a shipowner’s financial prospective. Ship costs include the costs from the shipyard, the cost of the operating cycle and elements of cost of the ship through its economic life. Clearly, the cost of construction and operation will affect the shipping tariff, but the financial conditions in the market will have an equally important impact. It is the relationship between the two which forms the conceptual basis for the economic model implemented for decision making about CNG fleet composition.

The most popular tool for analysis of engineering economics is a Discounted CashFlow (DCF) model, which allows to determine the Net Present Value (NPV) for an expected rate of return and over the assumed project life. Discounting allows for the time value of money, which is an effective tactic for evaluating the future value of a project in terms of today’s money. The decision criterion for investment selection is the NPV which can be represented by a linear function of the economic variables, that is, the object function which is the sum of the discounted annual cash flows. Cash flows are calculated on an after-tax basis and are assumed to occur on an end-of-year basis. In formulating the NPV equation, the fiscal depreciation of every ship is established by the linear method on the basis of the scrap value at the end of the ships’ estimated economic life. On the basis of NPV evaluated for each feasible fleet, it is straightforward to determine the best fleet composition, e.g. that one which requires the minimum tariff for unit energy transport by setting NPV = 0 for the expected rate of return to the shipowner.

Calculation of DCF involves deduction of running costs and capital costs from assumed revenue earned. What is left from annual income after these expenditures and amortisation is generally subject to taxes. Income is assumed as the amount of money the shipping company gets for shipping the gas between the origin and destination points. While income is assumed constant year-to-year, some running costs (crew and victualing, maintenance, administration) are escalated over time.

Running costs cover:

- operating costs related to the daily running of the ship (crew, lubricants, stores and consumables, routine maintenance and repairs, insurance, administration), assumed to represent an Italian registered ship with national crew
- voyage costs (gas consumption from main engines and auxiliaries, gas price at wellhead),
- CNG handling costs.

Capital costs include:

- capital repayment (loan to finance the project, terms of loan),
- interest payments (source of loan, interest rate, terms of loan),
- periodic maintenance (age of ship, survey cycles, class regulations, maintenance policy).

Many assumptions are made when calculating the profitability of a CNG fleet. Material costs of the ship are difficult to estimate accurately as public prices from manufacturers are usually not available. Moreover, since CNG ships belong to a quite novel concept, cost estimate of some special features of the ship like gas containment system are necessarily imprecise. Operational cost calculations are somewhat less inaccurate.
2.3.2 Capex

The cost estimate for building an innovative ship and infrastructure facilities (Capex) is performed with a top-down approach for main items by project elements and then by discipline. Each discipline contains details as to outline of the cost estimate breakdown structure, technical data sources, cost data sources and contingencies. Transferring the gas from the production facility to the CNG containment system on board is technically straightforward and requires a minor capital investment for a short subsea pipeline, riser pipes and gas transfer buoys. On average, 5% and 6% of the total investment on the CNG project is expected for loading and offloading terminals, respectively.

As regards the ship building cost, it is evaluated distinguishing between material costs, direct labour costs and overheads, including an expected profit margin by the shipyard. Breakdown of the material cost covers technological main groups such as hull structure, superstructure, engine rooms, propulsors, electrical plant, electronics & control system, auxiliary systems, outfit and furnishings, special systems. Cost of the pressure vessels is included depending on their number and length. Finally, a CGT (compensated gross tons) factor pertaining to an LNG ship of the same size is applied.

At concept design stage, based on statistics of LNG/LPG ships the ship acquisition cost in US dollars can be formulated as:

\[
C_{acq} = 10^3 (C_{HS} + C_M + C_E + C_{PV}) (1 + p)
\]

where \(C_{HS} = 3.365 \ W_{HS}^{0.224}\) is the hull steel cost as a function of steel weight \(W_{HS}\) in metric tonnes, \(C_M = 14.308 \ P_B^{0.669}\) is the machinery and auxiliaries cost depending on engines brake power \(P_B\) in kW, \(C_E = 14.770 \ W_{OD}^{0.932}\) is the equipment cost depending on outfitting, heating, ventilation, air conditioning, electric plant and deck machinery cost depending on their weight \(W_{OD}\) in metric tonnes, \(C_{PV}\) is the total cost of the containment system (pressure vessels, piping, compressors, etc.), \(p\) is the profit margin of the shipyard, in percentage. It is worth noticing that cost of pressure vessels is strictly confidential.

Manhours are the basis of all labor costs, and once estimated, it is only necessary to apply wage rates, overheads and yard profit to arrive at the total labor costs. At the simplest level deemed as sufficient at concept design stage, steelwork labor cost \(C_h\) can be estimated from:

\[
C_h = W_{HS} c_h h_{ch}
\]

where \(h_{ch}\) is manhours required for each steel tonne, \(c_h\) is the wage rate per manhour. It is worth noticing that manhours depend on steel weight, ship length, block coefficient and compensated gross ton coefficient. Productivity data available from an Italian shipyard made it possible to derive the regression relationships for manhours (Fig. 2) distinguishing between structures and outfitting building.

\[\text{Fig. 2 Manhours per tonne of steelwork and outfitting in Italian shipyards}\]
An average reduction factor is given to evaluate economics of scale for multiple ship production, considering the elimination of nonrecurring needs, such as drawings and templates, and considering the improvement resulting from the learning of the labour force.

2.3.1 Opex

Operating costs are the ongoing expenses which include most of the daily operational costs during the ship lifetime, but excluding the gas (fuel) which is assumed to be included in the voyage costs. The operation costs of the ship are divided in three main categories: fuel costs, personnel costs, and maintenance.

Fuel costs are estimated based on the operational profile calculated earlier. Crew salary costs are estimated based on the number of personnel onboard and their jobs. There are 20-25 crew members (captain, three officers, seamen). In addition to basic salaries and wages, personnel costs include basic salaries and wages, victuals, social insurance, repatriation expenses, recruitment and training costs. Maintenance costs are divided into hull maintenance, engine maintenance and general maintenance. Engine maintenance is assumed to cost about 3750 USD/kW per year, whereas general maintenance is assumed to cost 0.5% of ship total price per year. Insurance costs are assumed to be 1.2% of ship total price per year.

In the overall maintenance cost, a significant part is on the dry-docking cost $C_{DD}$. It has been estimated as [25]:

$$C_{DD} = a + b \cdot y + c \cdot DWT + d \cdot SD + e \cdot P_B$$

(5)

where $y$ denotes number of years from the date the ship is delivered, $DWT$ its deadweight and $SD$ are the days from the ship’s arrival date to the yard and the departure date. The coefficients in the regression formula have been determined from data on a number of chemical carriers with deadweight below 30,000 tons.

3. Operative scenario and financial parameters

The case study is centred on the specific case of the gas shipping in the Mediterranean Sea area, from North Africa to Europe. More in detail, the case of gas transportation between the offshore field of Zohr, part of the Egyptian economic area, and the South of Adriatic Sea (Brindisi). The selected scenario is representative of a source-destination distance of about 1000 nm and a production rate of 2 billion cubic meters of gas (bcm) per year.

A hub-and-spoke continuous-continuous service scheme is adopted, assuming a constant service speed, to be determined as economically optimal, over the entire voyage. Furthermore, the following assumption have been set:

- Loading/offloading rate: 5 mmmscm/d, where 4 mmmscm/d is the delivered capacity;
- STL connection and SPM disconnection time: 1.5 hours per operation;
- Operating days per year: 355 (as required by rules, the CNG ships will be taken out of service for dry-docking once every 5 years for 40 days and at each 30-month intermediate point for about one week during an underwater survey; this totally amounts to about 50 days in 5 years).

In addition, in order to properly take into account unpredictable situations that might occur to the ship during its lifecycle, each one-way voyage duration is rounded up to the nearest half-a-day time.

Furthermore, in the present case study, the following financial assumptions are made: monetary values at a base value of 2018 US dollars and an inflation rate of 2% for revenues and expenses, a 30% corporate income tax; linear depreciation of ships for a period of 15 years. Capital for ship building will be acquired through both equity and debt, at a rate of 30% equity to 70% debt. The loan term is assumed to be 8 years at an interest rate of 5.5%.
Interest payments begin at the end of the project’s initial year. For the three years of the CNG ships’ construction, the principal repayment is assumed to be 40%, 30%, and 30% of the total project cost. The expected rate of return for the CNG project is assumed to be 12.5%, a reasonably fair standard value for the oil & gas companies. Italian shipyards average hourly cost has been assumed in building cost estimation, whilst daily operating costs were derived from average data for LNG ships, while fuel costs from present market information concerning price of natural gas at wellhead.

4. Propulsion systems

Different propulsion systems installations have a remarkable effect on shipping efficiency when applied on CNG ships [20]. The impact on hull forms (not limited on the aft-body), propeller design and general arrangement, should be taken into account since the concept design stage. Moreover, since natural gas is flammable and potentially explosive, the application of redundant propulsion systems and steering gears is advisable. Thus, the single screw propulsion system should be avoided in favour of twin-screw arrangements. In the present study, a comparison between two alternative twin-screw solutions is carried out highlighting the effect on the required freight rate (tariff) and DP capability. The first solution is a twin-skeg ship with conventional mechanic propulsion driven by 2-stroke dual-fuel engines. The second is a full diesel-electric solution with pods. In this section the two propulsion systems are briefly described highlighting the differences and peculiarities rising up in the ship modelling during concept design stage.

4.1 2-stroke diesel engine system

The first option under investigation is a conventional propulsion system adopting 2-stroke dual fuel engines as prime movers. Such a solution adopts high efficiency engines, rotating at low speeds to be directly coupled with the propellers. Hence, all the propulsive drive will be extremely essential, reducing the amount of losses from engine brake power \( P_B \) to propeller delivered power \( P_D \). However, this kind of propulsion system requires a propeller rotating at low speed. Such a low rotational rate implies utilisation of large diameter propellers, being mountable only in case of a sufficiently high draught. Because of the adoption of light Type III PVs, it is not reasonable to presume to have a high draught even for large capacity ships; for such a reason, in order to keep the propeller diameter under control, the selected hull form for this kind of propulsive solution will be of the twin-skeg type.

This kind of propulsion layout requires a different modelling of the propulsion since conceptual calculations. Although the total resistance in calm water can be evaluated according to statistical methods [26], the determination of propulsive coefficients (especially for wake fraction \( w \)) is not comparable with a conventional twin-screw ship. The presence of the gondola is effectively decreasing the relative flow velocity coming into the propeller disk, leading to values of \( w \) more similar to those of single-screw ships. That is why the wake fraction \( w \) has been estimated according to an equivalent single-screw statistic method [26], while the thrust deduction factor \( t \) was estimated with conventional twin-screw statistical formula [27].

To take into account the actual operative scenario since propeller selection process, also added wave resistance has been included according to statistical formulations [28]. The propeller selection has been carried out with a bounded iterative procedure; considering 4-bladed B-Series propellers [29], a maximum diameter \( D_{\text{MAX}} \) according to hull clearances and a database of 2-strokes diesel engines. Selection of the diesel engine has to ensure the matching between propeller and engine revolution rates.
4.2 Diesel-electric system

Another interesting solution for the propulsive system can be the adoption of a diesel-electric propulsion. The adoption of such a system is required by the installation of pod propulsive devices, which can be considered as an improving solution for the positioning ability of the ship during loading/offloading operations on/from a fixed buoy. Besides, the installation of the pods leads to an aft-body hull form totally different with respect to the twin-skeg, but more similar to a conventional twin-screw ship. It means that the metamodel used to define the geometric hull characteristics in ships generation will be different between the two cases, in order to capture the peculiarities of each candidate hull form.

The propulsive coefficients evaluation in a podded ship can be performed, as first approximation, in the same way of a twin-screw ship. However, a different modelling of kinematics and dynamics coefficients of the propulsors is needed. On this purpose, the reproduction of characteristic thrust and torque curves can be done starting from B-series propeller open water curve and applying a procedure derived from thruster units [30]. Open-water curves have been modified according to the following coefficients:

\[
\lambda_T = \frac{K_T}{K'_T}, \quad \lambda_Q = \frac{K_Q}{K'_Q}, \quad e_T = \frac{J}{J_0}, \quad e_Q = \frac{J}{J_0}
\]  

(6)

where subscript 0 denotes original B-series values. The coefficients as per equation (6) modify the standard open water curve, simulating the behaviour of the complete pod unit. An example is given in Figure 3. The tuning of the coefficients has been performed to reproduce the pod hydrodynamic characteristics given by the manufacturer.

Fig. 3 Open water chart comparison between a B series propeller (black) and equivalent pod unit (red).
4.3 Concept design ship layout modelling

The selection of two different propulsive systems influences not only the hull form modelling and algorithms for power performance prediction, but also the internal layout of the ship, hence, the areas needed to fit the engine room and consequently the main dimensions of the ship. As described in [18], the internal layout modelling of a CNG ship can be done as a function of the so-called primitive cargo unit, which for a CNG ship is the PV diameter. Some specific spaces should be taken into account. In fact, besides cargo length and engine room length (common to all the type of ships), other longitudinal spaces should be dedicated to equipment necessary to gas loading and offloading process. This is the case of the conical recess in the fore part of the flat of bottom for connection to the STL system and of the main compressors. For such a reason dedicated space should be considered to install the above mentioned equipment. A schematic representation of the different configurations is given in Figure 4. With such a discretisation, considering the system origin on the fore perpendicular, the length overall \( L_{OA} \) definition becomes:

\[
L_{OA}_{\text{win-safe}} = L_{SM} + L_{CB} + L_{STL} + L_{H} + L_{ER} + L_{MAN} 
\]

\[
L_{OA}_{\text{pod}} = L_{SM} + L_{CB} + L_{STL} + L_{H} + L_{ER} 
\]

where \( L_{SM} \) is the stem overhang length, \( L_{CB} \) is the collision bulkhead distance from the origin, \( L_{STL} \) is the STL longitudinal space, \( L_{H} \) is the total length of the holds, \( L_{ER} \) is the engine room length, and \( L_{MAN} \) is the manoeuvring machinery space. It must be noted that in the podded case \( L_{ER} \) includes the length of both pods \( L_{POD} \) and gen sets \( L_{GEN} \), so that there is no necessity to add an extra space for manoeuvring equipment. The engine room length is then function of the machinery size selected by the mathematical metamodel.

5. Dynamic positioning

A CNG ship, as it has been presented throughout this paper, has another issue, e.g. the dynamic positioning, to tackle in order to load/offload the gas in the expected time. In fact, while the ship is stationing on the STL buoy for the gas loading, should keep position and then manoeuvre autonomously [31], thus requiring a DP system. Therefore, the DP issue should be considered since the concept design stage.
The determination of the capabilities of a DP system is not an easy task and may be assessed typically using two different approaches [32]: a quasi-steady approach and/or a time domain one. The time domain approach [33,34] is used once the available ship details are sufficient to accurately define the whole system mounted on board. This is not the case at the concept design stage, where fast and sufficiently reliable tools should be used to assess the performance of each attribute. For such a reason, a quasi-steady approach [35] has been used to perform station-keeping capability of the ship, where only the equilibrium between the external forces and the thruster delivered forces should be evaluated on the horizontal plane:

\[
\begin{align*}
F_{X_{\text{ENV}}} + F_{X_{\text{THR}}} + F_{X_{\text{EXT}}} &= 0 \\
F_{Y_{\text{ENV}}} + F_{Y_{\text{THR}}} + F_{Y_{\text{EXT}}} &= 0 \\
N_{Z_{\text{ENV}}} + N_{Z_{\text{THR}}} + N_{Z_{\text{EXT}}} &= 0
\end{align*}
\]  

(9)

In system (9), \( F_{\text{ENV}} \), \( N_{\text{ENV}} \) are the environmental, \( F_{\text{THR}} \), \( N_{\text{THR}} \) the thruster and \( F_{\text{EXT}} \), \( N_{\text{EXT}} \) the external forces and moments. By modelling the DP in the concept design stage, the external forces have been neglected, since they are related to the mutual interaction between the ship and STL buoy, forces that are initially unknown. Since it will be extremely complicated and time consuming to perform a complete DP assessment during a generation process as described in Section 2, it has been decided to restrict the analysis to a single environmental condition and encounter angle. Here, only the 15-deg case is calculated per each generated ship, considering a wind speed of 20 knots and a collinear wave according to IMCA correlation [36] and a 1 knot collinear current speed. By doing that, each generated ship can be accepted or discarded according to the selected criteria. This kind of approach, slightly reduces the calculation time and allows to use more reliable allocation algorithm for the equilibrium resolution.

5.1 Thrust allocation algorithm

Between the several possibilities to solve system (9), here an optimisation algorithm has been used, capable to manage non-linear objective functions subject to non-linear constraints [37]. As objective function, the minimum absorbed power has been selected, considering the following non-quadratic formulation:

\[
f(x) = \sum_{i=1}^{n} x_{i}^{\frac{1}{2}}
\]  

(10)

where \( n \) is the number of thrust devices and \( x_{i} \) are the thrust values delivered by each single device. In the specific, for the total thrust determination, interaction effects have been considered according to ABS indications [38]. Moreover, for the back thrusters of the podded ship, forbidden zones have been implemented to avoid the possibilities of the thruster to work in an excessive interaction zone, due to the presence of the skeg and the other thruster.

In case of the twin-skeg ship, the rudder has been considered of the Becker type, using lift and drag coefficients coming out from literature.

5.2 Environmental loads determination

The environmental forces have been modelled in such a way to take into account the effects of wind, waves and current. Since it is difficult to determine in the conceptual design stage the current and wave loads as a function of the main parameters of the ship, simplified methods given by regulations have been applied [39]. As regards the wind loads, a more detailed analysis has been carried out, since more information are available from the scientific literature.
Several possibilities are available to determine the wind loads in an early-design stage [40], adopting coefficients that depend on the main parameters of the ship superstructures. In literature, different databases are available for direct use of experimentally derived coefficients for wind loads [41]. However, the novelty of the ship type proposed in this study has made it impossible to find a suitable comparable geometry in standard databases. For such a reason other methods have been investigated to determine the wind loads.

Here Isherwood [42], Fujiwara [43] and DNV [39] methods are compared on the standard superstructure layout considered for the two ship types. The methods are using different ways to obtain non-dimensional wind coefficients. Here all the coefficients are represented in the following form to have a homogeneous comparison:

\[
\begin{align*}
C_X &= \frac{F_X}{0.5 \rho_{\text{air}} A_T V_W^2} \\
C_Y &= \frac{F_Y}{0.5 \rho_{\text{air}} A_L V_W^2} \\
N_Z &= \frac{N_{Z}}{0.5 \rho_{\text{air}} A_L L_{OA} V_W^2}
\end{align*}
\]

where \( \rho_{\text{air}} \) is the air density, \( V_W \) the wind speed, \( L_{OA} \) the overall length and \( A_T, A_L \) the transversal and lateral area projections respectively. \( A_T \) and \( A_L \) are estimated as a function of ship main dimensions and PV length.

In Figure 5, a comparison is presented between the different methods for the two ship types. It can be seen that the Isherwood method is more scattered and is giving the lower values for the maximum loads. The DNV method is giving a really simple trend, being in line with the maximum loads of Isherwood method for lateral winds. The Fujiwara method is giving the higher values for the transverse and lateral forces. It has been decided to adopt the Fujiwara method to perform the DP calculations, in such a way to take a margin on the loads.

6. Best ships comparison

The design process described in section 2 has been applied to the operative scenario defined in section 3 leading to assess the best CNG fleet composition for both the twin-skeg and podded ship concepts installing different propulsion systems. Figure 6 shows all the feasible fleets for both the ship families. For each fixed number of sister ships (5,6,7,8 and 9 ships), the combination of ship capacity and service speed (from 17 to 24 knots in step of 1 knot) is determining the shipping tariff. Than the minimum tariff can be evaluated per each fleet composed by different number of ships. Comparing the relative minima, the optimum fleet is determined. Some scattering on tariffs between contiguous design solutions is mainly caused by the discrete nature of machinery sizes available on the market. This behaviour is more evident for the podded solution.

For both families of ships, an optimal fleet can be defined, i.e. the one requiring the minimum tariff. In the given scenario, for the twin-skeg solution the best fleet composition corresponds to 6 ships sailing at a service speed of 23 knots, resulting in a total gas capacity of 290 mmscf per ship. As to the podded solution, the diagram presents less evidence for the optimum determination. In fact, there is a range of ship capacities yielding fleets requiring almost the same shipping tariff. Nevertheless, an absolute minimum can be determined corresponding to a fleet of 7 ships with lower service speed of 20 knots and a capacity of 251.6 mmscf.
Fig. 6 Shipping tariff for all the feasible fleet composition for both podded and twin-skeg solutions.

Fig. 7 Body plan of the twin-skeg ship (top) and podded ship (bottom).
Fig. 8 Specific resistance comparison between the two ship types.

Table 3 Main particulars of the best ships composing the two optimal fleets.

<table>
<thead>
<tr>
<th>Items</th>
<th>Symbol</th>
<th>Twin-skeg</th>
<th>Podded</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length over-all</td>
<td>$L_{OA}$</td>
<td>216.82</td>
<td>200.20</td>
<td>m</td>
</tr>
<tr>
<td>Length between perpendiculars</td>
<td>$L_{BP}$</td>
<td>201.02</td>
<td>185.60</td>
<td>m</td>
</tr>
<tr>
<td>Breadth</td>
<td>$B$</td>
<td>30.82</td>
<td>28.86</td>
<td>m</td>
</tr>
<tr>
<td>Design draught</td>
<td>$T$</td>
<td>7.42</td>
<td>6.00</td>
<td>m</td>
</tr>
<tr>
<td>Volume of displacement</td>
<td>$\nabla$</td>
<td>27950.3</td>
<td>22304.5</td>
<td>m$^3$</td>
</tr>
<tr>
<td>Prismatic coefficient</td>
<td>$C_p$</td>
<td>0.622</td>
<td>0.710</td>
<td>–</td>
</tr>
<tr>
<td>Midship coefficient</td>
<td>$C_M$</td>
<td>0.978</td>
<td>0.978</td>
<td>–</td>
</tr>
<tr>
<td>Waterplane area coefficient</td>
<td>$C_{WP}$</td>
<td>0.830</td>
<td>0.886</td>
<td>–</td>
</tr>
<tr>
<td>Transversal projected exposed area</td>
<td>$A_T$</td>
<td>930.74</td>
<td>856.58</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Lateral projected exposed area</td>
<td>$A_L$</td>
<td>4855.98</td>
<td>4271.58</td>
<td>m$^2$</td>
</tr>
<tr>
<td>Service speed</td>
<td>$V$</td>
<td>23.0</td>
<td>20.0</td>
<td>kn</td>
</tr>
<tr>
<td>Available shaft power</td>
<td>$P_S$</td>
<td>22304.5</td>
<td>19601.0</td>
<td>kW</td>
</tr>
<tr>
<td>Propeller diameter</td>
<td>$D$</td>
<td>4.80</td>
<td>4.35</td>
<td>m</td>
</tr>
<tr>
<td>Pitch diameter ratio</td>
<td>$P/D$</td>
<td>1.325</td>
<td>0.874</td>
<td>–</td>
</tr>
<tr>
<td>Expanded area ratio</td>
<td>$A_E/A_0$</td>
<td>0.634</td>
<td>0.598</td>
<td>–</td>
</tr>
<tr>
<td>Blades number</td>
<td>$Z$</td>
<td>4</td>
<td>4</td>
<td>–</td>
</tr>
<tr>
<td>Bow thruster propeller diameter</td>
<td>$D_T$</td>
<td>2.40</td>
<td>2.20</td>
<td>m</td>
</tr>
<tr>
<td>Bow thruster nominal power</td>
<td>$P_T$</td>
<td>1825.0</td>
<td>1500.0</td>
<td>kW</td>
</tr>
<tr>
<td>Ship capacity</td>
<td>$\nabla_g$</td>
<td>290.0</td>
<td>251.6</td>
<td>mm$^{scf}$</td>
</tr>
<tr>
<td>Number of PV</td>
<td>$N_{PV}$</td>
<td>305</td>
<td>287</td>
<td>–</td>
</tr>
<tr>
<td>Length of PV</td>
<td>$l_{PV}$</td>
<td>25.40</td>
<td>23.20</td>
<td>m</td>
</tr>
<tr>
<td>Delivered gas per cycle</td>
<td>$w_{gd}$</td>
<td>4576.9</td>
<td>3945.8</td>
<td>t</td>
</tr>
<tr>
<td>Gas consumption for propulsion</td>
<td>$w_{gc}$</td>
<td>211.6</td>
<td>168.1</td>
<td>t</td>
</tr>
<tr>
<td>Cycles per year</td>
<td>$c_{py}$</td>
<td>51.5</td>
<td>50</td>
<td>–</td>
</tr>
</tbody>
</table>
In both cases, the service speed corresponding to the minimum tariff for a given number of ships increases as the number of ships decreases. Furthermore, it is worth noticing how, in the same operative scenario, even the number of sister ships changes with adoption of different propulsion systems resulting in very different design solutions in terms of main dimensions, capacity, service speed, etc. All the characteristics of the two optimal ships (Fig. 7) have been assessed via metamodels; the resulting main characteristics are provided in Table 3. Analyzing the main dimensions and coefficients, it rises up that the podded ship, even though of smaller size, is less slender than the twin-skeg solution.

The main cause is the higher Froude number ($F_n=0.26$) for the twin-skeg ship at service speed of 23 knots, whereas the podded ship has $F_n=0.24$. The different required capacities of the two solutions have a heavy impact on the ratio between cargo capacity and total displacement, which is reflected on the block coefficient too.

The difference between hull form shapes has led to obtain a better specific resistance for the twin-skeg ship in all the speed range, as shown in Figure 89. The same consideration is still valid when propulsive issues are considered. In fact, analysing the trial condition for the two ships, the following conclusion can be drawn for the two ships considering the different propulsive power mounted on board of each configuration as per Table 3.

For the twin-skeg concept, mounting a propeller with geometrical characteristics similar to the one of Table 3, a sustained speed of 24.45 knots is expected with propellers rotating at 124.0 rpm absorbing a shaft power of 22304.5 kW without presence of wind and waves. Under the same conditions, a speed of 21.55 knots with propeller rotating at 203.8 rpm with an available shaft power of 19601.0 kW is expected for the podded ship. It must be noted that the selected propellers parameters have been chosen searching for the design point at service speed, taking into account the added resistance evaluated in the Eastern Mediterranean sea.
The results of the speed-power predictions on trials have been reported in Figure 9. It can be seen that the propulsive curve for the podded solution is rapidly becoming steeper after 21.5 knots. That is due to rising up of cavitation, which dramatically increases the propeller revolution rates and consequently the absorbed power.

Even though the two ship types intrinsically respect the DP constraints, which filter the feasible designs during the generation process, it is interesting to visualize the different capability plots [36] of the two optimal ships. The capability plot reports the maximum sustainable wind that the DP system can face at each encounter angle, making easy a reliable comparison between the station-keeping ability of the two ships. In Figure 10, the two DP capabilities of the two optimal ship concepts are presented on the same plot, highlighting significant differences.

In fact, while compliant with imposed DP constraints, the twin-skeg ship has lower capability with respect to the podded ship. This is quite reasonable to predict, since the propeller-plus-rudder configuration is giving less freedom to properly face the environmental loads, especially from stern encounter angles. For the specific operations near the buoy, it is reasonable to presume that the ship will always be oriented in a direction around 0 deg; that’s why the constraint was assumed at 15 deg during the generation process of candidate ships. Given such a big difference between the two resulting capability plots, it should be then advisable to investigate whether the operability of the two ships will be reduced or not depending on the different configurations.
At concept design stage it is hard to directly consider the effective downtime period due to adverse weather during DP operation, which is important for the logistic model described in Section 2. By doing a detailed analysis on the effective DP operability of the ship, then the downtime period could be better determined, improving the quality of the fleet determination. In fact, as mentioned, the logistic is considering the same downtime period for both ships. For such a reason, DP issue should be kept in serious consideration in further studies and analyses.

Besides pure hydrodynamic considerations, it is interesting to analyse the two different internal layouts coming from the conceptual design phase. As mentioned beforehand, the developed design method is suitable to analyse the internal layouts on the base of the primitive cargo units, fitting the most suitable holds configuration according to the ship dimensions. In Figure 11 it is possible to visualise the two resulting conceptual general arrangement plans which are strongly influenced by both cargo spaces and machinery spaces.

As highlighted in [18], the main constraint acting on cargo spaces is the waterline width at tank-top, since it is mandatory to assure a minimum distance of cargo holds from hull shell according to the International Gas Code. Both ships have a cargo space subdivided in 5 holds, but the number of PV per hold is higher for the twin-skeg ship. The maximum transversal number is 9 for both designs, since the differences in breadth between the two units did not allow installation of an additional PV for the twin-skeg case. However, the number of lines per hold is different, being 7 for the podded solution and 8 for the twin-skeg. It is also interesting
to notice how the holds are disposed. The podded ship, having a relatively bigger parallel middle body, is suitable to have almost 4 fully loaded holds; on the contrary, the twin-skeg solution is having only three holds fully loaded, while the two foremost holds are strictly limited by the waterline width at tank-top. As discussed in subsection 4.3, the propulsive system has a strong influence on the general arrangement. In fact, the engine room for the podded ship is less extended compared to the twin-skeg solution, leading to a much shorter afterbody. The twin-skeg ship, even though is slenderer, has a more extended parallel body ranging from section 8 to 10, while it is limited to sections 8 and 10 in the podded ship.

7. Conclusions

A process aimed to determine an optimal CNG fleet has been presented, considering a specific transportation scenario in the Mediterranean Sea. Two different ship concepts have been conceived, designed at conceptual design level and analysed in techno-economic terms. Comparison between the shipping tariffs on the selected scenario for the optimal fleets composed by two ship concepts has highlighted that the twin-skeg solution has a lower tariff compared to the podded one. Analysing more in detail the two ship types composing the optimal fleets, it results that the twin-skeg solution presents better hydrodynamic performances with respect to the podded solution. The tariff evaluation of the podded ship shows some discontinuities that are due to the fact that pod size cannot be customized, but are roundly fixed by the manufacturers. The twin-skeg solution is not presenting this discontinuity since the propeller selection is more flexible. Also the dynamic positioning problem has been considered for the two solutions. Besides the constraints on a single DP criteria for an environmental condition and heading, the two ships composing the best fleets have been compared considering the capability plot. The analysis highlights that the podded solution has a higher capability with respect to the twin-skeg one, since the pod system is more flexible to balance the environmental and external load. The possibility to consider also DP inside the concept design stage has to be further analysed, since it can be source of a better definition of the ship operability. This is influencing the downtime period of the ships, relaxing the constraints in the logistic model. The developed strategy for optimal fleet selection simultaneously provides the number of fleet’s ships, the gas capacity and the economically service speed of each sister ship on the basis of minimising a shipping tariff at the expected rate of return.

REFERENCES


Determination of an Optimal Fleet for a CNG Transportation Scenario in the Mediterranean Sea

Francesco Mauro, Braidotti Luca, Giorgio Trincas


Submitted: 13.12.2018. Francesco Mauro (*corresponding author), fmauro@units.it
Luca Braidotti, lbraidotti@units.it
Accepted: 23.05.2019. Giorgio Trincas, trincas@units.it
Department of Engineering and Architecture, University of Trieste, Via Valerio 10, 34127 Trieste (Italy)
EFFECT OF BREAKING WAVE SHAPE ON IMPACT LOAD ON A MONOPILE STRUCTURE

UDC 52-782: 621.3.016.3
Original scientific paper

Summary

A numerical model is derived to investigate the effect of breaking wave shape on impact load on a monopile structure. The derived model combines potential flow model with a Navier-Stokes/VOF solution. The analysis indicates that the breaking wave impact on a monopile structure results in an extremely rapid increase of pressure to high amplitudes. The peak impact pressure occurs in the region below the overturning wave jet. The breaking wave impact leads to extremely high slamming forces. It is observed that the slamming coefficient corresponding to the peak impact force approaches $2\pi$. The area directly affected by the impact force is much higher than the impact area considered in engineering practice. Moreover, the analysis shows that the vertical load distribution is far more realistic than a rectangular shape distribution commonly applied in engineering practice. The results also show that the parameters of the rectangular shape distribution applied in engineering practice are complex function of the breaking wave shape and cannot be uniquely defined beforehand. This is because the vertical load distribution strongly depends on breaking wave shape and it is difficult to uniquely approximate such a complex load distribution by a rectangle. The derived results are compared with experimental data from laboratory experiments on irregular breaking wave loads on a monopile structure. Numerical results are in reasonable agreement with experimental data.

Key words: breaking wave; wave impact forces; wave impact load distribution; monopile structure; Computational Fluid Dynamics

1. Introduction

The prediction of wave loads on maritime structures is of fundamental importance for coastal and offshore engineering. For slender cylindrical structures the breaking wave impact load is a dominant component of hydrodynamic load. Although many studies have been conducted on the interaction between breaking wave and a vertical cylindrical structure, much uncertainty remains. A better understanding of the phenomena may lead to an improved design methodology and eventually to optimization of numerous coastal and offshore structures, including monopile support structures for offshore wind turbines.
In general, the prediction of wave loads on maritime structures must include nonlinear wave load component that may exceed a corresponding first-order quantity (Sulisz, 1998, 2013) several folds. However, for slender vertical cylindrical structures, especially located in relatively shallow waters, wave impact load due to breaking waves constitute a dominant component of the total load and it must be included in the design analysis. The problem is that the breaking wave impact on a vertical cylindrical structure is a complex 3D phenomenon, characterized by very short durations and extremely high pressures.

Up to now, the interaction between breaking waves and vertical cylindrical structures have mainly been studied experimentally. However, the laboratory measurements are very challenging and uncertainties are high. The values of measured peak pressure are very scattered, even when experimental conditions are repeated such as in the study of Zhou et al. (1991). The peak pressure generally range between \(1-50 \rho c_b^2\), where \(c_b\) is the wave phase speed. Moreover, the disadvantage of experimental studies is that the resolution of the pressure measuring points is relatively low which makes complete understanding of the phenomenon difficult. Measurements from pressure transducers are usually obtained only every 10-15 degree around the cylindrical span (Hildebrandt & Schlurmann 2012), which is not sufficient to conduct a detailed analysis of breaking wave impact on a cylindrical structure.

The application of numerical model enables us to evaluate impact pressures on the structure with high spatial and temporal resolution. Therefore, the results from numerical model can help to improve the understanding of the impact of breaking waves on a structure. In the last two decades, numerous numerical studies have been conducted on the attack of breaking waves on a vertical cylinder. The typical numerical models are based on the solution of the Navier-Stokes (NS) equations for a two-phase incompressible flow by applying the finite volume method. These numerical models can represent the breaking wave characteristic with sufficient accuracy, as shown by Chella et al. (2015) where the numerical results are validated with experimental data of Ting & Kirby (1996). The numerical solution of the breaking solitary wave impact on a cylinder is successfully validated with the laboratory PIV measurements by Mo et al. (2013). The potential of the numerical models for calculation of the violent wave loads on a monopile structure is presented in study of Bredmose& Jacobsen (2010, 2011). However, their numerical results were not compared with experimental data. Xiao & Huang (2015) conducted analysis on breaking solitary wave loads on a pile installed at different positions along an inclined bottom. The computed breaking wave forces from their study are consistent with the numerical results of Mo et al. (2013). The results from the study of Xiao & Huang (2015) show that the reduction of wave loads can be achieved by a proper selection of the location of a pile on a sloping beach.

Choi et al. (2015) investigated the effect of the vibration of a structure on hydrodynamic loads. They validated their numerical model with the filtered and the Empirical Mode Decomposition data from the study of Irschik et al. (2002), which are also used for the validation of the numerical model of Kamath et al. (2016). Kamath et al. (2016) investigated different stages of the plunging breaking wave impact on a vertical cylinder. A similar approach was applied in laboratory experiments by Wienke et al. (2001). Both studies show that the location of the cylinder with respect to the wave breaking point has a significant effect on breaking wave forces. The highest force occurs when the overturning wave jet hits the cylinder just below the wave crest level, and the lowest force is obtained when the wave breaks behind the cylinder.

While the most numerical studies include analysis of wave impact force, the pressure and load distribution on the structure during the wave impact are rarely discussed. In recent study of Ghadirian et al. (2016), discussion on the impact pressure distribution is mainly
related to the validation of the numerical model. More detailed discussion on the impact pressure distribution during the wave impact is provided by Hildebrandt & Schlurmann (2013). They investigated temporal impact pressure distribution on the tripod foundation due to phase-focused breaking wave attack. The model is validated with measured wave elevations and impact pressures obtained from the large scale model tests (1:12). By integrating the computed impact pressures around the structure, the temporal characteristic of the vertical load distribution is estimated. The maximum obtained slamming coefficient is $C_s=1.1\pi$, which is considerably lower than the slamming coefficients assessed by applying a simplified approach by Wienke (2001), $C_s=2\pi$.

The effect of the breaking wave shape on the characteristic of the impact pressures and the vertical load distribution has not been investigated so far. The objective of this study is to investigate the effect of the breaking wave shape on the impact wave loads on a monopile structure. The derived results are compared with experimental data from laboratory experiments conducted on irregular breaking wave impact on a monopile structure.

2. Theoretical models

2.1 Analytical approach

The inline force on the slender cylindrical structures excited by the breaking wave is usually calculated as:

$$ F = F_M + F_i $$

The load $F_M$ in this study is approximated by the Morison’s equation (Morison et al., 1950), which is calculated assuming corrections presented by Rainey (1989) for cylinder where the axial dimension is not slender. The impact load $F_i$ induced by the breaking wave is calculated according to the recommendations specified in DNV (2010, 2014) and IEC (2005). According to DNV (2010, 2014):

$$ F_i = \frac{1}{2} \rho C_s A u^2 $$

where $A$ is the area on the structure which is exposed to the slamming force, and $C_s$ is the slamming coefficient. The velocity $u$ should be taken as $1.2c_b$ of the breaking wave height $H_b$.

The area exposed to the wave impact corresponds to the height of $0.25H_b$ and the azimuth angle 45°.

$$ A = \frac{45}{360} \pi D \frac{H_b}{4} $$

For the smooth cylindrical surface the slamming coefficient is in the range $3<C_s<2\pi$. According to IEC (2005), the impact force is calculated based on a simplified approach proposed by Wienke (2001).

$$ F_i(t) = \rho c_b^2 R C_s(t) \lambda \eta_b $$

where the slamming coefficient $C_s(t)$ is adapted from a simplified solution of an infinitely long cylinder hitting the calm water with the constant speed. The curling factor $\lambda$ defines the vertical area of the impact with respect to the wave crest height $\eta_b$. In the study of Wienke & Oumeraci (2005) the curling factor is estimated semi-empirically according to the experimentally measured impact force and the maximum theoretical value of the slamming
Coefficient \( C_s(t=0)=2\pi \). For the case of plunging breaking wave impact on the vertical pile, Wienke & Oumeraci (2005) assessed \( \lambda = 0.4-0.6 \).

2.2 Numerical approach

A numerical model used in this study is based on the decomposition technique suggested by Paulsen et al. (2014), where the wave propagation in the outer region is solved by applying a fully nonlinear potential flow model, OceanWave3D, while the process of wave breaking and the breaking wave impact on a structure is derived by applying the NS equations and the open-source CFD toolbox OpenFoam®.

The NS equations are solved by applying the Volume of Fluid (VOF) method. The incompressible NS-VOF set of equations are discretized using a finite volume approximation on unstructured grids. The conservation of mass is governed by the continuity equation:

\[
\nabla \cdot \mathbf{u} = 0
\]

(5)

where \( \mathbf{u} = (u, v, w) \) and \( u, v \) and \( w \) are the velocity components in the Cartesian coordinate system. The incompressible Navier-Stokes equation is:

\[
\frac{\partial}{\partial t} \rho \mathbf{u} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}^T) = -\nabla p^* - (\mathbf{g} \cdot \mathbf{x}) \rho + \nabla \cdot (\mu \nabla \mathbf{u})
\]

(6)

where \( \rho \) is the density, \( p^* \) is the pressure in excess of the hydrostatic pressure, \( \mathbf{g} \) is the acceleration due to gravity, \( \mu \) is the dynamic molecular viscosity. The free surface separating the air and water phase is captured using a volume of fluid surface capturing scheme, which solves the following equation for the water volume fraction \( \alpha \):

\[
\frac{\partial \alpha}{\partial t} + \nabla \cdot \mathbf{u} \alpha + \nabla \cdot \mathbf{u} (1 - \alpha) = 0
\]

(7)

In which \( \mathbf{u}_r \) is the relative velocity, which helps to retain a sharp water-air interface (Berberović et al., 2009). The marker function \( \alpha \) is 1 when the computational cell is filled with water and 0 when it is empty. In the free surface zone the marker function will have a value in the interval \( \alpha \in [0;1] \) indicating the volume fraction of water and air, respectively. The fluid density and viscosity are assumed continuous and differentiable in the entire domain and the following linear properties are adopted:

\[
\rho = \alpha \rho_w + (1 - \alpha) \rho_a
\]

\[
\mu = \alpha \mu_w + (1 - \alpha) \mu_a
\]

(8)

The time step is controlled by adaptive time stepping procedure based on Courant-Friedrichs-Lewy criterion. For all the computations, the maximum Courant number is kept below 0.2. The symmetry plane is introduced and only half of the domain is considered. The width of the numerical domain is \( 5D \). The solution of the NS-VOF in the inlet and outlet zones are relaxed towards the known solution of the potential flow (Fig. 1).

Fig. 1 Decomposed numerical domain
At the atmosphere boundary inlet/outlet boundary conditions are applied. At the seabed and the lateral boundary the slip condition is applied. Moreover, the slip condition is also applied on the monopile structure, as a result the viscous effects on wave loads are neglected. In the area of the breaking wave impact on the structure, the viscous effect can be neglected due to impulsive loading, while in the area below the breaking wave impact, which is characterized by an oscillating flow, the inertia forces dominate due to a low Keulegan-Carpenter number $KC<15$ (e.g. Lupieri & Contento, 2016).

The sharpness of the air-water interface where $\alpha\epsilon[0;1]$ depends on the size of the computational grid. The thickness of the air-water interface is higher for the coarser computational grid. The density of the air-water mixture depends on the marker function $\alpha$. For the higher thickness of the air-water interface, the rate of change from $\rho=\rho_a$ to $\rho=\rho_w$ is slower, which affects the pressure in the momentum equation. As the thickness of the air-water interface is higher, the damping effects are stronger (Veic, 2018). Fig. 2 shows how the thickness of the air-water interface affects the magnitude of the hydrodynamic force and the peak impact pressure.

![Fig. 2 Effect of the thickness of air-water interface on the impact pressure and the impact force](image)

If the thickness of the air-water interface tends to zero, the impact pressure stabilizes. The computed peak pressure is almost 10 times higher when the air-water interface thickness is $\approx0.02R$, compare to the thickness of the air-water interface equal to $0.3R$. The effect of the
thickness of the air-water interface on the magnitude of the impact force is less pronounced. The computed peak force for the air-water interface thickness $0.3R$ is only 1.5 times lower than for the thickness of air-water interface equal to $0.02R$. The difference between the magnitude of the force for the thickness of air-water interface $0.04R$ and $0.02R$ is usually less than 5%. In order to provide an adequate accuracy with the acceptable computation cost, the calculations were conducted for the thickness of air-water interface equal to $0.04R$. This is achieved by applying the size of the computational cells in the zone of the wave impact $dx=dy=dz=0.006D$ (Fig. 3).

2.3 Results

The vertical impact load distribution for selected cases of the breaking wave impact, characterized by the different steepness of a wave front are presented. The selected cases refer to the plunging breaking with breaking location slightly before the structure, so that the overturning wave jet hits the monopile just below the wave crest level. This is usually identified as the most violent breaking wave stage. The range of the length of the overturning wave jet analysed in this study is between $l=0.2R-0.5R$ (Fig. 4). The breaking wave celerity, $c_b$, is identified as the horizontal water particle velocity at the toe of the overturning wave jet, while the curling factor, $\lambda$, is estimated as the distance from the toe of the overturning wave jet to the wave crest height (Fig 4). The monopile diameter is $D=7.2m$.

In order to obtain the desired breaking wave shapes, different wave generation techniques are applied including the propagation and transformation of irregular wave over the flat seabed (case 1) and over the sloped seabed $m=1:20$ (case 2), generation of phase-focused breaking wave (case 3), and propagation of monochromatic waves over a sloped seabed $m=1:10,1:20,1:50,1:100$ (cases 4.1-4.4 and cases 5.1-5.3). The main geometric wave characteristics for the different cases are presented in Table 1. The case 1 and case 2 describe the irregular breaking wave generated by the OceanWave3D model which corresponds to time series of the wavemaker used in laboratory experiments. For case 1, wave breaks over the flat seabed at $d_b=30m$, while for case 2, wave breaks over the sloping seabed ($m=1:20$) at $d_b=17m$. The case 3 describes the phase-focused breaking wave, which breaks over the flat seabed at $d_b=30m$. For the case of monochromatic waves, the depth at the toe of the sloping seabed is $d_t=45m$, while the depth at the tip of the sloping seabed is $d_t=27m$. The offshore wave height corresponds to the maximum wave height obtained from experiments presented in section 4, $H_o=H_{max}=18.5m$. The chosen offshore wave period for the cases 4.1-4.4 is $T_o=17s$, while for the cases 5.1-5.3 is $T_o=23.5s$. Fig. 5 shows the comparisons of the parameters of breaking waves used in the present study and breaking wave criterion applied in
DNV (2014). The test case 3 refers to deep water waves, while the other cases refer to intermediate water depths.

Table 1 Characteristic of the analyzed breaking waves

<table>
<thead>
<tr>
<th>No</th>
<th>Name</th>
<th>Method</th>
<th>(d_b) [m]</th>
<th>(\eta_b) [m]</th>
<th>(H_b) [m]</th>
<th>(L_0) m</th>
<th>(T_b) [s]</th>
<th>(L_{b'}) [m]</th>
<th>(H_0) [m]</th>
<th>(T_0) [s]</th>
<th>(L_0) [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Case1</td>
<td>Irreg. flat</td>
<td>30.0</td>
<td>12.4</td>
<td>16.0</td>
<td>250</td>
<td>14.8</td>
<td>23</td>
<td>/</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>2</td>
<td>Case2</td>
<td>Irreg. (m=1/20)</td>
<td>17.1</td>
<td>9.9</td>
<td>15.7</td>
<td>329</td>
<td>14.8</td>
<td>4</td>
<td>/</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>3</td>
<td>Case3</td>
<td>Phase focused</td>
<td>30.0</td>
<td>8.1</td>
<td>11.0</td>
<td>111</td>
<td>7.4</td>
<td>5</td>
<td>/</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>4</td>
<td>Case4.1</td>
<td>Reg. (m=1/10)</td>
<td>27.0</td>
<td>14.9</td>
<td>19.9</td>
<td>277</td>
<td>17.0</td>
<td>13</td>
<td>18.5</td>
<td>17.0</td>
<td>322</td>
</tr>
<tr>
<td>5</td>
<td>Case4.2</td>
<td>Reg. (m=1/25)</td>
<td>28.6</td>
<td>16.1</td>
<td>21.0</td>
<td>302</td>
<td>17.0</td>
<td>20</td>
<td>18.5</td>
<td>17.0</td>
<td>322</td>
</tr>
<tr>
<td>6</td>
<td>Case4.3</td>
<td>Reg. (m=1/50)</td>
<td>29.3</td>
<td>16.7</td>
<td>21.2</td>
<td>320</td>
<td>17.0</td>
<td>25</td>
<td>18.5</td>
<td>17.0</td>
<td>322</td>
</tr>
<tr>
<td>7</td>
<td>Case4.4</td>
<td>Reg. (m=1/100)</td>
<td>31.3</td>
<td>17.0</td>
<td>22.0</td>
<td>329</td>
<td>17.0</td>
<td>34</td>
<td>18.5</td>
<td>17.0</td>
<td>322</td>
</tr>
<tr>
<td>8</td>
<td>Case5.1</td>
<td>Reg. (m=1/10)</td>
<td>27.0</td>
<td>17.6</td>
<td>23.0</td>
<td>302</td>
<td>21.1</td>
<td>14</td>
<td>18.5</td>
<td>23.5</td>
<td>495</td>
</tr>
<tr>
<td>9</td>
<td>Case5.2</td>
<td>Reg. (m=1/25)</td>
<td>27.0</td>
<td>18.9</td>
<td>23.4</td>
<td>329</td>
<td>21.1</td>
<td>19</td>
<td>18.5</td>
<td>23.5</td>
<td>495</td>
</tr>
<tr>
<td>10</td>
<td>Case5.3</td>
<td>Reg. (m=1/50)</td>
<td>30.2</td>
<td>20.0</td>
<td>24.5</td>
<td>392</td>
<td>21.1</td>
<td>31</td>
<td>18.5</td>
<td>23.5</td>
<td>495</td>
</tr>
</tbody>
</table>

Fig. 5 Parameters of breaking waves and breaking wave criterion applied in DNV(2014)

Fig. 6 The shape of the wave fronts scaled to match identical wave crest height

The steepness of the breaking wave front is different in each analysed case. In order to show the differences between the profiles of the breaking wave front, all selected cases are scaled to match the identical crest height (Fig. 6). The range of the wave crest front steepness parameter \(s=\eta_b/L_{b'}\) (Bonmarin, 1989) is presented in Fig. 7. For the case of monochromatic waves, the parameter of the breaking wave crest front steepness is higher when wave breaks over the steeper slopes. For an identical slope, the longer offshore waves leads to higher
steepness of the breaking wave front. The highest values of the crest front steepness parameter are achieved for the case of phase-focused breaking wave and irregular breaking wave over a sloping seabed. Table 2 shows values of the estimated breaking wave celerity, the curling factor, and the length of the overturning wave jet for all selected cases.

![Table 2 Estimated values of the breaking wave parameters l, c_b, and λ.](image)

<table>
<thead>
<tr>
<th>No</th>
<th>Name</th>
<th>Method</th>
<th>c_b [m/s]</th>
<th>l/R</th>
<th>λ</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Case1</td>
<td>Irreg. flat</td>
<td>17.4</td>
<td>0.19</td>
<td>0.13</td>
</tr>
<tr>
<td>2</td>
<td>Case2</td>
<td>Irreg. m=1/20</td>
<td>16.8</td>
<td>0.26</td>
<td>0.55</td>
</tr>
<tr>
<td>3</td>
<td>Case3</td>
<td>Phase focused</td>
<td>14.4</td>
<td>0.20</td>
<td>0.45</td>
</tr>
<tr>
<td>4</td>
<td>Case4.1</td>
<td>Reg. m=1/10</td>
<td>20.1</td>
<td>0.31</td>
<td>0.39</td>
</tr>
<tr>
<td>5</td>
<td>Case4.2</td>
<td>Reg. m=1/25</td>
<td>20.5</td>
<td>0.25</td>
<td>0.19</td>
</tr>
<tr>
<td>6</td>
<td>Case4.3</td>
<td>Reg. m=1/50</td>
<td>20.8</td>
<td>0.28</td>
<td>0.15</td>
</tr>
<tr>
<td>7</td>
<td>Case4.4</td>
<td>Reg. m=1/100</td>
<td>21.2</td>
<td>0.19</td>
<td>0.11</td>
</tr>
<tr>
<td>8</td>
<td>Case5.1</td>
<td>Reg. m=1/100</td>
<td>20.8</td>
<td>0.44</td>
<td>0.46</td>
</tr>
<tr>
<td>9</td>
<td>Case5.2</td>
<td>Reg. m=1/25</td>
<td>22.1</td>
<td>0.25</td>
<td>0.27</td>
</tr>
<tr>
<td>10</td>
<td>Case5.3</td>
<td>Reg. m=1/50</td>
<td>22.8</td>
<td>0.23</td>
<td>0.18</td>
</tr>
</tbody>
</table>

2.3.1 Non-impact force

Fig. 8 presents the comparison between the computed hydrodynamic force and hydrodynamic force calculated by applying the Morison’s equation. For the estimation of the fluid velocity and acceleration, two approaches are used. In the first approach, the wave kinematics are estimated according to the stream function wave theory, which describes the non-linear symmetric waves up to the limit of \( H = 0.9H_b \). In the second approach, the breaking wave is simulated by applying 2D NS-VOF model. The wave velocity and acceleration field from the numerical model are taken at the position before wave reaches the breaking wave limit. For the breaking wave cases, which are characterized by the low steepness of the breaking wave front, the Morison’s force derived by applying the stream function wave kinematics, provides relatively good approximation of the non-impact force (case 1). As the steepness of the breaking wave front increases, the Morison’s force derived by applying the stream function wave kinematics is not appropriate for approximation of the non-impact force.
force. However, the Morison’s force based on the wave kinematics derived by applying NS-VOF model approximates the computed non-impact force fairly well.

![Comparison between the computed inline force and the Morison’s force](image)

**Fig. 8** Comparison between the computed inline force and the Morison’s force

### 2.3.2 Impact force

This section presents comparison between the impact forces obtained by applying the derived model and results of simplified approaches applied in engineering practice. Numerical results for the case 4.4 show that the presence of the wave run-up considerably interferes with the interaction between the overturning wave jet and a monopile structure. Fig. 9 shows that the impact force on the structure is damped and, as a consequence, reduced due to the interaction between the overturning wave jet and the wave run-up jet. The results of the case 4.4 are not adequate for a direct comparison with the results from the derived model, hence, the case 4.4 is omitted in the figure.

![Interference between the wave run-up and overturning wave jet for case 4.4](image)

**Fig. 9** Interference between the wave run-up and overturning wave jet for case 4.4

It is estimated that the impact force on a monopile structure occurs when dynamic pressure on the structure surface exceeds \( p_{\text{dyn}} > 0.5 \rho c_b^2 \). Fig. 10 presents plots of the impact forces \( F_i \) for all selected cases. The force is normalized by \( F_i(t)/\rho R c_b^2 \eta_b \) and is shown in terms of the slamming coefficient \( C_{sr}(t) \). The values of the peak slamming coefficient are

33
scattered, ranging from $C_{sr}=0.9\pi$ (case 2) to $C_{sr}=2.1\pi$ (case 1). Fig. 10 also shows the comparison between the computed slamming coefficient $C_{sr}(t)$ and a corresponding slamming coefficient obtained by applying a simplified approach (Wienke, 2001). The peak of the slamming coefficient in Fig.10 corresponds to $t=0$ s. The time $t<0$ s refers to the rising impact force phase which cannot be derived from a simplified approach suggested by Wienke (2001), while the time $t>0$ s refers to the decaying impact force phase. For the time interval $0<t<0.12R/c_b$ the computed force decays much faster than forces derived from a Wienke approximation.

Fig. 10 Slamming coefficients derived from the present model and corresponding results obtained by applying Wienke (2001) approximation ---

Fig. 11 shows the value of the computed slamming coefficient $C_{sr}$ as a function of the crest front steepness parameter $s_f$. The results show that the value of the slamming coefficient $C_{sr}$ is inversely proportional to the steepness of the breaking wave front $s_f$.

Fig. 12 shows the computed peak impact forces and corresponding results obtained by applying simplified approaches. The slamming coefficient is considered as $C_{sr}=2\pi$ for both simplified approaches. The discrepancies between presented results arise from simplification applied in approximate approaches.
Effect of Breaking Wave Shape on Impact Load on a Monopile Structure

2.3.3 Vertical load distribution

The distribution of the impact pressures on the monopile structure is similar for all analyzed cases. The highest impact pressures computed in this study are \( p_{\text{max}} \approx 25 \rho_c R^2 \), with the corresponding rising time \( t_{\text{rise}} \approx 0.025 R/c_b \). Those values are very similar to the laboratory measurements conducted by Zhou et al. (1991) and Chan et al. (1995). The highest impact pressure occurs in the region below the overturning wave jet. Fig. 13 shows the shape of the breaking wave profile and corresponding pressure distribution on a structure for the case 4.1. Additionally, Fig. 13 shows vertical impact pressure distribution along the front line of the monopile structure. It can be observed that the highest impact pressure occurs in the region where the overturning wave jet meets the wave run-up on the structure. The vertical pressure distribution on the front is also characterized by a rapid decay of pressure from the peak value.

By integrating the pressure around the strips of the monopile structure, the vertical load distribution can be determined. The monopile structure is divided into small strips \( dz = 0.09 \text{m} \) and the strip forces are calculated. The obtained vertical impact load distribution is normalized by \( \rho R c_b^2 dz \). Fig. 14 shows the obtained slamming coefficients \( C_s \) at the moment of the maximum impact force for the case 1 and the case 2. The results show that the peak slamming coefficients occur in the zone of the highest pressure. The value of the peak slamming coefficient approaches \( 2\pi \) for all analysed cases. The slamming coefficient values decay rapidly as we move away from the peak region. The area of the impact load on the structure is significantly higher than the impact area which is defined by the curling factor \( \lambda \) (Table 2).
As mentioned, the approximation of the vertical impact load distribution by the rectangular shape leads to the non-unique and confusing determination of the curling factor and the slamming coefficient. This is presented in Fig. 14, where for a geometrically determined curling factor $\lambda$, the slamming coefficient for the case 1 is $C_s=1.8\pi$, while for the case 2 is $C_s=1\pi$.

Fig. 15 shows vertical impact load distribution for all cases. The slamming coefficient is presented in terms of normalized vertical position, $(z-z_{pmax})/\eta_b$, where $z_{pmax}$ is the vertical location of the maximum impact pressure. For $z>z_{pmax}$ the impact load distribution can be approximated by a linear function - the maximum value occurs at $z=z_{pmax}$ and zero value is located at $z=\eta_b$. For $z<z_{pmax}$ the impact load is characterized by the rapid decay from the peak value $C_s=2\pi$. Fig 16 presents the temporal distribution of impact force on the monopile strip located at $z=z_{pmax}$. The rising phase of the impact force ($t<0s$) can be approximated by a linear function. Then, the impact force decays rapidly from the peak as clearly show the plots in Fig. 16.

2.3.4 Impact force

The computed impact forces are compared according to suggested impact load distribution for different moments of the impact $C_s(z,t)$, and presented in Fig 17. The vertical distribution of the impact load $C_s(z,t)$ is shown for 7 different moments in time $t_0$ to $t_6$. The diagram of the impact load distribution is divided in two parts. For the zone $z>z_{pmax}$ the impact begins at $t_0 = -0.9l/c_b$, while for the zone $z<z_{pmax}$ the impact begins at $t_0 = 0.06R/c_b$. For the calculation of the impact force according to the load distribution suggested in the Fig 17, the
location of the maximum impact pressure $z_{pmax}$ and the length of the overturning wave jet $l$ is required. These parameters can be obtained from a 2D simulation of breaking wave by applying NS-VOF model. In this analysis it is approximated that the impact pressure occurs approximately at $z_{pmax}=1.15z_{toe,jet}$.

Fig. 17 Suggested impact load distribution in time $C_s(t,z)$

Fig. 18 shows comparison between the computed and estimated impact force according the aforementioned procedure. The temporal characteristic of the impact force is captured with good accuracy. Discrepancies observed in the impact force peak zone are up to 30%. However, compared to the results obtained from Wienke's simplified approach presented in Fig. 10, the results from suggested procedure provide significantly better approximation of the impact force. These results encourage us for further development of the proposed procedure which may eventually lead to an alternative method for preliminary estimation of the impact forces.

Fig. 18 Comparison between the computed impact forces and approximate solution

3. Experimental validation case

This section presents comparison between the results of the presented numerical model and experimental data obtained from laboratory experiments conducted in Deltares, Delft. The 50-years storm conditions in the German Bight were simulated in a wave basin. The parameters of the applied JONSWAP wave energy spectrum were: $H_s=10m$, $T_p=13s$, $\gamma=3.3$. The diameter of the monopile is $D=7.2m$. The model scale is 45.
Laboratory experiments were conducted for two scenarios. In the first series of experiments the monopile structure was installed on the flat seabed, \( d=30 \) m. In the second series of experiments the monopile structure was installed at the sand bar at the depth \( d=25 \) m. The slope of the sand bar is \( m=1:21 \). The validation of the derived model focus on measurements for which the maximum hydrodynamic force on the structure is recorded. The validation case 1 relates to the breaking wave over the flat seabed, while validation case 2 relates to the breaking wave over the sand bar.

The wave basin has a length of 75 m, a width of 8.7 m, and a maximum water depth of about 1.2 m. Hydrodynamic force and impact pressures were measured during tests and the wave-structure interaction were recorded with a high-frequency video camera. To obtain actual hydrodynamic loads, and remove the effects of structural vibrations the measurements were filtered out (Veic, 2018). The structure was equipped with 10 pressure transducers distributed evenly along the front line of the structure, every 22 mm in the wave impact zone. A sampling frequency of 1000 Hz was applied. Waves were generated by a piston-type wavemaker. For each scenario 5 separate tests were conducted. The duration of each test case was 1744 s (~3 h in full scale) which corresponds to approximately 1000 waves. For the case of the experiments with the sand bar, the breaking wave phenomena occurred more frequently. Taking into account the total number of waves (5000), probability of occurrence of the slamming event in the case of the flat seabed was 0.3%, while in the case of the sand bar was 1.2%. The maximum inline force measured for the case of the sand bar was more than 50% higher than for the case of the flat seabed.

![Fig. 19](image1.png) **Fig. 19** The comparison between predicted and measured wave elevations

![Fig. 20](image2.png) **Fig. 20** The comparison between predicted and measured hydrodynamic forces
The experimental results were reproduced by implementing the kinematic wavemaker boundary condition in the numerical model. The obtained numerical results were compared with experimental measurements of wave elevation, hydrodynamic force, and the pressures along the front line of the monopile. The validation was also conducted by applying the records from the HF-video camera available for the breaking wave over the flat seabed. The comparison between the numerical and experimental wave elevation are presented in Fig. 19. The wave elevation was measured at a distance of 3D from the monopile centre line position. Fig. 20 shows that the computed forces are in reasonable agreement with experimental data. Small discrepancies are observed only for the peak value of the impact force in the case 2. These differences may be attributed to the relatively low sampling rate used in measurements, or effects of the air compressibility, which is not taken into account in conducted computations.

![Fig. 20 Snapshot of the breaking wave impact](image)

The snapshot from the HF-video camera for the case 1 is very similar to the numerical visualization (Fig. 21). The numerical results for the case 1 show that wave breaks just in front of the structure so the wave run-up reduces the magnitude of impact force. Fig. 22 shows the magnitude of impact force for the structure moved 1.25R upstream. The computed impact force in this case is 4 time higher. This shows that the problem of the position of a structure is especially important for irregular wave attack on a structure where a wave breaking position is difficult to predict beforehand.

![Fig. 21 Computed impact force for the translated monopile structure](image)

Fig. 23 and Fig. 24 show comparison between the computed and measured impact pressures at the front line of the monopile structure. For both cases the numerical results are in excellent agreement with the experimental results. The discrepancies between the numerical and experimental results for case 2 occur basically only in the peak pressure zone. The observed discrepancies may be attributed mainly to the relatively low sampling rate of pressure measurements (1kHz) which is not sufficient to accurately record pressure in a rising phase. Nevertheless, at the moment of the maximum impact force, which occurs after the
moment of the peak pressure occurrence at the front line of the monopile (Fig. 24), the values of computed and measured pressures are similar. The results from this section show that the presented numerical model can reproduce experimental results with sufficient accuracy.

Fig. 23 Comparison of the impact press.-flat seabed

Fig. 24 Comparison of the impact press.-sand bar

4. Conclusions

The numerical model is derived to study the effect of the shape of breaking waves on wave impact loads on a monopile structure. The derived model combines potential flow model with a NS VOF solution. The investigations focus on the computation of the vertical impact load distribution arising from the breaking waves of different wave front steepness.

The results show that the highest impact pressure occurs in the region below the overturning wave crest, where the overturning wave crest meets the wave run-up on the structure. The analysis shows that at this location the breaking wave impact is maximum and the slamming coefficient is about $C_s=2\pi$. Away from the peak region, wave impact loads decays rapidly. The area of the impact load on the structure is significantly higher than the impact area defined and applied in engineering practice.

The analysis shows that the approximation of the vertical load distribution by a rectangular is a simplification which cannot uniquely approximate real load distribution arising from breaking wave attack on a monopile structure.

The derived results indicate a possibility for development of improved design methodology for a preliminary estimation of the breaking wave loads on monopile structures. The investigations show that the non-impact wave loads may be derived from the Morison equation, while the breaking wave impact forces may be approximated according to the proposed diagram of the temporal vertical load distribution, $C_s(t,z)$.

The derived results are compared with data from laboratory experiments on irregular breaking wave loads. The geometric characteristics of breaking waves obtained in laboratory experiments are successfully reproduced in the numerical domain. Breaking wave forces are computed with sufficient accuracy. Differences between measurements and computations are found only in the very narrow region of the peak values, where computed values are up to
10% higher. These differences may be attributed to the relatively low sampling rate used in measurements, or effects of the air compressibility.

Air pockets, which can be trapped between the overturning wave jet and the structure during the wave impact are compressible and can pass through expansion and contraction phases. This results in the pulsating impact pressure during the wave impact. Influence of the pulsating impact pressure on the total impact loads is expected to be more pronounced for the modal scale than for the prototype scale. As the effects of the air compressibility on the braking wave impact loads may be significant, it is recommended to perform future analysis by applying a compressible numerical model.

ACKNOWLEDGMENTS

A part of this work was carried out during MareWint project under research area FP7-PEOPLE-2012-ITN Marie-Curie Action: “+Initial Training Networks” and during secondment at the Dutch research institute, Deltares, Delft. The authors are indebted to Bo Terp Paulsen, Ph.D. from Deltares for constructive comments.

REFERENCES


41


[27] DNV. (2010). Environmental conditions and environmental loads, DNV-RPC205, DNV GL AS


Submitted: 11.07.2018. Duje Veic dujeveic@gmail.com
Accepted: 04.06.2019.
Institute of Hydro-engineering, Polish Academy of Sciences ul. Kościerska 7, 80-328 Gdańsk, POLSKA
Wojciech Sulisz sulisz@ibwpan.gda.pl
Institute of Hydro-engineering, Polish Academy of Sciences ul. Kościerska 7, 80-328 Gdańsk, POLSKA
Rohan Soman
Institute of Fluid Flow Machinery, Polish Academy of Sciences 14, Fiszera Street Gdańsk 80-231, Poland

42
SIMPLIFIED AND ADVANCED APPROACHES FOR EVACUATION ANALYSIS OF PASSENGER SHIPS IN THE EARLY STAGE OF DESIGN

UDC 629.541.4:629.5.067
Professional paper

Summary

In order to improve the survival capability of passenger and ro-ro ships in event of fire or flooding, in the last few years a set of international Regulations has been issued. In particular, the Regulation SOLAS “Safe Return to Port” is addressed towards design criteria able to guarantee adequate functionality of the ship when a casualty occurs. The Regulation requires the evacuation of the ship when a given threshold of damage (i.e., fire and flood) is exceeded. The evacuation analysis has become of primary importance even in the early-stage design. Indeed, the new amendments to SOLAS Regulation II-2/13.3.2.7 makes the evacuation analysis mandatory for both new and existing passenger and ro-ro ships since the early steps of the project. In this paper, the current IMO Guidelines have been analysed, and a case study for the evacuation of a 3600-person cruise ship has been carried out by means of both a simplified and an advanced method. To perform advanced calculations, two different software, based on virtual reality, have been used and the results have been compared with simplified method ones.

Keywords: Evacuation analysis; Ship Safety; Safe Return to Port; Passenger ship

1. Introduction

In the last few years, evacuation analysis of passenger ships has become one of the most important theme in shipbuilding. The International Maritime Organization (IMO) has decided to regulate this issue via the International Convention for the Safety of Life at Sea (SOLAS). From 1999 to 2007 the IMO Maritime Safety Committee (MSC) issued three Circulars [1, 2 and 3] about the evacuation analysis and the procedures for calculating the evacuation times for a passenger ship. Based on these Circulars, several studies have been carried out [4, 5 and 6]. Other studies, instead, have been oriented to evacuation time calculations by means of IMO certified software [7, 8, 9, 10, 11, 12, 13, 14 and 15]. Starting from data ensuing from a
number of evacuation simulations [16 and 17], in June 2016 IMO issued a new Circular [18] in order to determine the evacuation time and, in case, the presence of congestion points.

Since this could have a huge impact on both the General Arrangements (GA) and the escape routes of the ship, it is mandatory to perform the evacuation analysis immediately after a first GA is at disposal. That means calculations should be performed in the first stages of preliminary design of a vessel.

In this paper, a comparison between the methods to calculate the evacuation time in accordance with the last IMO Circular [18] is presented, along with a test case previously performed by the authors [13 and 19]. The evacuation analysis has been carried out through both simplified and advanced method. In particular, for the advanced method, the two certificated software EVI and AENEAS have been used. In order to properly compare the two methods, a simple scenario that provides a common passenger distribution has been chosen, i.e., the Case 1 (primary evacuation case, night) defined by both the cited Circulars.

2. Overview of the Regulation framework

2.1 Superseded Regulations

In the last few years, IMO issued some Regulations regarding the evacuation analysis:

- May, 1999 - MSC/Circ.909 "Interim guidelines for a simplified evacuation analysis on ro-ro passenger ships" [1]. With this Circular the evacuation analysis became mandatory for ro-ro ships only. To calculate the evacuation time, a unique method (called "simplified method") based on a hydraulic or flow model was required.

- June, 2001 - MSC/Circ.1001 “Interim guidelines for a simplified evacuation analysis of high-speed passenger crafts”. With this Circular the simplified evacuation analysis was expanded also to high-speed passenger crafts.

- June, 2002 - MSC.1/Circ.1033 - "Interim guidelines for a simplified evacuation analysis for new and existing passenger ships" [2]. For the first time the evacuation analysis was applied to all passenger ships. To calculate the evacuation time, two distinct methods were allowed: the "simplified method", which was already present in the previous Circulars, and a so-called "advanced method" based on experimental data. The first applications of the advanced method were criticized due to the lack of significant experimental data. In fact, there were little or no data relating to passenger response times in maritime environment, so that data used in the guidelines were derived from land-based building measurements.

- October, 2007 - MSC.1/Circ.1238 "Guidelines for evacuation analysis for new and existing passenger ships" [3]. To overcome the limits of the previous Regulation, the findings of a particular project (called Fire Exit) funded by the European Union were adopted in this Circular. In such a project, sea trials concerning the response times during the evacuation of a Grimaldi ro-ro ferry sailing between Civitavecchia and Barcelona in April 2005 were performed. The sea trials clearly demonstrated that the response times calculated by the MSC/Circ.1033 were not sufficiently accurate. Therefore, the Fire Exit sea trial data were submitted to IMO and incorporated within the new analysis protocol reported in the MSC/Circ.1238. Also in such a Circular both a simplified and an advanced method was proposed.

- June, 2015 - MSC/Circ.1166 "Guidelines for a simplified evacuation analysis for high-speed passenger crafts". This Circular superseded the previous interim MSC/Circ.1001.
2.2 Regulation in force

On June 6, 2016 - MSC.1/Circ. 1533 "Revised guidelines on the evacuation analysis for new and existing passenger ships" has been issued and all the previous Regulations on the matter were superseded. Through this Circular, evacuation analysis is made mandatory not only for ro-ro passenger ships, but also for other passenger ships built on or after January 1, 2020. The Regulation makes mandatory the determination of the evacuation time and encourages to perform such an analysis also on existing passenger ships. In this manner it is possible to identify, in case, the presence of congestion points or critical areas where to adopt proper operational measures in order to not exceed the maximum allowable evacuation time. As the previous Regulations, also this one allows to evaluate the evacuation time by either the simplified or the advanced method. The simplified method is based on a "fluid-dynamic similarity", in which corridors and stairs are the tubes, while the passengers are the fluid that runs through them; for the analysis it is necessary to follow a specific procedure as indicated in the Circular. In the advanced method, instead, passengers are represented as individuals with particular features, and the calculation of the evacuation time is carried out by a software based on virtual reality. In both methods, the main objective is the evaluation of the total evacuation duration, as well as the identification of congestion points and areas where intense counter or cross flows are present.

The main innovations introduced by the MSC.1/Circ.1533 are:

- explicit recommendations to perform the evacuation analysis already during the early-stage design;
- the harmonization of the evacuation time calculation carried out by the simplified or the advanced method;
- documentation of the crossing, counter or dividing flow areas, in order to allow the crew to take adequate measures in tackling these critical areas;
- the introduction of a voluntary additional day case that foresees the crowding in large open deck spaces;
- the introduction of a voluntary additional case concerning the embarkation in order to prove that such an operation can be achieved within 30 minutes.

3. Simplified method

By the simplified method, the total evacuation duration is calculated and compared with an allowable time \( n \), which depends on the vessel type (ro-ro or other passenger ship), and the number of Main Vertical Zones (MVZs). This represents the time necessary for all the persons on board (passengers and crew) to reach the pertinent assembly stations, embark on the survival crafts and launch them. To tackle the evacuation analysis, the following components must be calculated:

- **Response duration** \((R)\), which considers for each person the effective reaction time to an emergency situation. This duration begins upon initial notification of the emergency (usually an alarm) and ends when the passenger realizes the situation and starts to move toward the assembly station.

- **Total travel duration** \((T)\), which includes the time spent to move all the persons on board from the place occupied upon the emergency notification to the assembly stations.

- **Embarkation and launching duration** \((E+L)\), which represents the time required for the ship abandonment of all the persons present on board. The evaluation of \( E+L \) must be done with reference to full-scale trials, simulations or data provided by the manufacturers of the evacuation systems. In any case \( E+L \) must not exceed 30 min.
About the initial distribution of people on board, the MSC.1/Circ.1533 is based on the Ch.13 of the International Code for Fire Safety Systems (FSS Code), distinguishing between "night scenario" and "day scenario" as follows:

- **"night scenario"** – passengers in cabins with maximum berthing capacity fully occupied; 2/3 of crew members in their cabins and the remaining 1/3 so distributed:
  - 50% located in the service spaces;
  - 25% located at their emergency stations;
  - 25% initially located at the assembly stations; successively they should proceed towards the most distant passenger cabin assigned to that assembly station, in counter flow with the evacuees. Once this passenger cabin is reached, these crew members are no longer considered in the simulation.

- **"day scenario"** – passengers in public space occupy 3/4 of the maximum capacity; 1/3 of the crew members in their accommodation spaces (cabins and crew day spaces); 1/3 of the crew members in the public spaces and the remaining 1/3 distributed as follows:
  - 50% located in the service spaces;
  - 25% located at their emergency duty locations;
  - 25% initially located at the assembly stations; successively, they should proceed towards the most distant passenger cabin assigned to that assembly station, in counter flow with the evacuees. Once this passenger cabin is reached, these crew members are no longer considered in the simulation.

The Circular imposes to consider in the evacuation analysis at least four scenarios (Cases 1 to 4), which are characterized by different distributions and features of persons (both passengers and crew):

- **Case 1 (primary evacuation case, night)** – persons distributed as in "night scenario"; the entire ship is considered; evacuation is carried out along the main means of escape.

- **Case 2 (primary evacuation case, day)** – persons distributed as in "day scenario"; evacuation scenario as per Case 1.

- **Case 3 (secondary evacuation case, night)** – persons distributed as in "night scenario"; only the MVZ generating the longest assembly duration is further investigated according with one of the following alternatives (but Alternative 1 is recommended):
  - Alternative 1: a main escape route previously used in the considered MVZ is unavailable for the simulation;
  - Alternative 2: 50% of persons of the most populated adjacent MVZ is forced to move into the considered MVZ.

- **Case 4 (secondary evacuation case, day)** – persons distributed as in "day scenario"; evacuation scenario as per Case 3.

One of the innovations of the Circular consists in the introduction of two new optional scenarios, which may be considered if appropriate:

- **Case 5 (open deck, day)** – if there is an open deck outfitted for use by passengers, having a gross surface area larger than 400 m² or with more than 200 persons accommodated, the open deck is considered as an additional public space with an initial density of 0.5 person/m², and so a new person distribution for Case 2 must be simulated.

- **Case 6 (embarkation)** – if embarkation and assembly stations are not coincident, the travel duration from assembly station to the entry point of the survival crafts should be considered in the $E+L$ evaluation. All persons, which the ship is certified to
carry, are distributed in the assembly stations according to their capacity. Possible 
congestions directly in front of the entry point of the survival crafts should be 
considered in the simulation.

An innovation introduced by the MSC.1/Circ.1533 is the harmonization of the total 
evacuation duration ($t_{TOT}$) for both the simplified and the advanced method (Fig.1). Indeed, 
the performance standard to be verified for the two methods is the same:

$$t_{TOT} = 1.25(R+T) + \frac{2}{3}(E + L) \leq n$$

![Fig. 1 Representation of performance standard for simplified and advanced method.](image)

The reference duration $n$ must be equal to 60 minutes both for ro-ro vessels and for 
other passenger ships with less than three MVZs, and to 80 minutes for passenger ships with 
more than three MVZs.

About the response duration, the Circular imposes to consider $R$ equal to 10 minutes for 
the night time scenarios (Cases 1 and 3) and to 5 minutes for day time scenarios (Cases 2, 4 
and 5).

In the simplified method, the travel duration $T$ is calculated as a function of the 
passenger and crew flow. In the Circular, a procedure to establish the travel duration $T$ is 
described, as a function of the highest travel duration in ideal condition denoted by $t_I$. For 
each Case analysed, with reference to a deck, $t_I$ is evaluated through the following sum:

$$t_I = t_F + t_{deck} + t_{stair} + t_{assembly}$$

where:

- $t_F$ represents the flow duration. It depends on the specific flow of persons ($F_S$), 
  which is the number of persons passing a point of the escape route per unit time and 
  unit clear width. The specific flow $F_S$ is given from tables of the Circular as a 
  function of both the type of facility (corridors, stairs down or up, doorways) and the 
  initial density of persons ($D$). It is so possible to predict the number of persons 
  passing a particular point of an escape route per unit time, and determine the 
  calculated flow of person ($F_C$):

$$F_C = F_S W_C$$  

$$F_C = F_S W_C$$

47
in which \( W_C \) is the clear width of the different means of escape. The flow duration is the total time needed for \( N \) persons (passengers and crew) to move past a point in the egress system and it is given by the following formula:

\[
T_F = \frac{N}{F_C}
\]  

\[ (4) \]

- \( t_{\text{deck}} \) is the deck travel duration. It represents the time necessary to move from the farthest point of the escape route of a deck to the stairway.
- \( t_{\text{stair}} \) is the stairway travel duration. It is the time required to travel the stairway.
- \( t_{\text{assembly}} \) is the assembly travel duration. It represents the time necessary to move from the end of the stairway to the entrance of the assigned assembly station.

In order to exceed the limits of the hydraulic similarity, two corrective factors (\( \gamma \) and \( \delta \)) are introduced, so that the travel duration \( T \) of a specific Case is given by:

\[
T = (\gamma + \delta)T_f
\]

\[ (5) \]

where \( \gamma \) must be taken equal to 2 for Case 1 and 2, and to 1.3 for Case 3 and 4, while \( \delta \) (the counter-flow correction factor) must be taken equal to 0.3 for all Cases.

The performance standards should be carried out through the formula (1) for all the Cases analysed. After verifying that the allowable duration \( n \) imposed by the Circular is respected, also possible congestion points should be identified. These points are:

- the spaces where the initial density is higher than 3.5 person/m\(^2\);
- the locations where the difference between the inlet and outlet calculated flow (\( F_C \)) is more than 1.5 person per second.

Aim of the evacuation analysis is to identify and eliminate, as far as practicable, the congestion points because anyway they represent a potential danger during the evacuation and abandonment of the ship. The identification of these points is quite difficult through the simplified method. Therefore, also for this reason, the IMO Maritime Safety Committee introduced the advanced method since the MSC.1/Circ.1238 issued on October, 2007.

4. Advanced method

In the advanced method the hydraulic similarity of the simplified method is exceeded since all persons on board are represented as individuals with particular abilities and response durations; moreover, the relationship between passengers, crew and layout of the ship is considered.

In this method, the evacuation analysis is performed with a specific software based on virtual reality, which uses different pedestrian algorithms for simulating the behaviour of evacuees. Through a randomized approach, the travel duration \( T \) is calculated: for each Case analysed a minimum of 500 different simulations should be done, but this number may be reduced to 50 if an appropriate convergence criterion is complied with. The simulations are made up considering 100 different randomly-generated populations, and for each population a set of parameters influencing the evacuation and collected in 4 different categories are fixed:

- category geometrical, which considers the layout of the escape route;
- category population, which considers a range of parameters concerning persons and population demography;
- category environmental, which considers the static and dynamic conditions of the ship (even if presently the figures of such parameters are not reliable due to the lack of experimental data). For this reason, in the last few years several experiments on
human behaviour in case of storm have been performed using particular platforms which simulate the ship motions [21, 22 and 23];
– category procedural, which considers the assistance of the crew in emergency situations.

Through this procedure, all scenarios are investigated and for each one a value of total assembly duration $t_A$ (defined as the maximum individual assembly duration) is determined. The travel duration $t_I$ to be associated at each Case corresponds at the 95-percentile of $t_A$ obtained from the various simulations. The total travel duration $T$ to be considered in the performance standard (1) is the maximum value of the travel durations $t_I$ drawn from the analyses of all Cases examined.

The virtual reality simulators use different pedestrian algorithms, which are based on peculiar mathematical models (statistical models, queuing models, route-choice models, gas-kinetic models, etc.). The main certified software are:
– EVI, developed by the University of Strathclyde-Glasgow;
– AENEAS, developed in collaboration between DNV-GL and TraffGo;
– EXODUS, developed by the University of Greenwich;
– ODIGO, developed by the French engineering company Principia;
– VELOS, developed by the National Technical University of Athens.

In this paper, AENEAS and EVI software are used to carry out the calculations required by the advanced method to determine the total travel duration $T$.

4.1 AENEAS

AENEAS, developed by DNV-GL and TraffGo, is a multi-agent software in which passengers and crew (generically, the so-called agents) are represented with particular attitudes, abilities and goals. The software is based on a square-cell discretization of the various decks (Fig. 2) with the cell side 0.4 m long. Each agent can occupy only one cell at each time step, and he moves from cell to cell.

The software is composed by three modules:
– AENEASed (editor), which allows to import a CAD file and to convert it into a relevant cellular grid. After the discretization it is possible to modify the decks geometries. Furthermore, it is necessary to define the escape routes and distribute the agents in the ship distinguishing between passengers and crew.
– AENEASsim (simulator), which allows to load the project created in the editor and to run the simulation. The demographic parameters of passengers and crew may be fixed either by default (according to the indication present in the MSC.1/Circ.1533) or properly established by the user (in accordance with the actual population present on board). The results of each Case are synthesized by the probability density function of the travel evacuation duration associated to each simulation and by the relevant significant evacuation time (assumed as the 95-percentile). Moreover, sketches of all the congestion points of the simulation having a travel evacuation duration equal to the determined significant evacuation time are given.
– AENEASview (viewer), which allows to view the evacuation process in a three-dimensional environment. Through this video, the user can easily identify the congestion points in the escape routes during the evacuation phases.

In order to represent the decks in a realistic manner, the following types of cell can be used (Fig. 2 and 3):
– Free cells (white coloured), which can be occupied by the agents during the evacuation;
– Wall cells (black coloured) for the obstacles (as walls, furniture, etc.) along the escape routes. These cells cannot be occupied by the agents;
– Goal cells (identified by a tag given by the user) for the objective (as an assembly station or a cabin) to be achieved by each agent;
– Door cells (red coloured) for the doors, where a reduction of the walking speed of the agents occurs;
– Step cells (cyan coloured) for stairways (both up- and down-stairs).

Fig. 2 Discretization of a cabin in AENEAS

During the simulation, each agent jumps from cell to cell, using the neighbouring free, door or step cell in order to reach its assigned goal cell. Each agent runs along an escape route, which may be specified either by the user or by default as the way towards the closest
goal cell. For the orientation, agents make use of a "potential value" associated to each cell: for a given route, the cells have a value that increases proportionally to the distance from the goal cells. The agent finds its way by comparing the potential value of the cell occupied with the value of the adjacent cells. During the evacuation, the generic agent can choose between 8 cells neighbouring (Fig. 4).

Standing on a certain cell \( i=0 \) (central cell), the occupation of an adjacent cell \( i \) is done on the basis of the greatest probability \( p_i \):

\[
p_i = e^{\frac{(P_0 - P_i) + S}{S}}
\]

being \( P_0 \) the value of the potential of the central cell \( i=0 \) and \( P_i \) that of a generic adjacent cell \( i \). Indeed, \( S \) is the value of the Sway that represents the accuracy with which an agent follows the course of the potential.

4.2 EVI

EVI, developed by the Ship Stability Research Centre of the University of Strathclyde-Glasgow and distributed by Safety at Sea, is another multi-agent software. It bases on the concept of “ evacuability” (indeed, EVI is the acronym of EVacuation Index) to be intended as the ability of a person to evacuate the ship. This index takes into account a wide range of parameters, which can be collected in two separate groups, namely the Initial Conditions and the Evacuation Dynamics. The first group considers the layout of the ship (\( env \)), the demography of the population and its initial distribution (\( d \)) and the response time to danger (\( r \)), while the second group takes into account the walking speed of each agent, the interaction between passengers and crew and between agent and layout of the ship.

The ship environment is set up starting from the 2D general arrangement drawings of decks in order to create a 3D model of the spaces involved in the evacuation (Fig.5 and Fig.6).
5. Case Study

An existing cruise ship (Fig. 7) has been considered to carry out the evacuation analysis in accordance with the MSC.1/Circ.1533.

In particular, both the simplified and the advanced method have been performed with reference to the MVZs 5 and 6 (Fig. 7), which represent the most critical Main Vertical Zones of the ship due to the presence of public areas.

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, overall</td>
<td>285.00 m</td>
</tr>
<tr>
<td>Breadth</td>
<td>32.20 m</td>
</tr>
<tr>
<td>Gross Tonnage</td>
<td>86300 GT</td>
</tr>
<tr>
<td>Decks</td>
<td>11</td>
</tr>
<tr>
<td>Speed, max</td>
<td>24.00 kn</td>
</tr>
<tr>
<td>Speed, cruise</td>
<td>22.00 kn</td>
</tr>
<tr>
<td>Passengers</td>
<td>2667</td>
</tr>
<tr>
<td>Crew</td>
<td>933</td>
</tr>
</tbody>
</table>

Fig. 5 General deck in EVI

Fig. 6 3D model in EVI

Fig. 7 Ship characteristics and the MVZs considered for the evacuation analysis in the case study
5.1 Simplified method application

In the simplified method, only the primary cases (i.e., Case 1 and Case 2) have been considered for the performance standard verification. Only the main escape routes have been taken into account to calculate the total evacuation duration. In the first step, this method verifies the layout of the considered MVZs. In particular, the length of the corridors, the width of the doors and of the stairways, and the area of the landings should comply with the requirements of Ch.13 of FSS Code. An example of the topological analysis, with reference to Deck 6, is reported in Table 1 and Figure 8. Successively, in accordance with the MSC.1/Circ.1533 the considered MVZs are populated, and the calculation procedure can start. The population distribution obtained for the case study is shown in Table 2. By means of the formulae reported in the Circular, the *stairway travel duration* $t_{\text{stairs}}$, the *deck travel duration* $t_{\text{deck}}$ and the *flow duration* $t_{\text{F}}$ have been calculated for each deck, while the *assembly travel duration* $t_{\text{assembly}}$ has been determined only for the deck where the assembly stations are located. The results for Case 1 are reported in Table 4 as example of the above described procedure. The sum of these values corresponds to the *highest travel duration in ideal condition* $t_{I}$ for each deck. The *travel duration* $T$ to be applied in the performance standard (1) is the maximum $t_{I}$ calculated, corrected by $\gamma$ and $\delta$. The results obtained for Case 1 and 2 are reported in Table 7: in both cases, the performance standard as indicated in the Circular (i.e., 80 minutes) is fully met.

![Fig. 8 Layout of Deck 6](image)

<table>
<thead>
<tr>
<th>Deck 6</th>
<th>Width W_{c} [m]</th>
<th>Length [m]</th>
<th>Area [m²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Corridor 1</td>
<td>0.90</td>
<td>36.90</td>
<td>37.40</td>
</tr>
<tr>
<td>Corridor 2</td>
<td>0.90</td>
<td>36.90</td>
<td>37.40</td>
</tr>
<tr>
<td>Door 1</td>
<td>0.90</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>Door 2</td>
<td>0.90</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>Stairway 7</td>
<td>5.20</td>
<td>3.63</td>
<td>/</td>
</tr>
</tbody>
</table>
Table 2 Population distribution (simplified method)

<table>
<thead>
<tr>
<th>Deck</th>
<th>Night (Case 1)</th>
<th>Day (Case 2)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Passengers</td>
<td>Crew</td>
</tr>
<tr>
<td>C</td>
<td>0</td>
<td>8</td>
</tr>
<tr>
<td>B</td>
<td>0</td>
<td>43</td>
</tr>
<tr>
<td>A</td>
<td>0</td>
<td>30</td>
</tr>
<tr>
<td>1</td>
<td>142</td>
<td>2</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>4</td>
</tr>
<tr>
<td>3</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>4</td>
<td>60</td>
<td>2</td>
</tr>
<tr>
<td>5</td>
<td>89</td>
<td>2</td>
</tr>
<tr>
<td>6</td>
<td>64</td>
<td>2</td>
</tr>
<tr>
<td>7</td>
<td>48</td>
<td>2</td>
</tr>
<tr>
<td>8</td>
<td>78</td>
<td>2</td>
</tr>
<tr>
<td>9</td>
<td>0</td>
<td>7</td>
</tr>
<tr>
<td>10</td>
<td>47</td>
<td>1</td>
</tr>
<tr>
<td>11</td>
<td>20</td>
<td>1</td>
</tr>
<tr>
<td>TOT</td>
<td>548</td>
<td>106</td>
</tr>
</tbody>
</table>

Table 3 Population distribution (advanced method)

<table>
<thead>
<tr>
<th>Deck</th>
<th>Night (Cases 1 and 3)</th>
<th>Day (Cases 2 and 4)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Passengers</td>
<td>Crew</td>
</tr>
<tr>
<td>C</td>
<td>0</td>
<td>8</td>
</tr>
<tr>
<td>B</td>
<td>0</td>
<td>175</td>
</tr>
<tr>
<td>A</td>
<td>0</td>
<td>89</td>
</tr>
<tr>
<td>1</td>
<td>142</td>
<td>4</td>
</tr>
<tr>
<td>2</td>
<td>0</td>
<td>11</td>
</tr>
<tr>
<td>3</td>
<td>0</td>
<td>17</td>
</tr>
<tr>
<td>4</td>
<td>121</td>
<td>4</td>
</tr>
<tr>
<td>5</td>
<td>150</td>
<td>4</td>
</tr>
<tr>
<td>6</td>
<td>173</td>
<td>4</td>
</tr>
<tr>
<td>7</td>
<td>124</td>
<td>4</td>
</tr>
<tr>
<td>8</td>
<td>108</td>
<td>4</td>
</tr>
<tr>
<td>9</td>
<td>0</td>
<td>14</td>
</tr>
<tr>
<td>10</td>
<td>92</td>
<td>20</td>
</tr>
<tr>
<td>11</td>
<td>20</td>
<td>7</td>
</tr>
<tr>
<td>TOT</td>
<td>930</td>
<td>365</td>
</tr>
</tbody>
</table>

Table 4 Results of the different durations obtained from the simplified method (Case 1)

<table>
<thead>
<tr>
<th>Deck</th>
<th>tF [s]</th>
<th>tdeck [s]</th>
<th>tstairs [s]</th>
<th>tassembly [s]</th>
<th>tI [s]</th>
<th>T [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>11</td>
<td>63.5</td>
<td>25.3</td>
<td>53.2</td>
<td>13.1</td>
<td>155.1</td>
<td>356.8</td>
</tr>
<tr>
<td>10</td>
<td>63.5</td>
<td>25.3</td>
<td>46.4</td>
<td>13.1</td>
<td>148.3</td>
<td>341.1</td>
</tr>
<tr>
<td>9</td>
<td>63.5</td>
<td>17.7</td>
<td>39.8</td>
<td>13.1</td>
<td>134.1</td>
<td>308.4</td>
</tr>
<tr>
<td>8</td>
<td>63.5</td>
<td>40.2</td>
<td>33.0</td>
<td>13.1</td>
<td>149.8</td>
<td>344.6</td>
</tr>
<tr>
<td>7</td>
<td>63.5</td>
<td>34.9</td>
<td>26.4</td>
<td>13.1</td>
<td>137.9</td>
<td>317.1</td>
</tr>
<tr>
<td>6</td>
<td>63.5</td>
<td>37.5</td>
<td>19.8</td>
<td>13.1</td>
<td>133.9</td>
<td>308.0</td>
</tr>
<tr>
<td>5</td>
<td>63.5</td>
<td>42.8</td>
<td>13.2</td>
<td>13.1</td>
<td>132.6</td>
<td>304.9</td>
</tr>
<tr>
<td>4</td>
<td>63.5</td>
<td>36.8</td>
<td>6.60</td>
<td>13.1</td>
<td>120.0</td>
<td>276.1</td>
</tr>
<tr>
<td>3</td>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td>13.1</td>
<td>13.10</td>
<td>30.20</td>
</tr>
<tr>
<td>2</td>
<td>69.6</td>
<td>11.7</td>
<td>10.6</td>
<td>13.1</td>
<td>105.0</td>
<td>241.5</td>
</tr>
<tr>
<td>1</td>
<td>69.6</td>
<td>64.8</td>
<td>18.9</td>
<td>13.1</td>
<td>166.4</td>
<td>382.7</td>
</tr>
<tr>
<td>A</td>
<td>69.6</td>
<td>31.3</td>
<td>26.6</td>
<td>13.1</td>
<td>140.7</td>
<td>323.6</td>
</tr>
<tr>
<td>B</td>
<td>69.6</td>
<td>58.2</td>
<td>33.8</td>
<td>13.1</td>
<td>174.7</td>
<td>401.7</td>
</tr>
<tr>
<td>C</td>
<td>69.6</td>
<td>7.20</td>
<td>41.0</td>
<td>13.1</td>
<td>130.9</td>
<td>301.0</td>
</tr>
</tbody>
</table>

5.2 Advanced method application

The advanced method has been carried out by both AENEAS and EVI tools. In both software, the MVZs considered in the case study have been modelled and populated following the indications of the MSC.1/Circ.1533 (Table 3). In particular, in accordance to Annex 3 of the Circular, the population’s composition in terms of age and gender has been determined (Table 5). The two software give a 3D representation of the evacuation phases (Fig. 9 and 10).
Simplified and advanced approaches for evacuation analysis of passenger ships in the early stage of design

Nasso Carlo, Bertagna Serena, Mauro Francesco, Marinò Alberto, Bucci Vittorio

**Table 5** Composition of the population in the MVZs 5 and 6

<table>
<thead>
<tr>
<th>Population groups - passenger</th>
<th>Percentage [%]</th>
<th>Night Cases [persons]</th>
<th>Day Cases [persons]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Females younger than 30 years</td>
<td>7</td>
<td>65</td>
<td>98</td>
</tr>
<tr>
<td>Females 30-50 years</td>
<td>7</td>
<td>65</td>
<td>98</td>
</tr>
<tr>
<td>Females older than 50 years</td>
<td>16</td>
<td>149</td>
<td>223</td>
</tr>
<tr>
<td>Females older than 50 years, mobility impaired (1)</td>
<td>10</td>
<td>93</td>
<td>139</td>
</tr>
<tr>
<td>Females older than 50 years, mobility impaired (2)</td>
<td>10</td>
<td>93</td>
<td>139</td>
</tr>
<tr>
<td>Males younger than 30 years</td>
<td>7</td>
<td>65</td>
<td>98</td>
</tr>
<tr>
<td>Males 30-50 years</td>
<td>7</td>
<td>65</td>
<td>98</td>
</tr>
<tr>
<td>Males older than 50 years</td>
<td>16</td>
<td>149</td>
<td>223</td>
</tr>
<tr>
<td>Males older than 50 years, mobility impaired (1)</td>
<td>10</td>
<td>93</td>
<td>139</td>
</tr>
<tr>
<td>Males older than 50 years, mobility impaired (2)</td>
<td>10</td>
<td>93</td>
<td>139</td>
</tr>
<tr>
<td>TOTAL passengers</td>
<td>100</td>
<td>930</td>
<td>1394</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Population groups - crew</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Crew females</td>
<td>50</td>
<td>182</td>
<td>137</td>
</tr>
<tr>
<td>Crew males</td>
<td>50</td>
<td>183</td>
<td>138</td>
</tr>
<tr>
<td>TOTAL crew</td>
<td>100</td>
<td>365</td>
<td>275</td>
</tr>
</tbody>
</table>

For each Case, randomly generated simulations have been run. In AENEAS the results of each Case are statistically elaborated in order to obtain the total assembly duration $t_A$, probability frequency (Fig. 11, with reference to Case 1), and the 95-percentile value is assumed as the highest travel duration $t_I$ for the considered Case (the green bar in Fig. 11). Successively, the same analysis is carried out for the other Cases, and the travel duration $T$ to be adopted for the performance standard verification is assumed as the maximum of the four calculated travel duration $t_I$ (Table 8).

In EVI, for each Case, fifty randomly generated simulations have been analysed and the corresponding evacuation durations $t$ evaluated. In Table 6 the evacuation durations $t$ of the simulations for the Case 1 are given along with the corresponding evacuability index. In particular, it is highlighted the evacuability at the 95-percentile of the distribution that defines the highest travel duration $t_I$ of the considered Case.

![Fig. 9 From AENEAS, screenshot of a simulation step (reference to Deck 6, Case 1)](image-url)
Fig. 10 From EVI, screenshot of a simulation step (reference to Deck 6, Case 1)

Fig. 11 From AENEAS, frequency of the evacuation duration in Case 1 (green bar is at the 95-percentile)

<table>
<thead>
<tr>
<th>$t$ [s]</th>
<th>Evacuability index</th>
<th>$t$ [s]</th>
<th>Evacuability index</th>
<th>$t$ [s]</th>
<th>Evacuability index</th>
</tr>
</thead>
<tbody>
<tr>
<td>980.5</td>
<td>0.01</td>
<td>1005.0</td>
<td>0.35</td>
<td>1023.0</td>
<td>0.69</td>
</tr>
<tr>
<td>986.0</td>
<td>0.03</td>
<td>1005.5</td>
<td>0.37</td>
<td>1023.0</td>
<td>0.71</td>
</tr>
<tr>
<td>990.5</td>
<td>0.05</td>
<td>1006.5</td>
<td>0.39</td>
<td>1024.5</td>
<td>0.73</td>
</tr>
<tr>
<td>990.5</td>
<td>0.07</td>
<td>1007.0</td>
<td>0.41</td>
<td>1025.5</td>
<td>0.75</td>
</tr>
<tr>
<td>991.5</td>
<td>0.09</td>
<td>1009.0</td>
<td>0.43</td>
<td>1028.0</td>
<td>0.77</td>
</tr>
<tr>
<td>994.5</td>
<td>0.11</td>
<td>1011.5</td>
<td>0.45</td>
<td>1028.0</td>
<td>0.79</td>
</tr>
<tr>
<td>995.5</td>
<td>0.13</td>
<td>1012.0</td>
<td>0.47</td>
<td>1028.5</td>
<td>0.81</td>
</tr>
<tr>
<td>997.0</td>
<td>0.15</td>
<td>1012.0</td>
<td>0.49</td>
<td>1030.5</td>
<td>0.83</td>
</tr>
<tr>
<td>997.0</td>
<td>0.17</td>
<td>1012.5</td>
<td>0.51</td>
<td>1031.0</td>
<td>0.85</td>
</tr>
<tr>
<td>997.5</td>
<td>0.19</td>
<td>1016.0</td>
<td>0.53</td>
<td>1032.0</td>
<td>0.87</td>
</tr>
<tr>
<td>999.0</td>
<td>0.21</td>
<td>1016.5</td>
<td>0.55</td>
<td>1033.0</td>
<td>0.89</td>
</tr>
<tr>
<td>1000.0</td>
<td>0.23</td>
<td>1017.0</td>
<td>0.57</td>
<td>1033.0</td>
<td>0.91</td>
</tr>
<tr>
<td>1001.5</td>
<td>0.25</td>
<td>1017.5</td>
<td>0.59</td>
<td>1033.0</td>
<td>0.93</td>
</tr>
<tr>
<td>1002.0</td>
<td>0.27</td>
<td>1019.0</td>
<td>0.61</td>
<td><strong>1033.5</strong></td>
<td><strong>0.95</strong></td>
</tr>
<tr>
<td>1002.5</td>
<td>0.29</td>
<td>1019.5</td>
<td>0.63</td>
<td>1042.0</td>
<td>0.97</td>
</tr>
<tr>
<td>1004.0</td>
<td>0.31</td>
<td>1020.0</td>
<td>0.65</td>
<td>1053.5</td>
<td>0.99</td>
</tr>
<tr>
<td>1004.0</td>
<td>0.33</td>
<td>1021.5</td>
<td>0.67</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
5.3 Comparison between the methods and software

In both software, the Case 4 is the worst one (Table 8 and 9), but the performance standard (80 minutes) is fulfilled. Through the 3D representation, it is easy to verify the presence of possible congestion points (for instance, in the case study considered a congestion point at Deck 6 has been noted), while the same analysis is quite difficult within the simplified method. Indeed, the simplified method has not identified any possible congestion point within the two considered cases.

In order to make a complete comparison between the simplified and the advanced methods, the calculation times required by both methods have been analysed. With regard to the simplified method, the low complexity of the various operations involved, makes the calculation time negligible. However, it is necessary to consider not only the pure calculation time, but also the time required to update the population distribution for every scenario. As regards the advanced method, the calculation time required is higher, but the result consists of a batch running of 50 different simulations. Here, the calculation time is strongly influenced by the complexity and dimension of the pedestrian path. For the vessel considered in this study, the total calculation time per each scenario is of about 15 minutes on a regular 2.7 GHz 4 cores workstation. The two software have almost the same calculation time and performances. For more complex and bigger ships, calculation time may increase up to 2 hours per case.

In conclusion, for a geometry like the one considered in this study, the total time needed for the problem modelling and scenarios calculation is fully comparable between simplified and advanced methods.

<table>
<thead>
<tr>
<th>Case</th>
<th>$T$ [s]</th>
<th>$t_{TOT}$ [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>402</td>
<td>2452</td>
</tr>
<tr>
<td>2</td>
<td>1328</td>
<td>3235</td>
</tr>
<tr>
<td>3</td>
<td>not considered</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>not considered</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Case</th>
<th>$t_i$ [s]</th>
<th>$T$ [s]</th>
<th>$t_{TOT}$ [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>980</td>
<td>2344</td>
<td>4505</td>
</tr>
<tr>
<td>2</td>
<td>667</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>1483</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>2344</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Case</th>
<th>$t_i$ [s]</th>
<th>$T$ [s]</th>
<th>$t_{TOT}$ [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1033.5</td>
<td>2510</td>
<td>4713</td>
</tr>
<tr>
<td>2</td>
<td>643</td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>1354.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>2510</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

6. Conclusions

The new IMO Regulation MSC.1/Circ.1533 issued on May 2016 about evacuation analysis of passenger ships has been thoroughly presented, and a case study regarding a 3600-person cruise ship has been reported. Both simplified and advanced methods have been considered. In particular, the advanced method has been carried out by two multi-agent software based on virtual reality, both certified by the IMO: AENEAS by DNV-GL and TraffGo, and EVI by the University of Strathclyde-Glasgow. It is worth noting that in the investigated case study the result (in terms of total evacuation duration $t_{TOT}$) stemming from simplified analysis is much smaller than that evaluated by the advanced method. For instance, with reference to the Case Study reported in the paper, by the simplified method the total evacuation duration $t_{TOT}$ is equal to 3235 s, whereas the advanced method carried out with AENEAS gives $t_{TOT} = 4505$ s, and carried out with EVI gives $t_{TOT} = 4713$ s. Although both simplified and advanced methods meet the performance standard set by the Circular, a deeper investigation should be done in order to make the results of the simplified method more reliable.
REFERENCES


Simplified and advanced approaches for evacuation of passenger ships in the early stage of design

Nasso Carlo, Bertagna Serena, Mauro Francesco, Marinò Alberto, Bucci Vittorio

https://doi.org/10.1007/978-3-319-41682-3_33


Submitted: 13.03.2018. Nasso Carlo, cnasso@units.it
Bertagna Serena, sbertagna@units.it
Accepted: 05.06.2019. Mauro Francesco, fmauro@units.it
Marinò Alberto, marino@units.it
Bucci Vittorio, vbucci@units.it

Department of Engineering and Architecture,
University of Trieste, via A. Valerio, 10, I-34127 Trieste
ENVIRONMENTAL AND COST-EFFECTIVENESS COMPARISON OF DUAL FUEL PROPULSION OPTIONS FOR EMISSIONS REDUCTION ONBOARD LNG CARRIERS

UDC 629.542:629.5.016

Review paper

Summary

The selection of the suitable propulsion system for LNG carrier highly affects the ship capital and life cycle costs. The current paper compares between the available propulsion systems for LNG carriers from environmental and economic points of view operated with heavy fuel oil (HFO) and marine gas oil (MGO). In addition, the cost-effectiveness for emission reduction due to using dual fuel propulsion options using natural gas fuel (NG) is calculated. As a case study, large conventional LNG carrier class has been investigated. The results show that steam turbine (ST), Ultra-ST, dual fuel diesel engine (DFDE), and combined gas and steam (COGAS) propulsion options can comply with NO\textsubscript{x} and SO\textsubscript{x} emissions regulations set by IMO using dual fuel mode with NG percentages of 87.5%, 82%, 98.5% and 94%, respectively. DFDE operated with pilot HFO and NG is the most economic propulsion option. It reduces the dual fuel costs by 1.37 MUS$/trip compared with HFO cost. The annual cost-effectiveness for the most economic and emission compliance propulsion option is 6.07 $/kg, 6.39 $/kg, and 0.55 $/kg for reducing NO\textsubscript{x}, SO\textsubscript{x}, and CO\textsubscript{2} emissions, respectively.

Key words: LNG carriers; Propulsion options; Boil-off gas; Environmental and economic analysis; EEDI; Fuel saving cost-effectiveness

1. Introduction

The demand on natural gas supply has been increased in the last years to reduce the exhaust gas emissions especially the greenhouse gas [1, 2]. Because of these demands, liquefied natural gas (LNG) market is increasing with the increased number of LNG vessels [3-5]. LNG reduces the gas volume by 600 times using deep cooling of −163 °C at a pressure slightly higher than the atmospheric pressure [6, 7]. Boil-off gas (BOG) is one of the main characteristics of the LNG tanks [8]. Therefore, the selection of the LNG carrier propulsion system is constrained by LNG properties and different economic and environmental factors [9]. There is no standard marine power plant for LNG ships [10]. Different propulsion systems are installed onboard varying from turbines to internal combustion engines.

LNG carriers are designed according to the gas code regulations of the international maritime organization (IMO). The gas tanks are built using “cargo containment system”.
They are arranged as spherical (moss), membrane, or prismatic type tanks [11]. BOG occurs in these tanks due to the heat transfer from the surrounding environment which results in evaporation of the liquefied gases. This evaporation rate is increased during cargo transportation [12-14].

On the other hand, the application of natural gas in marine engines depends on its properties. Natural gas is lighter than air, and in the case of leakage it disperses to the atmosphere. Evaporation process of the LNG makes it easy to float away unlike other liquid fuels which remain near the engine and the bilge. The flammability of NG is only possible within a tight mixture with air ranging (5%: 15 %). The properties of NG and conventional marine fuel oil are summarized in Table 1 [15-17].

<table>
<thead>
<tr>
<th>Property</th>
<th>Marine fuel oil</th>
<th>Natural gas</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ignition temperature, °C</td>
<td>250</td>
<td>600</td>
</tr>
<tr>
<td>Density, kg/m³@ 1 bar</td>
<td>850</td>
<td>0.74</td>
</tr>
<tr>
<td>LCV, MJ/kg</td>
<td>42</td>
<td>50</td>
</tr>
<tr>
<td>Carbon contents (%)</td>
<td>84.7</td>
<td>70</td>
</tr>
<tr>
<td>Hydrogen contents (%)</td>
<td>12</td>
<td>20</td>
</tr>
</tbody>
</table>

2. Propulsion options for LNG carriers

The type and the classification of LNG propulsion system are highly affected by the generation of the BOG and the emission regulations set by the IMO [12]. Steam turbine (ST) based propulsion system was the first system to be used for LNG carriers since 1960 [18]. It is allowed for burning the used fuel together with the generated BOG during transportation. In 2003, internal combustion engines replaced the ST, due to the improvement in their performance and efficiency. In addition, the dual fuel diesel engine (DFDE) permits the burning of the BOG with the heavy fuel oil [19]. DFDE was started in 4-stroke engine, since 2003. At present, 2-stroke engines can also use NG as a fuel. This can lead to a dramatically change in the LNG propulsion system [14]. The main propulsion systems used in LNG carriers are steam turbine, DFDE, slow speed diesel engine, and gas turbine in combined cycle.

2.1 Steam turbine propulsion (ST)

Steam turbine is the first propulsion system used for LNG carriers because of the boiler flexibility to burn the natural BOG from the cargo. This propulsion system normally consists of two boilers each produces steam with a rate of 80-90 ton/hr at 60-70 bar and 520 °C [20]. The total power of the plant is 35-45 MW produced through high, intermediate, and low pressure turbines. For speed reversal, the low pressure turbine incorporates a stern turbine on the same rotor shaft. The electric power demand onboard is supplied by two steam turbines generators and one medium-speed diesel generator. The estimated overall thermal efficiency of 30 MW conventional steam power plant powered by Mitsubishi is 35% [21]. In order to improve the thermal efficiency of the steam power plant, reheating of the high pressure steam turbine is incorporated [14, 22]. This improved cycle is called Ultra Steam Turbine (UST) as shown in Fig. 1a. The modified cycle saves 15% of the fuel consumption compared with the conventional steam power plant with an overall fuel efficiency of 41%. This can be
considered as a competitive to the DFDE power plant from fuel consumption point of view [23-25].

2.2 Dual fuel diesel engine (DFDE)

Medium speed diesel engines can be considered as an alternative to the conventional steam turbines with low fuel efficiency. They can burn both the liquid and gas fuels in the dual fuel mode. The BOG is used in the gas mode operation with lean air to fuel ratio on the principle of the Otto cycle with pilot diesel fuel injection in the cylinder for ignition. The engine is operated using a completely liquid fuel, marine diesel oil (MDO) or heavy fuel oil (HFO), when the amount of the BOG is insufficient. In this case, the BOG is burned in the gas combustion unit (GCU) with the disadvantage of the energy loss. This loss associated with the losses of the electrical components of the used propulsion system can be ranged from 6% to 8%, when comparing DFDE with other marine power plants. Fig. 1b shows the DFDE propulsion plant for an LNG carrier. This system uses electric propulsion where the electrical power for both the propulsion and the cargo handling are in altered operating time phase which reduces the net power requirement compared with the mechanical propulsion plant. On the other hand, this propulsion system requires a complex control system especially air to fuel ratio controller [2, 26].

2.3 Slow speed diesel engine (SSDE)

Slow speed diesel engines are used for LNG carrier propulsion especially for large capacities over 200,000 m$^3$ and the long distance tradeoff ships. It is the most efficient propulsion engine used onboard ships, at the moment. The main advantages of slow speed diesel engine are the high efficiency, low maintenance and operating costs, and the possibility of burning low-quality cheap fuels [27]. This propulsion system uses both the gas combustion unit (GCU) and the reliquefaction plant for the naturally generated BOG as shown in Fig. 1c. The reliquefaction plant converts the generated BOG into a liquid and this reduces any loss in the transported cargo. In case of any breakdown in this system or during any maintenance procedures, the GCU is used to burn the BOG to avoid any damage in the LNG tanks because of the increase in the storage pressure [14]. The auxiliary and electric powers in this propulsion system are provided using 4-stroke diesel generators. In case of twin screw propellers, shaft disconnecting devices are used in each shaft line to immediately disconnect the failed engine from the propulsion shaft line and continue the ship voyage [28, 29].

2.4 Gas Turbines in combined cycle (COGAS)

The combined cycle is an unusual propulsion system for LNG carriers, because it does not provide a good flexibility especially for auxiliary power generators. Although gas turbines (GT) have many advantages such as good reliability, high power to weight ratio, compact size, and quick response to power demand, ship owners do not prefer using it because of the low fuel efficiency. Most of the applications of the gas turbines in the marine field are used in their combined cycle especially for naval and offshore industry [30-32]. Fig. 1d shows a combined gas and steam turbines (COGAS) propulsion power arrangement for LNG carrier. The gas turbine provides the propulsion torque after using a reduction gear. The exhaust gas boiler is operated using the heat lost in the exhaust gases generated from the gas turbine. It is
coupled to a generator which provides a mechanical power through the reduction gear to the ship propeller during cruise. At port, both turbines are stopped and three power generators are used for power generation during cargo loading and uploading operations [19, 33].

![Diagram](image1.png)

**Fig. 1** Different LNG propulsion options

The current paper aims to compare between the available propulsion systems for LNG carriers from environmental and economic points of view. The comparison will be performed for the most used marine fuels in LNG carriers’ propulsion options. The used fuel for all propulsion options is the heavy fuel oil (HFO) except COGAS operates with marine gas oil (MGO) and DFDE uses both HFO and MGO [2, 14, 22, 34]. In addition, the cost-effectiveness for emission reduction due to using the dual fuel propulsion systems is investigated for large conventional LNG carrier.

### 3. Large conventional LNG carriers case study

LNG carriers can be classified into five main classes based on its volumetric capacity of LNG in m³. These classes are small, small conventional, large conventional, Q-flex, and Q-mass. They range from small volumes up to 90,000 m³ for the small class and more than 260,000 m³ for Q-mass class. One of the most common classes, which is selected for the current case study, is the large conventional. The average particulars for this class can be listed in Table 2 [35]. The maximum design draft is limited to 12 m due to the available port facilities. This results in quite high beam to draft ratio above 4.0. For this case, twin-screw propulsion system will reduce the required main engine power up to 9.0% compared with the single screw system. In addition, most of the LNG carrier propulsion engines are designed to
use BOG from the cargo. The case study ship is assumed to export the LNG from Qatar, the largest exporter in the world, to Japan, the largest importer of LNG in 2017 [36, 37]. The average distance from Qatar to Japan is 6347 nautical miles. The tip time will be 26 days and 10.0 hours, using ship speed of 20.0 knots [38]. The number of trips per year is assumed to be 10.0.

**Table 2** Large conventional LNG carrier using twin-screw propeller average particulars

<table>
<thead>
<tr>
<th>LNG carrier item</th>
<th>Particulars</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Membrane Type</td>
</tr>
<tr>
<td>Ship size, LNG capacity</td>
<td>150,000 m³</td>
</tr>
<tr>
<td>Scantling draught</td>
<td>12.3 m</td>
</tr>
<tr>
<td>Length overall</td>
<td>288.0 m</td>
</tr>
<tr>
<td>Length between pp</td>
<td>275 m</td>
</tr>
<tr>
<td>Breadth</td>
<td>44.2 m</td>
</tr>
<tr>
<td>Design draught</td>
<td>11.6 m</td>
</tr>
<tr>
<td>Average design speed</td>
<td>20.0 Knots</td>
</tr>
<tr>
<td>Power (MCR)</td>
<td>2x14,900 kW</td>
</tr>
</tbody>
</table>

4. Environmental and economic modeling

The emission of pollutant (j) over a complete ship trip in tones \(m_{e,j}\) can be calculated using Eq. (1) [39, 40].

\[
m_{e,j} = P \cdot L \cdot T \cdot EF_j
\]

where, \(P\) is the engine power in kW with its load factors \((L)\), \(T\) is the trip time in hours, and \(EF_j\) is the emission factor of the pollutant \((j)\) expressed in ton/kWh. Table 3 shows the different emission factors \((EF)\) for gas turbine (GT), steam turbine (ST), slow and medium speed marine diesel engine (SSDE and MSDE) operating on heavy fuel oil (HFO), marine gas oil (MGO), and natural gas (NG) [14, 40-45].

NG can be used in a dual fuel mode in LNG carriers. The emission factor in case of using dual fuel engine \((EF_{DF,j})\) for each pollutant emission can be calculated using Eq. (2).

\[
EF_{DF,j} = x_m EF_m + x_{NG} EF_{NG}
\]

where, \(x_m\) and \(x_{NG}\) are the percentages of the main fuel and the NG fuels in dual-fuel engine (DFE), \(EF_m\) and \(EF_{NG}\) are the emission factors in g/kWh for the main and the natural gas engines.

The percentage of the BOG in the dual fuel mode during one trip can be calculated using the boil-off rate (BOR). It represents the quantity of the evaporated LNG per day expressed as a percentage of the total cargo (%/day) [46, 47]. BOR can be calculated using Eq. (3).

\[
BOR = \left( \frac{Q}{H_{latent} \times 3600 \times 24} \times \frac{100}{\rho_{LH_2} V_{carg o}} \right)
\]
where, \( Q \) is the heat exchange in LNG tanks in kW, \( \rho \) is the density of LNG in kg/m\(^3\), and \( H_{\text{latent}} \) is the heat of vaporization in kJ/kg. The average BOR values for new LNG tankers range from 0.10 to 0.15% /day for loaded voyage and from 0.06 to 0.10% /day for ballast voyage [48-50].

Table 3: Emission factors for different LNG propulsion options in g/kWh

<table>
<thead>
<tr>
<th>Engine type</th>
<th>Fuel used</th>
<th>Emission factors (g/kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>NO(_x)</td>
</tr>
<tr>
<td>SSDE</td>
<td>HFO (2.7%S)</td>
<td>17</td>
</tr>
<tr>
<td>MSDE</td>
<td>MGO (0.1%S)</td>
<td>13.2</td>
</tr>
<tr>
<td></td>
<td>NG</td>
<td>2.16</td>
</tr>
<tr>
<td>COGAS</td>
<td>MGO (0.1%S)</td>
<td>14</td>
</tr>
<tr>
<td></td>
<td>NG</td>
<td>0.9</td>
</tr>
<tr>
<td>ST</td>
<td>HFO (2.7% S)</td>
<td>11.0</td>
</tr>
<tr>
<td></td>
<td>NG</td>
<td>0.4</td>
</tr>
<tr>
<td>UST</td>
<td>HFO (2.7% S)</td>
<td>8.25</td>
</tr>
<tr>
<td></td>
<td>NG</td>
<td>0.3</td>
</tr>
</tbody>
</table>

The emission factors for LNG ship should be compared with the required IMO emission rates for NO\(_x\), SO\(_x\), and CO\(_2\). For NO\(_x\) emissions, the emission limit equations, expressed in g/kWh, of the applicable Tier III values, only for NECA (NO\(_x\) Emission control areas), based on the rated engine speeds in rpm are shown in Eq. (4) [28, 40, 44, 51-53].

\[
NO_{x,Tier\ III} = \begin{cases} 
3.4 & \text{for } rpm < 130 \\
9 \times rpm^{-0.2} & \text{for } 130 \leq rpm < 2000 \\
2.0 & \text{for } rpm \geq 2000 
\end{cases} 
\]  

(4)

SO\(_x\) emissions are limited by the sulfur percent in the used marine fuel. For 2020 IMO SO\(_x\) regulations, the permitted sulfur percent in the fuels is 0.5% [40, 51, 54-56]. On the other hand, greenhouse gas (GHG) emissions especially CO\(_2\) emissions are limited by IMO through introducing Energy Efficiency Design Index (EEDI) and Energy Efficiency Operational Indicator (EEOI) [57]. The calculated EEDI should be compared with the reference values for EEDI in three phases according to the ship type. For LNG ship, the reference and the calculated values for the EEDI are based on the ship deadweight (DWT) as expressed in Eqs. (5) and (6) [46, 47, 58-62].

\[
EEDI_{ref.} = 2253.7 \times DWT^{0.474} 
\]  

(5)

\[
EEDI_{cal.} = \frac{P_{ME} \times \left( C_{F,pilot} \times SFC_{ME,pilot} + C_{F,fuel} \times SFC_{ME,fuel} \right)}{f_c \times V_{ref} \times DWT} + \frac{P_{AE} \times \left( C_{F,pilot} \times SFC_{AE,pilot} + C_{F,fuel} \times SFC_{AE,fuel} \right)}{f_c \times V_{ref} \times DWT} 
\]  

(6)
Environmental and cost-effectiveness comparison of dual fuel propulsion options for emissions reduction onboard LNG carriers

Nader R. Ammar

where, $P_{ME}$ is the main engine power, it can be calculated using Eq. (7). $P_{AE}$ is the auxiliary power required to operate the accommodation of crew and the main engine, $V_{ref}$ is the reference ship speed in knots, $C_F$ is the fuel conversion factor to CO$_2$ emissions. If LNG carrier uses rellifiquefaction plant, $P_{AE}$ will include $P_{AE,Reliq}$ for the EEDI calculation, $COP_{Reliq}$ and $COP_{cooling}$ are the coefficients of performance for the rellifiquefaction and cooling plants, respectively as expressed in Eqs. (8) - (10) [47, 63, 64].

$$P_{ME} = 0.75 \times MCR$$

(7)

$$P_{AE} = 0.025 \times MCR + 250 + P_{AE,Reliq}$$

(8)

$$P_{AE,Reliq} = V_{cargo} \times BOR \times COP_{Reliq}$$

(9)

$$COP_{Reliq} = \frac{\rho_{LH_2} \times H_{latent}}{24 \times 3600 \times COP_{cooling}}$$

(10)

The cubic capacity correction factor ($f_c$), used in Eq. (6), equals 1.0 except for direct-diesel-driven LNG carrier. It can be calculated using Eq. (11), where $R$ is the ship deadweight divided by the cargo capacity.

$$f_c = R^{-0.56}$$

(11)

From economic point of view, the annual cost for installation each propulsion system (AC) depends on the capital cost value (CC), the average expected working years (n), and the interest rate (i) [65]. AC can be calculated using Eq. (12).

$$AC = CC \times \frac{i(1+i)^n}{(1+i)^n - 1}$$

(12)

In addition, the annual fuel saving cost due to using NG ($FS_{NG}$) in dual fuel mode onboard LNG carrier can be calculated using Eq. (13).

$$FS_{NG} = (C_{DO} - C_{DF}) \times (1 \pm PI)^n$$

(13)

where, $C_{DO}$ and $C_{DF}$ are the diesel fuel and the dual-fuel costs, respectively. (PI) is the annual fuel price change percent over the working years (n) of the ship life cycle.

Finally, the annual cost-effectiveness of each propulsion system (ACE) for reducing a pollutant emission ($j$) after using dual fuel engine can be calculated using Eq. 14 [40, 42].

$$ACE_j = \frac{AC + OC}{ER_j}$$

(14)

where, OC is the operating and maintenance costs for the propulsion system in $/year. ER$_j$ is the annual emission reduction in ($j$) after using dual fuel engine expressed in ton/year.

5. Results and discussion

In this section, the environmental results for different LNG carrier propulsion options using HFO and MGO are discussed. In addition, the economic and cost-effectiveness analysis for the dual fuel operated propulsion options, using NG, are calculated.
Fuel efficiency not only affects the operating costs of the marine propulsion power plant but also extremely influence the emitted exhaust gas emissions. Although the efficiency of steam power plant is lower than internal combustion engines, UST with reheating has improved the efficiency of the steam cycle to a comparative level. Typical efficiencies of 157,000 m$^3$ LNG carrier using ST, UST, SSDE with reliquefaction plant, dual fuel MSDE, and COGAS power plants are 35%, 41%, 40%, 42%, and 50%, respectively [2, 29, 66]. UST emission factors are reduced by 25% compared with the simple GT cycle [14]. LNG carrier propulsion systems can fulfill the required levels of NO$_x$ and SO$_x$ emission levels set by IMO depending on the used plant and the fuel type. Fig. 2 shows the relative NO$_x$ and SO$_x$ emissions from the five most used marine power plants for LNG carriers using HFO and MGO. Any observed power plant satisfies IMO standards (for both SO$_x$ and NO$_x$) if the relative emissions are 100% or lower.

Due to the strict IMO regulations that limit the exhaust gas emissions from ships, it is an important factor to consider using the BOG as a secondary fuel in LNG carriers during the design process. This will help in reducing the exhaust gas emissions. From section 2, SSDE propulsion option use BOG either in a reliquefaction plant or in GCU. Thus, it is not included in the dual fuel mode. Fig. 3 shows NO$_x$ and SO$_x$ emissions from LNG propulsion plants using HFO and MDO in dual fuel mode using BOR of 0.15%/day for the loaded voyage [48-50]. The share percentages of these BOG in LNG propulsion fuel is calculated based on the fact that each cubic meter of diesel oil consumption is equivalent for 1197 m$^3$ of NG [44]. In addition, the volume of NG is reduced by 600 times when converted to the liquid state [40, 70]. The percentages of BOG in dual fuel mode range from 55.47% to 79.49% using ST and COGAS, respectively for the case study. Using BOG, both NO$_x$ and SO$_x$ emission rates cannot be complied with IMO regulations, using different LNG propulsion options, as shown in Fig. 3.
Environmental and cost-effectiveness comparison of dual fuel propulsion options for emissions reduction onboard LNG carriers

Nader R. Ammar

From Fig. 3, NG percent can be increased in the dual fuel mode to reduce the exhaust gas emissions to the accepted rates set by IMO for NO\textsubscript{x} and SO\textsubscript{x} emissions. The minimum NG percentages in dual fuel mode for the different propulsion options are 87.5%, 82%, 98.5% and 94% to achieve the required IMO values for NO\textsubscript{x} and SO\textsubscript{x} emissions using ST, UST, DFDE, and COGAS, respectively as shown in Fig. 4. The shares of BOG in these percentages are 55.47%, 65.0%, 66.67%, and 79.49%, respectively. Moreover, using dual fuel will reduce CO\textsubscript{2} emissions because of the reduced carbon content in NG compared with liquid marine fuels. Fig. 4 shows the relative CO\textsubscript{2} emissions of different LNG carrier propulsion systems using dual fuel propulsion systems with the accepted NO\textsubscript{x} and SO\textsubscript{x} emission levels set by IMO. CO\textsubscript{2} emissions from ships are one of the major concerns of the IMO due to its bad influence on the global warming. The highest and the lowest CO\textsubscript{2} emission reduction percentages are achieved by the ST and the COGAS with percentages of 64.83% and 12.75%, respectively. These reductions in CO\textsubscript{2} emissions will improve the energy efficiency of the ship through calculating EEDI and EODI [62].

CO\textsubscript{2} emissions presented in Fig. 4 have to be complied with the required IMO regulations. In addition, the newly built LNG carriers should be designed to be energy efficient to reduce carbon dioxide emissions through calculating the energy efficiency design index (EEDI). It depends on the type of the ship, the main and auxiliary engines, the construction, and the used fuel. It calculates the amount of CO\textsubscript{2} emissions per unit distance of cargo transportation. Fig. 5 shows the permitted CO\textsubscript{2} emissions set by IMO in gCO\textsubscript{2}/ton-NM (Required EEDI). The values of the EEDI for LNG carriers depend on the ship deadweight presented in three phases according to the ship built year. The base line values will be reduced
by 20 and 30 percentages in the second and the third phases in the years 2020 and 2025, respectively.

5.2 Economic and cost-effectiveness results

The economic feasibility of LNG carrier propulsion options can be judged using the total costs for each option and its environmental impact assessment. The total costs include the initial installation and operational costs. The initial costs per unit power for ST, DFDE, SSDE, and COGAS are 136 $/kW, 667 $/kW, 940 $/kW, and 1410 $/kW, respectively [47, 71, 72]. For LNG carrier of 150,000 m³ capacity, the initial installations costs for ST, DFDE, SSDE, and COGAS propulsion systems are 4.05 MUS$, 1.99 MUS$, 2.80 MUS$, and 4.20 MUS$, respectively. On the other hand, the fuel consumption cost is the highest percent of the operating costs over the life cycle of the propulsion power plants [47]. The prices of HFO, MGO, and NG are 556$/m³, 882 $/m³, and 0.3047 $/m³, respectively [73-75]. The cost of HFO consumptions per trip are 2.99 MUS$, 2.55 MUS$, 2.49 MUS$, and 2.62 MUS$ for the case study propulsion options using ST, UST, DFDE, and SSDE, respectively. On the other hand, the costs of MGO fuel per trip are 3.94 MUS$ and 3.31 MUS$ for DFDE and COGAS, respectively. Based on 2018 fuel oil and NG prices using dual fuel engines will save in the fuel consumption due to the low price of NG compared with the diesel oil prices. Fig. 6 illustrates the fuel saving cost per trip for different LNG propulsion options operated with dual fuel engines with different NG percentages. The highest fuel saving cost is 2.79 MUS$/trip for DFDE propulsion system operated with MGO. In contrast, the lowest fuel saving cost is 1.0 MUS$/trip for UST operated with dual HFO and NG fuels.
The economic assessment for LNG carrier propulsion options complied with IMO regulations using dual fuel mode can be evaluated using the total annual costs for each option. Fig. 7 shows the total annual costs and emission reduction percentages for ST, UST, DFDE, and COGAS propulsion options operated in dual fuel mode. The total annual costs for ST, UST, and DFDE, propulsion options operated with HFO-NG dual fuel are 15.05 MUS$, 15.90 MUS$, and 13.38 MUS$, respectively. The reduction percentages in NO<sub>x</sub> emissions for these options compared with the HFO operated engines are 84.32%, 79.02%, and 83.3%, respectively. For SO<sub>x</sub> emissions, the reduction percentages will be 87.5%, 82%, 98.5%, respectively. On the other hand, the total annual costs for MGO-NG dual fuel operated COGAS propulsion system is 17.38 MUS$ with zero SO<sub>x</sub> emissions and 87.96% NO<sub>x</sub> emission reduction percent.

From Fig. 7, DFDE operated with dual HFO and NG is the most economic and emission compliance propulsion option for large conventional LNG carrier. The annual costs for capital
cost recovery and the annual fuel saving costs for DFDE, compared with the HFO operated engine, are presented in Fig. 8. The annual costs are calculated over the assumed ship life cycle of 28 years [40, 76]. The annual costs for capital cost recovery and fuel saving, at the end of the ship life cycle, will be 2.14 MUS$ and 23.80 MUS$, assuming annual interest rate of 10% and fuel price increment of 2%, respectively.

![Fig. 8 Annual costs for DFDE propulsion option over LNG carrier life cycle](image)

In order to combine the environmental benefits and the economic analysis for the four propulsion options for LNG carriers achieved IMO emission requirements, the cost-effectiveness for each reduction in pollutant emissions is calculated. It assesses the economic benefits for the total costs of each propulsion option in terms of its environmental consequences. The cost effectiveness is calculated for the three most economic propulsion options achieved IMO NO\textsubscript{x} and SO\textsubscript{x} emission requirements. Fig. 9 compares the cost-effectiveness for reducing NO\textsubscript{x}, SO\textsubscript{x}, and CO\textsubscript{2} emissions after using dual fuel propulsion options for DFDE, UST, and COGAS at annual interest rate of 10%. On the Fig. 9 the lower value is better. The most economic propulsion option is the DFDE operated with dual HFO and NG fuels. It reduces NO\textsubscript{x}, SO\textsubscript{x}, and CO\textsubscript{2} emissions with annual cost-effectiveness of 6.07 $/kg, 6.39 $/kg, and 0.55 $/kg, respectively.

![Fig. 9 The annual cost-effectiveness for reducing NO\textsubscript{x}, SO\textsubscript{x} and CO\textsubscript{2} emissions](image)
6. Conclusions

Environmental, economic and cost-effectiveness analysis for the available LNG carrier propulsion options operated with heavy fuel oil (HFO), marine gas oil (MGO), and dual fuel (with natural gas) were investigated. These options include steam turbine (ST), ultra steam turbine (UST), dual fuel diesel engine (DFDE), slow speed diesel engine (SSDE), and combined gas and steam (COGAS) propulsion systems. The used fuel for all the propulsion options is the HFO except COGAS operates with MGO and DFDE uses both HFO and MGO. The main conclusions for large conventional LNG carrier with a capacity of 150,000 m³ are:

- From environmental point of view, ST, UST, DFDE, and COGAS propulsion options can fulfill the required IMO values for NOₓ and SOₓ emissions using NG percentages in dual fuel mode with percentages of 87.5%, 82%, 98.5% and 94%, respectively. The shares of boil off gas (BOG) in these percentages are 55.47%, 65.0%, 66.67%, and 79.49%, respectively. The highest CO₂ emission reduction percent is achieved by the UST with a reduction percent of 64.83% from the same cycle without NG.

- From economic point of view, Using BOG as fuel will save the cost of fuel consumption by 19.08%, 22.35%, 22.9% and 46.62% for ST, UST, DFDE and COGAS propulsion options, respectively. Increasing NG percentages to achieve the NOₓ and the SOₓ emission rates set by IMO for ST, UST, and COGAS propulsion options will save the dual fuel cost by 1.53 MUS$/year, 1.0 MUS$/year, and 2.02 MUS$/year, respectively. On the other hand, DFDE operated with dual HFO and NG is the most economic propulsion option with total annual costs of 13.38 MUS$ and emission reduction percentages of 83.30%, 98.50%, and 18.85% for NOₓ, SOₓ and CO₂ emissions, respectively.

- From cost-effectiveness point of view, the total annual costs for ST, UST, and DFDE, propulsion options operated with HFO-NG dual fuels are 15.05 MUS$, 15.90 MUS$, and 13.38 MUS$, respectively. On the other hand, the total annual costs for MGO-NG dual fuel operated COGAS propulsion system is 17.38 MUS$. DFDE operated with HFO and NG fuels is the most economic and IMO emission compliance propulsion option. It reduces NOₓ, SOₓ, and CO₂ emissions with annual cost-effectiveness of 6.07 $/kg, 6.39 $/kg, and 0.55 $/kg, respectively.

Nomenclature

<table>
<thead>
<tr>
<th>AC</th>
<th>Annual cost for installation, $/year</th>
</tr>
</thead>
<tbody>
<tr>
<td>BOR</td>
<td>Boil-off gas rate, %/day</td>
</tr>
<tr>
<td>C</td>
<td>Annual fuel cost, $/year</td>
</tr>
<tr>
<td>Cr</td>
<td>Fuel conversion factor to CO₂ emissions</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance</td>
</tr>
<tr>
<td>EEDI</td>
<td>Energy Efficiency Design Index, gCO₂/ton-NM</td>
</tr>
<tr>
<td>EF</td>
<td>Engine emission factor, kg/kWh</td>
</tr>
<tr>
<td>ER</td>
<td>Emissions reduction percentage,%</td>
</tr>
<tr>
<td>FS</td>
<td>Fuel saving cost, $/year</td>
</tr>
<tr>
<td>FSE</td>
<td>Fuel saving cost-effectiveness, $/ton</td>
</tr>
<tr>
<td>i</td>
<td>Annual interest rate, %</td>
</tr>
<tr>
<td>L</td>
<td>Engine load percent in ship modes</td>
</tr>
<tr>
<td>LCV</td>
<td>Lower calorific value, kJ/kg</td>
</tr>
<tr>
<td>BOG</td>
<td>Boil-off gas</td>
</tr>
<tr>
<td>CO₂</td>
<td>carbon dioxide</td>
</tr>
<tr>
<td>COGAS</td>
<td>Combined gas and steam</td>
</tr>
<tr>
<td>DFDE</td>
<td>Dual fuel diesel engine</td>
</tr>
<tr>
<td>GCU</td>
<td>Gas combustion unit</td>
</tr>
<tr>
<td>GT</td>
<td>Gas turbine</td>
</tr>
<tr>
<td>HFO</td>
<td>Heavy fuel oil</td>
</tr>
<tr>
<td>IMO</td>
<td>International Maritime Organization</td>
</tr>
<tr>
<td>LNG</td>
<td>Liquefied natural gas</td>
</tr>
<tr>
<td>MGO</td>
<td>Marine gas Oil</td>
</tr>
<tr>
<td>MSDE</td>
<td>Medium speed marine diesel engine</td>
</tr>
<tr>
<td>NG</td>
<td>Natural gas</td>
</tr>
<tr>
<td>NOₓ</td>
<td>Nitrogen Oxides Emissions</td>
</tr>
</tbody>
</table>

Abbreviations

- AC: Annual cost for installation, $/year
- BOR: Boil-off gas rate, %/day
- C: Annual fuel cost, $/year
- Cr: Fuel conversion factor to CO₂ emissions
- COP: Coefficient of performance
- EEDI: Energy Efficiency Design Index, gCO₂/ton-NM
- EF: Engine emission factor, kg/kWh
- ER: Emissions reduction percentage,%
- FS: Fuel saving cost, $/year
- FSE: Fuel saving cost-effectiveness, $/ton
- i: Annual interest rate, %
- L: Engine load percent in ship modes
- LCV: Lower calorific value, kJ/kg
Nader R. Ammar

Environmental and cost-effectiveness comparison of dual fuel propulsion options for emissions reduction onboard LNG carriers

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>m</td>
<td>Mass, kg</td>
</tr>
<tr>
<td>S</td>
<td>Sulfur</td>
</tr>
<tr>
<td>MCR</td>
<td>Maximum continuous rating of the engine, kW</td>
</tr>
<tr>
<td>SSDE</td>
<td>Slow speed marine diesel engine</td>
</tr>
<tr>
<td>n</td>
<td>Expected ship working years</td>
</tr>
<tr>
<td>SOx</td>
<td>Sulfur Oxides Emissions</td>
</tr>
<tr>
<td>P</td>
<td>Engine power at maximum continuous rating, kW</td>
</tr>
<tr>
<td>ST</td>
<td>Steam turbine</td>
</tr>
<tr>
<td>PI</td>
<td>Annual fuel price change percent, %</td>
</tr>
<tr>
<td>UST</td>
<td>Ultra steam turbine</td>
</tr>
<tr>
<td>SFC</td>
<td>Specific fuel consumption, g/kWh</td>
</tr>
<tr>
<td>T</td>
<td>Engine running time, h</td>
</tr>
<tr>
<td>Vref</td>
<td>Reference ship speed, knots</td>
</tr>
<tr>
<td>x</td>
<td>Fuel percentage in dual fuel engine</td>
</tr>
</tbody>
</table>

**Subscript**

DF | Dual fuel diesel engine
DO | Diesel oil
j | Type of pollutant, SOx, NOx, or CO2
m | Engine main fuel
NG | Natural gas
Reliq. | Reliquefaction

**REFERENCES**


[38] SEA-DISTANCES.ORG. Sea Distances/port distances - online tool for calculation distances between sea ports. 2018 17 October 2018]; Available from: https://sea-distances.org/.


Environmental and cost-effectiveness comparison of dual fuel propulsion options for emissions reduction onboard LNG carriers

Nader R. Ammar


Submitted: 26.11.2018. Nader R. Ammar1,2, Nader@alexu.edu.eg (corresponding author)

Accepted: 11.06.2019. 1Department of Marine Engineering, Faculty of Maritime Studies, King Abdulaziz University, 21589 Jeddah, Saudi Arabia. 2Department of Naval Architecture and Marine Engineering, Faculty of Engineering, Alexandria University 21544 Alexandria, Egypt

77
EVALUATION MODEL OF MARINE POLLUTION BY WASTEWATER FROM CRUISE SHIPS

UDC 262.3:629.541.42:628.3
Original scientific paper

Summary

In this paper, all factors that have an impact on marine pollution by sanitary wastewater from cruise ships were explored and analysed. A case study was done in which the movement of cruise ships in the Adriatic Sea was followed. Based on the results of that case study, a model of cruise ship traffic in the Adriatic Sea was developed, which, based on cruise ships itineraries in a certain period of time, gives retention times of ships in fourteen defined geographical areas of the Adriatic Sea. This model provides basic input parameters for the original evaluation model of marine pollution by wastewater from cruise ships, which is presented in this paper.

By changing operation modes of the ship in different scenarios, evaluation model enables simulation of various scenarios in order to obtain the desired or expected load values of wastewater from cruise ships in fourteen defined geographical areas of the Adriatic Sea.

Key words: cruise ships; sanitary wastewater; marine pollution from ships

1. Introduction

Cruise ship traffic is a part of a shipping industry, which is growing constantly both in number of ships and their capacity. Growth of cruise ship tourism in general is impressive - demand for cruising increased almost 50% in five-year period from 2000-2005 and then again by 50% in nine-year period from 2005 to 2014 [1] and at last 20.5% in the last five years [2]. Currently, there are 275 large cruise ships (capacity more than 500 passengers) [3]. Cruise tourism represents a high environmental impact activity with a global presence and rapid growth. It raises concerns due to its tendency of being environmental unsustainable [4]. Cruise ships represent less than 1% of the global merchant fleet yet it has been estimated that they are responsible for 25% of all waste generated by merchant vessels [5].

In 2018, 75 cruise vessels arrived in Croatian seaports and they realised 693 journeys. More than one million passengers were on board and they stayed for 1421 days in Croatia, that is, 2.1 days on average [6].

Main difference between cruise ships and all other merchant ships is in number of persons they carry. That raised question about discharge of sanitary wastewater from cruise ships and potential pollution from it.
Sanitary wastewater (WW) in order to increase efficiency and disposal is further divided into black water (BW) and grey water (GW). Black water is discharge from all types of toilets and urinals and Annex IV of MARPOL Convention regulates it. Black water may host many pathogens of concern to human health, including *Salmonella*, *Shigella*, hepatitis A and E, and gastro-intestinal viruses. Sewage contamination in swimming areas and shellfish beds pose potential risks to human health and the environment by increasing the rate of waterborne illnesses [7].

Annex IV of MARPOL Convention is governing standards for the discharge of sanitary wastewater according to three areas of navigation [8]. For each of these zones Annex IV prescribes standards of quality for discharged wastewater. However, problem lays in the fact that international legislation treats merchant ships with usually 30 persons on board same as cruise ships where the number of persons on board may exceed 8,000 people.

Grey water contains water from sinks, baths, showers, washing machines, saunas, swimming pools, sinks and water generated from washing ship’s surface and it is not recognized as pollutant by IMO so it can be discharged untreated into the sea. Grey water contains an array of pollutants from the highly acid (bleach, strong acids from some cleaning products giving water low pH) or strong alkalis (including many detergents, phosphates, whiteners, and foaming agents giving water high pH), to oil and grease, suspended solids and organic particles. In addition, degreasers found in washing up liquids and soaps strip the natural oils from fish gills making it difficult for them to breathe. In addition, researches have proven that untreated grey water contains bacteria and suspended solids concentrations equal to or exceeding black water [9].

Environmental impacts from ocean and coastal tourism include environmental degradation and pollution, destruction of habitat and ecosystem damage, and coastal pollution (wastewater, sewage and air). Such impacts have been documented in Australia, Caribbean and South Pacific Island nations, Europe, Central and South America and Polar Regions [10].

Since Adriatic Sea is semi-closed type of sea with great dependence of its countries on tourism, concern arises about pollution of the Adriatic from cruise ship sanitary water. Data about quantity and quality of discharged wastewater could not be found, therefore, a model that could evaluate this kind of pollution in the Adriatic Sea was developed.

Firstly, a case study was made in which each cruise ship was followed during its stay in the Adriatic. For each cruise ship, relevant data was noted: routes between the ports, time of entrance and exit from the Adriatic Sea and time of entrance and exit from four relevant navigation zones:

- ZONE 1: port/anchorag where ZONE 1a is Croatian port/anchorag, and ZONE 1b is foreign port/anchorag,
- ZONE 2: sea area to 3 M from the nearest land,
- ZONE 3: sea area from 3 to 12 M from the nearest land
- ZONE 4: sea area beyond 12 M from the nearest land.

After that, an evaluation model of marine pollution by sanitary wastewater from cruise ships was developed as a part of doctoral dissertation [11], which will be presented in following chapters.

The expected contribution to science in theoretical sense is proposed evaluation model that includes and recognizes all the factors that affect quantity and quality of discharged sanitary wastewater from cruise ships in the Adriatic Sea using scientific theoretical premises and scientific methods in collecting and data processing.
2. Research analysis

To create the evaluation model of marine pollution by wastewater from a cruise ship, three interdependent categories of factors affecting the pollution of the marine environment were used. These categories are:

- Movement of cruise ships in the Adriatic Sea,
- Quantity of discharged wastewater and
- Quality of discharged wastewater.

2.1 Cruise ship traffic in the Adriatic Sea

For the first category, movement of cruise ships, model of cruise ship traffic in the Adriatic Sea was developed [12], which, based on input parameters - cruise ships itineraries in a certain period of time, gives output parameters - retention times of ships in geographical areas of the Adriatic Sea. For purposes of this research, Adriatic Sea was divided on four zones of navigation and fourteen geographical zones (gz) shown in Fig. 1.

![Geographical zones of navigation in the Adriatic Sea](image)

**Fig. 1** Geographical zones of navigation in the Adriatic Sea

2.2 Quantity of discharged wastewater

Quantity of discharged wastewater depends on the following factors: capacity of ship (number of passengers and crew) and working mode of the ship.

Capacity of the ship allows us to calculate total number of persons onboard in a certain time period. Taking into account results of previous studies, in which is concluded that one person onboard a cruise ship produces 31.8 l/day of black water and 253 l/day of grey water [7], it is possible to calculate generated black and grey water onboard the ship in specified time period.

Ship in navigation uses four working modes regarding discharge of sanitary wastewater:

- MODE 1: Wastewater is not discharged, it is retained in ship’s holding tanks;
- MODE 2: Wastewater is discharged after treatment with advanced wastewater treatment systems installed on ship;
- MODE 2*: Wastewater is discharged after treatment with advanced wastewater treatment (AWT) systems installed on ship but without last treatment step – disinfection;
- MODE 3: Wastewater is discharged partially treated (comminuted and disinfected) with Marine sanitation device (MSD);
- MODE 4: Wastewater is discharged untreated directly to the sea.
Working mode of the ship is in direct correlation with the type of treatment system on board the ship because system performances must meet legal regulations for each zone of navigation.

2.3 Quality of discharged wastewater

The quality of sanitary wastewater is determined by the amount of certain substances and energy that is contained in wastewater. It should be noted that, depending on their source on ship, all wastewater contains various amounts and concentrations of waste products that are characterized with respect to their physical, chemical and microbiological properties [7, 13]. There are a great number of indicators, but with their impact on the marine environment and human health following wastewater, quality factors are distinguished:

- feacal or thermotolerant coliforms, FK,
- total suspended solids, TSS,
- 5-day biochemical oxygen demand, BOD₅ and chemical oxygen demand, COD,
- pH value,
- chlorine residual,
- total nitrogen and
- total phosphorus.

A comparison of quality of discharged sanitary wastewater from the treatment system considering quality criteria and minimum requirements of Annex IV of MARPOL Convention is shown in Table 1.

Table 1 Comparison of quality of discharged sanitary wastewater from the treatment system considering quality criteria and minimum requirements of MARPOL Convention [7, 8, 14, 15]

<table>
<thead>
<tr>
<th>Discharge quality for</th>
<th>Type of wastewater</th>
<th>operational mode/ zone</th>
<th>Discharged wastewater quality factors</th>
</tr>
</thead>
<tbody>
<tr>
<td>MSD</td>
<td>black water</td>
<td>MODE 3</td>
<td>FK/100 ml, mg/l, TSS, mg/l, COD, μg/l</td>
</tr>
<tr>
<td></td>
<td></td>
<td>MODE 4</td>
<td>2,040,000, 133, 627, 1,070</td>
</tr>
<tr>
<td></td>
<td>gray water</td>
<td>MODE 4</td>
<td>636,000,000, 526, 704, 372</td>
</tr>
<tr>
<td>AWT system</td>
<td>black and gray water</td>
<td>MODE 2</td>
<td>36,000,000, 1,140, 704, 372</td>
</tr>
<tr>
<td></td>
<td></td>
<td>MODE 2*</td>
<td>25,500, 7.99, 4.49, 338</td>
</tr>
<tr>
<td></td>
<td></td>
<td>MODE 4</td>
<td>103,000,000, 526, 704, 372</td>
</tr>
<tr>
<td>MARPOL Annex IV zones</td>
<td>black water</td>
<td>zone to 3 M old¹</td>
<td>250, 50, 100, 500</td>
</tr>
<tr>
<td></td>
<td></td>
<td>zone to 3 M</td>
<td>100, 25 · Qᵢ/Qₑ², 35 · Qᵢ/Qₑ², 500</td>
</tr>
<tr>
<td></td>
<td>black water</td>
<td>zone 3-12 M</td>
<td>Discharge of comminuted and disinfected black water</td>
</tr>
<tr>
<td></td>
<td>black water</td>
<td>zone outside 12 M</td>
<td>Discharge of untreated black water at a moderate rate when the ship is en route and proceeding at not less than 4 knots.</td>
</tr>
</tbody>
</table>

¹ it refers to ships with wastewater treatment systems installed prior to 1.1.2010.
² dilution compensation factor Qᵢ/Qₑ is used to take account of dilution, where Qᵢ is influent, liquid containing sewage, gray water or other liquid streams and Qₑ is effluent, treated wastewater produced by the sewage treatment plant.

Table 1 shows that treated black water from MSD has about 2 million faecal coliform while treated sanitary wastewater from AWT plant has 14.5 faecal coliforms in 100 ml of discharged wastewater. The values of other quality factors of sanitary wastewater are also several times higher at the exit of the MSD compared to the output from the AWT system. It
can be concluded that there is great disproportion in quality of discharged wastewater directly related to the type of wastewater treatment system installed on cruise ships. Defined quality can be even worse because of insufficient familiarisation of engineering crew with complex wastewater treatment plants unique for cruise ships [16].

3. Model

Evaluation model of marine pollution by wastewater from cruise ships enables us to calculate quantity and quality of discharged wastewater in specified geographic areas of the Adriatic Sea considering selected operational modes in those areas. Calculation steps are shown in flow chart in Fig. 2.

![Flow Chart](image)

**Fig. 2** Quality and quantity of discharged wastewater calculation flow chart

The model consists of the mandatory input parameters without which calculation cannot be performed:
- name of the ship with number of persons onboard,
- retention time in geographic zones $t_{gz}$,
- type of wastewater treatment system (AWT or MSD),
- selection of operational modes in each navigation zone.

By knowing the retention time in navigation zones for a particular route $t_{gz}$ and the number of persons on board a cruise ship i.e. its capacity $K$ it is possible to calculate quantity of generated black and gray wastewater [12]:

$$
GBW = \frac{K \cdot F_{BW} \cdot t}{1000} \quad [\text{m}^3] 
$$

$$
GGW = \frac{K \cdot F_{GW} \cdot t}{1000} \quad [\text{m}^3] 
$$

(1)  

(2)
where:

\( GBW \) is quantity of generated black water,

\( GGW \) is quantity of generated gray water,

\( K \) is average number of persons on the ship,

\( F_{BW} \) is black water constant and it is 1.325 l/person/h,

\( F_{GW} \) is gray water constant and it is 10.54 l/person/h,

\( t \) is retention time in sea area [h].

Knowledge of the type of wastewater treatment system installed on the ship enables us to recognize the quality of discharged sanitary wastewater according to operational mode of the ship (Table 1). Selection of operational modes in Annex IV navigation zones is directly depended on the system type on the ship. Fig. 3 shows all possible combinations of operational modes in navigation for AWT and MSD ships, where the minimum operational mode that can be selected is one legally allowed by Annex IV of MARPOL Convention.

Operational mode for black and gray water is separately chosen in MSD, while the AWT system collects and processes black and gray water together so the selection of operational mode applies to both. By changing operational modes of the ship in different scenarios evaluation model of marine pollution allows simulation of different scenarios in order to obtain the desired or expected values of specific sea area load with wastewater from cruise ships. That enables the evaluation of current legislation and identification of critical areas of the sea regarding wastewater pollution from cruise ships. As seen in Fig. 3, there are many possible scenarios; however, the operation logic is the same.

In zones selected for mode 1 generated wastewater is collected in holding tanks. Rate of discharge from holding tanks is an optional input parameter. The ship at sea navigates through specified geographic areas in the order defined by the route it is on and retention time \( t_{gz} \) in each of these areas provided by the cruise traffic model. Once the ship enters in the first zone in which discharge is allowed within the selected scenario, discharge from holding tanks can begin following one of these principles:
a) If the rate of discharge (m$^3$/h) is defined, holding tanks will be discharged in zones selected for discharging at that rate until they are empty.

b) If the rate of discharge is not defined, the tanks will be discharged continuously in zones in which discharge is allowed with intention of emptying them until the next port. Discharge will be carried at average discharge rate, which is obtained as the ratio of the quantity of wastewater in holding tanks (m$^3$) and retention time (h) in zones of allowed discharge on the route. If ship does not enter any zones of allowed discharge by the selected scenario on a route (e.g. the whole voyage takes place in zone 2) than the quantity of generated wastewater on this route is accumulated and transferred for discharge to the next route.

The scenario that strictly follows the rules of MARPOL Annex IV is called Scenario 1. Scenario 1 for MSD is shown in Fig. 4 and for AWT systems in Fig. 5.

Ships with MSD separately collect black and gray water. Only black water is processed while the gray is discharged without processing. MARPOL Convention does not recognize gray water as a pollutant and there are no regulations for the discharge of the same. Therefore, scenario 1 of MSD stipulates detention of gray water in holding tanks only while the ship is in port and discharge in all other zones. Black water is retained in holding tanks in zones 1 and 2 while it is discharged treated in zone 3 and untreated in zone 4. Discharge from holding tanks starts in zone 3 as shown in flow diagram in Fig. 4.

![Flow Diagram](image-url)  
**Fig. 4** Scenario 1 flow chart for MSD [11]

AWT system handles both black and gray water so in this case regulations for black water are applied also to the gray water because it is mixed with black. According to Annex IV of MARPOL Convention ships with AWT systems which have received Certificate of Type Approval by the Administration can continuously discharge sanitary wastewater processed with AWT system (mode 2). However, in zone 3 wastewater can be discharged partially treated (without UV disinfection - mode 2*), and in zone 4 untreated (mode 4) as shown in the flow diagram in Fig. 5.
Fig. 5 Scenario 1 flow chart for AWT system [11]

4. Simulation with model

Through the presented model it is possible to evaluate marine pollution by wastewater from cruise ships. According to the cruise traffic model [12], in the year of case study, cruise ships have spent: 1454 h in zone 2 which is approximately 61 days; 5087 h in zone 3, approximately 212 days and 19351 h in zone 4, approximately 806 days. It is important to note that zone 4 includes the entire Adriatic Sea outside the territorial sea of the Republic of Croatia which means that it also includes territorial waters of Italy, Slovenia, Montenegro and Albania. Since the focus of the research was pollution of the Croatian part of the Adriatic Sea retention times in zones 2 and 3 of these countries were included in zone 4 of the model.

Retention times in geographical areas $t_{gz}$ provided from cruise traffic model on basis of cruise ship itineraries together with the average capacity of the ships\(^1\) enables us to calculate generated black (GBW\(_{gz}\)) and gray water (GGW\(_{gz}\)) in each geographical area, using formula (1) and (2), which is presented in Table 2. This distribution of generated black and gray water presents entry parameters for simulation of different scenarios regarding selection of operational modes in evaluation model.

For purposes of this case study it is assumed that cruise ships do not discharge wastewater during their stay in the harbor/anchorage. Since only AWT systems can discharge treated wastewater in zone 2, part of the wastewater generated in zone 2 will be discharged in the first zone 3 that follows. In addition, the amount of wastewater generated in the Croatian port will be discharged in zone 2 (AWT systems) or in zone 3 (MSD systems), and generated wastewater in a foreign port will be discharged in the corresponding zone 4. Gray water of ships with MSD is discharged untreated always, except in the port when it is stored in holding tanks.

---

\(^1\) Average capacity of ships in this case study is calculated in respect of retention time of each ship in the Adriatic Sea to their capacity and it is 2909 persons.
Current world ratio regarding wastewater treatment system on large cruise ships is: 55% of ships with AWT systems and 45% of ships with MSD. However, in the year of the case study this ratio in Adriatic Sea was following: 52.4% of total number of cruise ships that entered Adriatic Sea had MSD. If we also take into account number of persons onboard we get this result: 52.7% of wastewater is processed on ships with AWT systems and 47.3% of wastewater is processed on ships with MSD systems. Guided by this ratio, the approximate quantity and quality of discharged wastewater in each geographical area in the case study year can be calculated as shown in Table 3. For the quality of discharged wastewater, it was assumed that ships follow the regulations of MARPOL Annex IV.

Summing quality of discharged wastewater (DW_gz) from AWT and MSD results in total load of geographical areas in a year of research, graphically shown in Fig. 6. It should be noted that all calculations used the average retention times on each route, average capacity of ships and therefore results may vary from the actual values of pollution. However, the results are sufficiently accurate for general picture of pollution of the marine environment in the Adriatic Sea. Accurate results can be obtained using Kruzeri, software developed for easier calculations from mentioned models. Software uses accurate information for each cruise ship: capacity of the ship, the time of navigation on routes regarding ships itinerary and exact treatment system that is installed on the ship.
### Table 3 Distribution of estimated quantity and quality of discharged black (DBW) and gray water (DGW) from cruise ships to the geographical zones of navigation in the one-year case study

#### AWT

<table>
<thead>
<tr>
<th>Geographic zone of the Adriatic</th>
<th>Quantity of discharged wastewater</th>
<th>Operational mode for black/gray water</th>
<th>Quality factors of discharged black/gray water</th>
<th>Quality of discharged wastewater</th>
</tr>
</thead>
<tbody>
<tr>
<td>ZONE 1</td>
<td>DBW</td>
<td>DGW</td>
<td>MODE 1</td>
<td>/</td>
</tr>
<tr>
<td>Croatian port</td>
<td>166</td>
<td>0</td>
<td>25500</td>
<td>7.99</td>
</tr>
<tr>
<td>foreign port</td>
<td>1215</td>
<td>9668</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ZONE 2</td>
<td>1425</td>
<td>11336</td>
<td>MODE 2</td>
<td>14.5</td>
</tr>
<tr>
<td>Istra north</td>
<td>280</td>
<td>2230</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Zadar - Unije</td>
<td>806</td>
<td>6413</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Solta - Korcula</td>
<td>999</td>
<td>7949</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lastovo and marginal sea</td>
<td>1425</td>
<td>11336</td>
<td>MODE 2*</td>
<td>25500</td>
</tr>
<tr>
<td>Mljet</td>
<td>1415</td>
<td>11257</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dubrovnik - Kotor</td>
<td>2584</td>
<td>20557</td>
<td></td>
<td></td>
</tr>
<tr>
<td>north Adriatic</td>
<td>27641</td>
<td>219906</td>
<td>MODE 4</td>
<td>103 mil.</td>
</tr>
<tr>
<td>ZONE 3</td>
<td>27526</td>
<td>218980</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Istra north</td>
<td>166</td>
<td>0</td>
<td>25500</td>
<td>7.99</td>
</tr>
<tr>
<td>Zadar - Unije</td>
<td>1215</td>
<td>9668</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Solta - Korcula</td>
<td>999</td>
<td>7949</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lastovo and marginal sea</td>
<td>1425</td>
<td>11336</td>
<td>MODE 2*</td>
<td>25500</td>
</tr>
<tr>
<td>Mljet</td>
<td>1415</td>
<td>11257</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dubrovnik - Kotor</td>
<td>2584</td>
<td>20557</td>
<td></td>
<td></td>
</tr>
<tr>
<td>south Adriatic</td>
<td>27526</td>
<td>218980</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

#### MSD

<table>
<thead>
<tr>
<th>Geographic zone of the Adriatic</th>
<th>Quantity of discharged wastewater</th>
<th>Operational mode for black/gray water</th>
<th>Quality factors of discharged black/gray water</th>
<th>Quality of discharged wastewater</th>
</tr>
</thead>
<tbody>
<tr>
<td>ZONE 1</td>
<td>DBW</td>
<td>DGW</td>
<td>MODE 1</td>
<td>/</td>
</tr>
<tr>
<td>Croatian port</td>
<td>149</td>
<td>0</td>
<td>204 mil.</td>
<td>133/</td>
</tr>
<tr>
<td>foreign port</td>
<td>1222</td>
<td>8678</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ZONE 2</td>
<td>1240</td>
<td>2002</td>
<td>MODE 3/4</td>
<td>2.04 mil.</td>
</tr>
<tr>
<td>Istra north</td>
<td>1222</td>
<td>8678</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Zadar - Unije</td>
<td>1240</td>
<td>2002</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Solta - Korcula</td>
<td>1224</td>
<td>6170</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lastovo and marginal sea</td>
<td>1279</td>
<td>10175</td>
<td>MODE 3/4</td>
<td>2.04 mil./</td>
</tr>
<tr>
<td>Mljet</td>
<td>1739</td>
<td>13833</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Dubrovnik - Kotor</td>
<td>6407</td>
<td>10104</td>
<td></td>
<td></td>
</tr>
<tr>
<td>north Adriatic</td>
<td>4894</td>
<td>18450</td>
<td>MODE 4</td>
<td>616 mil./36 mil.</td>
</tr>
<tr>
<td>middle Adriatic</td>
<td>6202</td>
<td>49335</td>
<td></td>
<td></td>
</tr>
<tr>
<td>south Adriatic</td>
<td>24706</td>
<td>196542</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Results of the evaluation model presented in Fig. 6 indicate that the zone 2 is highly affected with cruise ship traffic. Although this zone is protected from discharge of untreated and inadequately treated black water with Annex IV of MARPOL Convention, the problem is in discharge of gray water, which is not recognized as a pollutant (unless it is mixed with black water). The values in zone 4, or outside the territorial sea of the Republic of Croatia, are large as expected, since there are no requirements for wastewater treatment prior to their discharge. Attention should be paid to the northern Adriatic, which is, due to its small depth, particularly sensitive to all kinds of pollution and in which problem of eutrophication already occurred. Wastewater pollution in zone 3 is shown in more detail in the following chapter.

88
4.1 Identification of critical sea areas regarding sanitary wastewater pollution from cruise ships

Application of case study data and developed models for evaluation of marine pollution by wastewater from cruise ships in the Adriatic Sea provided quality and quantity of discharged wastewater in fourteen defined geographical areas. Detailed analysis of results for zone 3 allows us to identify critical areas regarding marine pollution of the Adriatic Sea by wastewater. We will consider pollution in two cases:

- Case 1: gray water from ships with MSD is discharged into zone 2 respecting Annex IV of MARPOL Convention.
- Case 2: gray water from ships with MSD is discharged into zone 3, which directly follows zone 2 respecting common practice of cruise ships.

Quantity and quality of discharged wastewater in case 1 is already shown in Fig. 6.

Fig. 6 Case 1 quality and quantity of discharged wastewater in the Adriatic Sea in one-year period considering treatment system and geographic areas of the Republic of Croatia

Fig. 7 Case 2 quality and quantity of discharged wastewater in the Adriatic Sea in one-year period considering treatment system and geographical areas of the Republic of Croatia
For the second case the quantity of gray wastewater generated in zone 2 is divided into zones 3 directly following zone 2 considering movement of the ships on routes. Thus, the quantity of discharged gray water increased in some geographical areas. The largest increase, by as much as 90%, is in the area of Mljet, followed by: Zadar - Unije with an increase of 87%; Vis - Lastovo with 59%; Dubrovnik - Kotor with 54% and Šolta - Kornati with an increase of 43%. Fig. 7 graphically shows the quality and quantity of discharged black and gray water in zone 3 of the territorial sea of the Republic of Croatia in case 2.

Critical areas, based on the results of the evaluation model, are:
- Mljet area,
- Dubrovnik – Kotor area and
- area of Lastovo.

In both cases, the largest quantity of wastewater was discharged in the Mljet area which stands out as the most critical area regarding marine pollution by wastewater from cruise ships. It is followed by the Dubrovnik - Kotor area and the area of Lastovo since it combines the contamination of the Lastovo area (DW36) and Vis - Lastovo area (DW34). Marginal sea area is also burdened by discharged wastewater, but is not considered as a critical area because it includes narrow sea area of the territorial sea in the central Adriatic near the state border and the sea area around the island of Palagruža. Because of the distance from the nearest land and the position of the area on high seas, it is concluded that discharged wastewater in this area is not critical to the marine environment.

4.2 Draft guideline for relocation of navigation routes on the larger distance from the Croatian coast

National systems have generally underdeveloped legislative and executive mechanisms to control and manage pollution. Subsequently they are not in a position to respond to the challenges of implementing the six annexes of MARPOL and other relevant conventions [4]. However, pollution can be reduced in different manner by relocation of navigation routes on the larger distance from the coast.

The direct reject of wastewater is one of the major factors of coastal and marine environment degradation, because it is discharged directly into sea with a high concentration of pollutants. Thus, if this discharge is not controlled, the effluent may return to the coastal regions without being sufficiently diluted; so it can contaminate areas for farming, fishing grounds or beaches [17].

So, marine pollution by wastewater from cruise ships has the greatest impact in the coastal area of the sea. Moving away from the mainland reduces the impact of pollution because the wastewater with its quantity and quality is discharged in larger volumes of the sea.

In the most common route in the case study it was noted that most of the ships in navigation between Dubrovnik and Venice tends to sail in close coastal zone (2-5 M). Therefore, wastewater is discharged in proximity of the Croatian National Park Mljet and Nature Park Lastovo. Only a small part of the ships chose to relocate the route outside the territorial sea of Croatia. In Fig. 8, the solid line shows the most common navigation route of cruise ships from Dubrovnik and Kotor to the ports of northern Adriatic and vice versa. The dotted lines mark proposed relocated navigation route used by some ships. It is considered necessary to introduce recommended route or even vessel routeing systems for cruise ships that sail between the ports of the eastern coast of the southern Adriatic (Dubrovnik, Kotor, Durres) and the northern Adriatic (Zadar, Koper, Trieste, Venice, Ravenna ...). With the relocated
routes, cruise ships extend their travel for a negligible 14 M but wastewater load of the protected nature of the Republic of Croatia is reduced to a minimum.

Fig. 8 Relocation of navigation routes on a greater distance from shore

Cruise ships are a special category of ships when choosing the route and speed of navigation on their travels. While merchant ships choose the optimal speed of navigation (regarding fuel prices and freight) and the optimal route (due to the length of time, weather and, of course, navigation restrictions), cruise ships adjust the route and the speed of navigation according to scheduled time of arrival in the next port of call. For these reasons, it is not necessary to choose the shortest possible route between two ports, which enables the realization of the proposed guideline without compromising cruising tourism in the Republic of Croatia.

5. Conclusion

Marine pollution from ships is always actual issue because it represents a major threat to the marine environment. This problem is particularly pronounced in closed type of seas like the Adriatic Sea and in countries in which economy is largely based on tourism, therefore, the purity of the sea, such as the Republic of Croatia. Traffic of cruise ships is in continuous growth. New cruise ships are built with constant increase in capacity, which now exceeds 8,000 people. Previous studies on discharge of wastewater from cruise ships deal mainly with quality indicators and treatment technologies of the same. However, there is no model that could evaluate pollution of particular sea areas with wastewater from cruise ships. Therefore, the scientific contribution of this paper is presented evaluation model of marine pollution by wastewater from cruise ships that allows the calculation of quantity and quality of discharged wastewater in a specified geographical areas within a certain period of time in different scenarios. By changing the input parameters and scenarios it is possible to obtain wanted or expected output parameters whose comparison can evaluate the current legal regulations, as well as it can guide us for future legal requirements regarding wastewater pollution.

Using the evaluation model in the scenario that follows the legal requirements of MARPOL’s Annex IV critical areas regarding marine pollution by wastewater from cruise ships were identified: Mljet area, Dubrovnik - Kotor area and the area of Lastovo. Evaluation model of marine pollution by wastewater from cruise ships allows us assessment of the level of threat
to the marine environment. That can guide authorities to future requirements related to the discharge of sanitary wastewater from cruise ships, which will consequently have an impact on the protection of marine environment, and therefore the preservation of tourism as strategic branch of economy and quality of life of coastal populations.

REFERENCES


Submitted: 02.05.2019. dr. sc. Tina Perić, tina.peric@pfst.hr
University of Split, Faculty of Maritime Studies, Rudera Boškovića 37, Split

Accepted: 18.06.2019. dr. sc. Vice Mihanović, vice.mihanovic@portsplit.hr
Split Port Authority, Gat Sv. Duje 1, Split

dr. sc. Nikola Račić, nikola.racic@pfst.hr
University of Split, Faculty of Maritime Studies, Rudera Boškovića 37, Split
STUDY OF CONTINUOUS ICEBREAKING PROCESS WITH
COHESIVE ELEMENT METHOD

UDC 629.5.016.5:629.561.5
Original scientific paper

Summary

Accurate simulation of the continuous icebreaking process in level ice is crucial for the
design of icebreakers. The crushing and bending failures of ice sheet, as well as the rotating,
sliding and accumulating of ice cusps broken from the ice sheet constitute a complex system
for the icebreaking process. In this paper, cohesive element method is combined with an
elastoplastic softening constitutive model to simulate continuous icebreaking process in level
ice. Firstly, the elastoplastic softening constitutive model in modelling ice local crushing is
calibrated by simulating the ice cone crushing tests. Three different softening laws are
proposed and their effects on simulation results are evaluated by comparing with the
experimental data. Then, the continuous icebreaking process in level ice is simulated by
cohesive element method. The regular tri-prism mesh is applied to ice bulk elements to realize
the random propagation of crack. The mesh dependency study is carried out, and the
simulation results are validated by comparing with model test results in both of time domain
and frequency domain. The ice failure patterns during continuous icebreaking process are also
compared between the simulated and experimental results. Finally, the influences of ship
velocity on ice resistance and ice failure patterns are investigated by numerical methods and
semi-empirical formulas.

Key words: icebreaking; cohesive element; constitutive law; ice resistance; Ice failure
patterns; ship velocity

1. Introduction

With the global warming, ice cap in the Arctic has been shrinking year by year. Consequently,
there will be more and more voyages across the Arctic. Accurate prediction of
ice resistance on the vessels is necessary for their design and safe navigation in the ice-
covered waters. When a ship is traveling in level ice, the ice sheet will experience breaking
failure, which mainly includes local crushing and bending failures. The generated ice cusps
rotate, slide and accumulate around the hull, which leads to additional ice resistance on the
ship. Based on the research by Croasdale and Cammaert [1], the proportion of accumulation forces in the total ice loads may reach up to 70%. Thus, it is important to simulate the whole ship-ice interaction process for a better prediction of ice resistance.

Since the mechanism of icebreaking process has not been fully understood yet, many researchers have applied empirical formulae to predict ice resistance since 1970 [2-4]. In these formulas, the empirical coefficients were derived by fitting the data from the model tests and full-scale trials. Due to the limited data, the expressions contain only a few empirical constants and their applications are limited. The semi-empirical approach with theoretically derived mechanical formulae can consider more details of ice-structure interaction. Lindqvist [5] summed up the sea trials in the Bay of Bothnia and divided ice resistance into crushing, bending and submersion components. Lindstrom [6] adopted fully derived mechanical formulas and manifested the physical concepts of ice-hull interaction. However, Lindstrom’s model only considered the bending failure of ice. Based on a series of full-scale tests in Baltic Sea, Riska et al. [7] presented a formula for ice resistance by improving the empirical formulas proposed by Lindqvist [5]. The Riska formula considered ice resistance as a function of ship main dimensions.

With the rapid promotion of computational power, discrete element method (DEM) and finite element method (FEM) have been widely used to model the ice-structure interactions. Paavilainen et al. [8] applied DEM to simulate ice pile-up process against an inclined structure. Zhan et al. [9] used a discrete element numerical modelling program, DECICE, to calculate interaction forces between ship and pack ice pieces. Wang et al. [10] developed a collision model for nonlinear dynamic finite element analysis on a LNG ship and a crushable ice using commercial code DYTRAN. Liu et al. [11] conducted nonlinear finite-element analysis by using a commercial code LS-DYNA to assess the internal mechanics of both icebergs and ship body. A plastic model was set for icebergs, where an empirical criterion was used to detect the failed ice element. Shi et al. [12] applied an elastic-perfect-plastic materiel model to simulate the collision between a FPSO ship and an iceberg. The gradient temperature effect was considered, where “Tsai-Wu” yield surface was applied in the simulations.

The interaction between the ice cusps fragmented from intact ice and the ship hull should not be ignored, as it affects both the ice failure patterns and ice resistance on the structure. The traditional FEM is based on the assumption of continuum material, and it is difficult to simulate the crack branching and the formation of the rubble accumulation. Paavilainen et al. [13,14] combined FEM and DEM to simulate the interaction of an inclined plate with ice sheet. FEM was used to simulate continuous failure of ice sheet while DEM was used to simulate ice rubble pile. The model presents great advantages in modelling the ice accumulating process and bulking behaviour of force chains, and provides good idea for the numerical investigations on the ice-structure interaction. In this paper, a 3D numerical model is established by combining cohesive element method (CEM) with an elastoplastic softening constitutive model to simulate continuous icebreaking process in level ice. The local crushing of ice is presented by the elastoplastic softening constitutive model of ice material, while the cracks caused by bending failures are realized by cohesive element method. Firstly, the proposed numerical models are introduced and the ice elastoplastic constitutive model is calibrated through simulating an ice crushing experiment. Then the continuous icebreaking process is simulated by combining the elastoplastic constitutive model and cohesive element method. The mesh dependency study is carried out and the simulated ice resistance and ice failure patterns are compared with model test results. Finally, the effects of the ship velocity on ice resistance and ice failure patterns are investigated as well.
2. Numerical models

2.1 Elastoplastic softening model

When a marine structure interacts with ice sheet, local crushing happens around the contact region. However, the mesh applied in the FEM simulation is always too coarse to present the feature of local crushing, i.e., such microscopic failure is difficult to be explicitly modeled in the simulation. Hilding et al. [15] proposed a numerical approach which took local crushing as a product of multiple micro-crack branching in the contact zone. An elastoplastic linear softening constitutive model, taken as a homogenization approach, was applied in the numerical model to present the influence of the arising and propagation of micro-cracks on ice material performance. The corresponding hardening curve is shown in Fig. 1.

![Fig. 1. Elastoplastic constitutive model given by Hilding et al. [15]](image)

Before reaching the crushing initial point (yield stress $\sigma_y$), the ice material is elastic. After reaching the crushing initial point, ice material starts to undergo a linear softening process. The macroscopic effect of local crushing is that ice is becoming softened with the development of micro-cracks whose amount is proportional to the deformation. When the plastic strain reaches crushed plastic strain $\varepsilon_c$, ice element is totally crushed and it behaves as a viscous fluid. The corresponding effective stress at this point is crushed effective stress $\sigma_c$, whose value should be relatively low which means ice material cannot bear external load at this stage. Finally, a failure strain $\varepsilon_f$ is set to avoid the ice elements to be excessively distorted. It should be noted that the linear behavior during the softening phase was a rough assumption by Hilding et al. [15]. The diverse softening phases could be used to represent different stress-strain relationships.

2.2 Cohesive element method

Cohesive element method (CEM) is an extension of cohesive zone model (CZM) in FEM simulations. CZM was firstly proposed by Hillerborg et al. [16] to model the local crack propagation of an unreinforced concrete beam. In their study, the fracture around the crack tip was simulated and the behavior of cohesive elements was controlled by a traction-separation law. CZM was applied to analyzing the behavior of ice by Mulmule and Dempsey [17,18]. Afterwards, some other researchers [19-23] applied CEM to simulate the interactions between level ice and marine structures.

In the framework of CEM, the cohesive elements are inserted into every internal face between the neighboring finite bulk elements, as shown in Fig. 2. Theoretically, the cohesive
element is a zero-thickness element which would respond to tension and shear on the cohesive surface. In practical simulation, the cohesive elements are defined as ultrathin elements for the numerical accuracy. The bulk elements and the cohesive elements share nodes, thus deformation and stress could transmit directly between them. When the separation of cohesive element reaches the critical value, the cohesive element is considered to be completely damaged and is deleted from the simulation. Consequently, the explicit crack is created along the cohesive element faces.

![Fig. 2. Illustration of ice bulk elements and cohesive elements](image)

Traditionally, 8-node three dimensional brick mesh, i.e., hexahedral mesh, is applied to bulk element for its good numerical property. However, the usage of hexahedral mesh will lead to a “zig-zag” crack path, which is longer than the actual path (e.g., $\sqrt{2}$ times for a 45° crack). This will cause an extra dissipation of fracture energy and produce a non-negligible influence on the simulation results. In this paper, the 6-node regular tri-prism mesh as shown in Fig. 2 is applied to the bulk element to attenuate this effect. This kind of mesh has shown great advantages in modeling bending fracture and random propagation path of crack during the ice-conical structure interaction in the numerical research by Wang et al. [22].

The behaviors of cohesive elements are mainly controlled by three factors: fracture energy, fracture stress and traction-separation law (TSL) curve. Fracture energy is the dissipated energy by the failure of the cohesive elements and fracture stress is the maximum stress obtained in this process. The TSL curve represents the relation between the crack separation and the cohesive force, which can be seen as the constitutive law of cohesive elements. Fracture energy can be obtained by fracture mechanic tests. Based on the in-situ tests carried out by Dempsey et al. [24], the size-independent fracture energy of multi-year ice was within the range of 23–47J/m². Normally, the fracture stresses are substituted by tensile strength in mode I fracture and shear strength in mode II fracture. Based on the test data from previous work, Timco and Weeks [25] obtained the tensile strength ranged from 0.2 MPa to 0.8 MPa, while the shear strength was within 0.55–0.9 MPa. According to the research by Cornec et al. [26] and Wang et al. [22], the shape of TSL curve brings limit effects on simulation results as long as the value of fracture energy is correct. The most commonly used traction-separation law in CEM is bilinear law, whose detailed expressions could refer to Wang et al. [23].
3. Numerical calibration of the constitutive model

Hilding et al. [15] roughly assumed the effective stress and plastic strain obey a linear softening relationship in the elastoplastic softening constitutive model. However, there are various softening laws could be applied and the effects of different shapes of softening laws should be evaluated. Fig. 3 shows three kinds of softening laws presented in this paper, i.e., linear law, exponential law and stepped law, which defines different ice softening process with the development of micro-cracks. A numerical study on ice cone crushing tests is performed to evaluate the effects of different softening laws on the simulation results, and also calibrates the proposed constitutive model in modeling ice local crushing failure. It should be noted that the cohesive element method is not introduced in this section.

The numerical setup is based on the ice cone crushing tests carried out by Kim et al. [27]. As shown in Fig. 4, a steel indenter impacts on an ice cone with constant velocities. The diameter of ice cone is \( D = 250 \text{mm} \), and the inclined angle is 30\(^\circ\). The width and the length of the steel plate are 400mm\( \times \)400mm. The bottom of ice cone is fixed and a velocity boundary is set on the rigid plate by the command of Prescribed-Motion to keep the plate impacting against the ice cone with two constant velocities, 1mm/s and 100mm/s. Based on the experimental data, the material parameters of ice cone are set as listed in Table 1. Three softening laws shown in Fig. 3 are applied in the ice elastoplastic constitutive model. The mesh size of ice cone is set as \( l_m = 2.5 \text{mm} \), which is determined by a prior mesh dependency study to obtain the convergent results. The material of the steel indenter is set as rigid without consideration of deformation, whose parameters are also shown in Table 1. Because the stiffness of ice cone is greatly different from that of rigid plate, the contact algorithm between ice cone and rigid plate adopts command of CONTACT-ERO-DING-NODES-TO-SURFACE, and the friction coefficient between ice and rigid plate is set as 0.2.
Table 1 Material parameters of ice cone and rigid plate

<table>
<thead>
<tr>
<th>Items</th>
<th>Value</th>
<th>Items</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m³)</td>
<td>910</td>
<td>Density (kg/m³)</td>
<td>7850</td>
</tr>
<tr>
<td>Elastic modulus (GPa)</td>
<td>5</td>
<td>Elastic modulus (GPa)</td>
<td>200</td>
</tr>
<tr>
<td>Poisson ratio</td>
<td>0.3</td>
<td>Poisson ratio</td>
<td>0.3</td>
</tr>
<tr>
<td>Yield stress σᵧ (MPa)</td>
<td>15</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Crushed stress σc (MPa)</td>
<td>0.1</td>
<td>Crushed strain εc</td>
<td>0.4</td>
</tr>
<tr>
<td>Failure strain εf</td>
<td>0.5</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 5 gives the comparison between the numerical results and the experimental results from Kim et al. [27] at different impact velocities. As shown in the figure, all of the load curves present “stepped” rising tendency due to the continuous impacting and local crushing between ice cone and rigid plate. The gentle load drops appeared in the simulations are caused by the softening and failure of ice elements, which also can be observed in the experimental results. The abnormal large drops of experimental loads at the displacement of 9mm and 18mm are caused by start-stop discontinuity of the indenter motion, which was explained by Kim et al. [27]. The mean values and standard deviations of calculated and measured impact forces are given by Table 2, in which the errors between the calculated and measure results are also provided. Among the three softening laws, the linear law obtains the closest results to the experimental results, while the stepped law derives higher loads and exponential law derives lower loads. In addition, the load fluctuations with stepped law are obviously heavier than those with other two laws. This result is more obvious under high impact velocity. This is because that the stress varies unevenly with the strain and even experiences sudden drops at some strain points, which ultimately induces larger fluctuations in the load curves.
Table 2 Comparison between simulated results and measured results

<table>
<thead>
<tr>
<th>Softening law</th>
<th>Impact velocity</th>
<th>Mean</th>
<th>StDev</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linear law</td>
<td>1mm/s</td>
<td>9.90kN (-7.5%)</td>
<td>8.37kN (-5.1%)</td>
</tr>
<tr>
<td>Exponential law</td>
<td></td>
<td>7.03kN (+34.3%)</td>
<td>6.00kN (-32.0%)</td>
</tr>
<tr>
<td>Stepped law</td>
<td></td>
<td>12.6kN (+17.8%)</td>
<td>10.7kN (+21.3%)</td>
</tr>
<tr>
<td>Model test</td>
<td></td>
<td>10.7kN (-)</td>
<td>8.82kN (-)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Softening law</th>
<th>Impact velocity</th>
<th>Mean</th>
<th>StDev</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linear law</td>
<td>100mm/s</td>
<td>12.2kN (-0.81%)</td>
<td>10.7kN (-6.14%)</td>
</tr>
<tr>
<td>Exponential law</td>
<td></td>
<td>8.64kN (-29.8%)</td>
<td>7.50kN (-34.2%)</td>
</tr>
<tr>
<td>Stepped law</td>
<td></td>
<td>15.6kN (+26.8%)</td>
<td>13.9kN (+21.9%)</td>
</tr>
<tr>
<td>Model test</td>
<td></td>
<td>12.3kN (-)</td>
<td>11.4 (-)</td>
</tr>
</tbody>
</table>

Fig. 6 presents the results of the pressure-area relationship of the numerical and experimental results. The pressure is obtained by dividing impact force with nominal contact area, which is the contact area projected onto the rigid plate. It can be seen that the simulation results capture the same trend of the experimental results, i.e., the rapid decreasing trend of the pressure with the increase of nominal contact area. The general shapes of the scattered point distributions are close to the exponential form. The similar pressure-area relationship was also observed in the research by Masterson et al. [28] on the various ice-structure impacting model tests. The fluctuation of the pressure is most serious when the contact area is less than 1000mm². With the continuous increases of the contact area, the variation of the pressure tends to be gentle. The measured pressure arises abnormal sudden drops at area of 900mm² and 3000mm², which is caused by start-stop discontinuity of the indenter motion in the model tests. The fitted curves with the format of $y = ax^b$ for pressure-area relationship results are shown in Fig. 7. The simulation results with the linear law fit best with the experimental results, while the pressure obtained by stepped law is higher and the pressure obtained by exponential law is lower than the measured pressure.
Fig. 6. Comparison of pressure-area relationships between simulations and experiments

Fig. 7. Comparison of fitted pressure-area curves between simulations and experiments
4. Simulation of continuous icebreaking process

The numerical simulation of the continuous icebreaking process is presented in this section. The cohesive element method is introduced in this section to simulate the bending failure of level ice. In addition, the linear softening elastoplastic model, which obtains closest results to the experimental results, is chosen as the constitutive model of ice bulk element to simulate the local crushing of ice sheet.

4.1 Geometry and numerical setup

The double-hull icebreaking tanker, MT Uikku, is chosen as the object in the present study. Zhou et al. [29, 30] carried out a series of model tests in the multifunctional ice basin of the Marine Technology Group at Aalto University. The ship model was set to be even keel without heel angle, and the scale factor $\lambda=31.56$. The principal dimensions of the ship are listed in Table 3 and the profile of the hull is shown in Fig. 8. The parameters used in the simulations are in full scale based on Froude scaling.

<table>
<thead>
<tr>
<th>Table 3 Principal dimensions of the icebreaking tanker - MT Uikku</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Items</strong></td>
</tr>
<tr>
<td>Length $L$ (m)</td>
</tr>
<tr>
<td>Breadth $B$ (m)</td>
</tr>
<tr>
<td>Draft $T$ (m)</td>
</tr>
<tr>
<td>Block coefficient $C_b$</td>
</tr>
</tbody>
</table>

In the model tests, the ship was towed by a carriage to keep a constant velocity. The 6-component force transducer and cameras were attached to the ship model to measure ice resistance and record the icebreaking process. Three drift angles ($0^\circ$, $45^\circ$ and $90^\circ$) were tested in the model test. Here, only the data from cases with drift angle $0^\circ$ are utilized to compare with numerical results. The ship velocities and measured ice properties in model scale and full scale are listed in Table 4. The numerical model of full-scaled continuous icebreaking scenario is established in LS-DYNA as shown in Fig. 9.

<table>
<thead>
<tr>
<th>Table 4 Ship velocity and measured ice properties in model scale and full scale</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Case No.</strong></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>103</td>
</tr>
<tr>
<td>104</td>
</tr>
<tr>
<td>205</td>
</tr>
<tr>
<td>206</td>
</tr>
</tbody>
</table>
The velocity boundary condition is imposed on the ship by the command of Prescribed-Motion to ensure that the ship impacts with level ice with the constant velocity. The ship is free to move in the x direction while the motions in other directions are constrained. This is in accordance with the setup in the model tests. The length of ice sheet is two times of ship length and the width is 4.5 times of ship breadth, which can reduce the computational burden and also help to eliminate boundary effect. The boundary condition of the three sides of ice sheet without contacting with ship is set as fixed to represent the restraining effect of the boundless level ice around the established ice domain. Meanwhile, non-reflecting boundary conditions are also set on these boundaries to further eliminate boundary effect. The mesh of ship and ice sheet are shown in Fig. 10. The ice elements are constituted by regular tri-prism bulk elements and ultrathin hexahedral cohesive elements, whose layout can refer to Fig. 2. There is only one layer of ice elements in the vertical direction and the thickness of ice cohesive elements is defined to be 1% of the length of ice bulk elements. Besides the parameters measured in the model tests as listed in Table 4, the other parameters of ice bulk elements and cohesive elements are given in Table 5. The material of ship is set as rigid steel without consideration of deformation during the interaction.
Study of continuous icebreaking process with cohesive element method

Feng Wang, Li Zhou, Zao-Jian Zou
Ming Song, Yang Wang, Yi Liu

Fig. 10 The mesh of ship and ice sheet

Table 5 Material parameters of ice bulk elements and cohesive elements

<table>
<thead>
<tr>
<th>Bulk elements</th>
<th>Value</th>
<th>Cohesive elements</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density $\rho_i$ (kg/m$^3$)</td>
<td>910</td>
<td>Density $\rho_i$ (kg/m$^3$)</td>
<td>910</td>
</tr>
<tr>
<td>Elastic modulus $E_i$ (MPa)</td>
<td>see Table 4</td>
<td>Tensile strength $\sigma_T$ (kPa)</td>
<td>500</td>
</tr>
<tr>
<td>Poisson ratio $\nu$</td>
<td>0.3</td>
<td>Shear strength $\sigma_S$ (kPa)</td>
<td>700</td>
</tr>
<tr>
<td>Compressive stress $\sigma_Y$ (kPa)</td>
<td>see Table 4</td>
<td>Fracture energy in mode I $G_{IC}$ (J/m$^2$)</td>
<td>30</td>
</tr>
<tr>
<td>Crushed stress $\sigma_c$ (kPa)</td>
<td>100</td>
<td>Fracture energy in mode II $G_{HIC}$ (J/m$^2$)</td>
<td>30</td>
</tr>
<tr>
<td>Crushed strain $\varepsilon_c$</td>
<td>0.4</td>
<td>Shape of TSL curve</td>
<td>Bilinear</td>
</tr>
<tr>
<td>Failure strain $\varepsilon_f$</td>
<td>0.5</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

In LS-DYNA, there are various of contact algorithms to treat the contact problems between the structures with different stiffness. The command of CONTACT-ERODING-NODES-TO-SURFACE is applied to treat interaction between ice and ship, and the command of CONTACT-ERODING-SINGLE-SURFACE is applied to treat interaction between ice and ice. Friction coefficient of 0.2 is set to the contact between ice and ship, while friction coefficient of 0.1 is set to the contact between ice and ice. In addition, the simplified buoyancy and damping models are established to simulate the interaction between ice and water base and incorporated in LS-DYNA solver by a user-defined subroutine. More details could be found in Wang et al. [22].

4.2 Mesh dependency study and validation of numerical methods

The mesh dependency study and validation of numerical methods are performed by comparing with experimental results in this section. In the previous numerical investigations on cohesive element method [21,22], it was found that the mesh size had an effect on the simulation results. In order to examine if such effect exists in the present numerical simulations, the mesh dependency study is carried out firstly. Based on the icebreaking scenarios as shown in Table 4, the ice resistance during the ship moving one hull length in ice sheet is calculated with three kinds of ice meshes, whose horizontal length are 4m, 2m and 1m. Label them as coarse mesh, medium mesh and dense mesh respectively.

Fig. 11 shows the relative proportion of mean resistance calculated with three meshes to those of model tests (the resistance obtained from the model tests has been converted to full-
scale values based on Froude criterion). It can be seen that the mean forces present growing and convergent trends with the refinement of mesh. Compared to the experimental results, the medium mesh obtains the closest results with the error of -8%~7%, while dense mesh slightly overestimates the forces for 7%~18% and coarse mesh obtains obviously lower forces with the relative proportion of 60%~77%.

![Graph showing relative proportion of simulated mean ice resistance with different meshes to experimental results](image)

**Fig. 11** Relative proportion of simulated mean ice resistance with different meshes to experimental results

Lu et al. [21] provided an explanation for this result. He found that the typical ice breaking length in the radial direction is about 2~3 times of the ice thickness based on the observations in the model tests of the ice-conical structure interaction. With a mesh size similar to the actual ice breaking length in the radial direction can help to approximate the size of crack and obtain a better simulation result. Based on the measurements in the model tests by Zhou et al. [29], the full-scaled radial ice breaking length during icebreaking process varies within 1.58m~3.16m, whose mean value is rather close to the horizontal length of medium mesh applied in the present simulations. The simulation results provide another evidence for the conclusion by Lu et al. [21]. Thus, taking the advantage of the length being close to ice radial breaking length, the medium mesh is applied in the following simulations for the accurate prediction of ice resistance. Table 6 gives the mean values and standard deviations of ice resistance from experiments and simulations with medium mesh.

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Model tests</th>
<th>Simulations</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mean (kN)</td>
<td>StDev (kN)</td>
</tr>
<tr>
<td>103</td>
<td>311</td>
<td>170</td>
</tr>
<tr>
<td>104</td>
<td>415</td>
<td>185</td>
</tr>
<tr>
<td>205</td>
<td>525</td>
<td>217</td>
</tr>
<tr>
<td>206</td>
<td>617</td>
<td>333</td>
</tr>
</tbody>
</table>

**Table 6** Mean and standard deviations of ice resistance for experiments and simulations with medium mesh

**Fig. 12** shows the comparison of time histories of horizontal ice force between the model tests and the numerical simulations with medium mesh in the range of 50m to 150m. In
order to eliminate the noise from the results, the presented force curves are obtained by processing the original data through the same filter. It is observed that the numerical results agree well with the experimental results, in which both the loads curves show the considerable oscillation around the mean value. The fluctuation amplitudes of calculated ice resistance are slightly higher than those of measured loads, which will be explained later.

**Fig. 12** Time histories of longitudinal force from experiments and simulations
Fig. 13 shows the power spectra analysis of longitudinal ice forces on the hull. It is observed that both the simulated and measured results show one or two obvious spectral peaks. The frequencies corresponding to these peaks are the main ice-breaking frequency, which represents the impulse period of the ice bending-induced force on the hull. It can be observed that the ice bending-induced force is low frequency force, i.e., the spectral peaks appear under 2 Hz. The spectral peaks of the model test appear around 0.2~0.6Hz, while the simulation results give a slightly higher frequency of 0.6~1.4 Hz. In addition, there are some low-power energy distributions in both of simulated and experimental spectra. This is mainly caused by local crushing force and the contact force between ice cusps and hull (e.g., the accumulating force and friction force during the ice cusps sliding along the hull). These energy levels are obviously lower than the energy peaks due to ice bending-induced force, which indicates that the ice bending failure is the dominating ice failure pattern during the continuous icebreaking process.

![Power spectra analysis of longitudinal ice forces on the hull](image)

**Fig. 13 Power spectra analysis of longitudinal ice forces on the hull**

Taken Case 205 as an example, the simulated continuous icebreaking process are presented in Fig. 14. The outline of generated icebreaking channel is highlighted by the red lines in Figs. 14a–c. In order to facilitate the observation of ice bending failure, the ice bulk elements are hided and only the ice cohesive elements are shown in Figs. 14d–f. The whole continuous icebreaking process can be observed in the simulation results. At the initial stage of ship-ice interaction (see Fig. 14a, d), the ice local crushing happens at the contact zone. There is no obvious bending failure at this stage due to small contact area between bow and level ice. With the continuous advance of ship and the increase of contact area, the bending failure of ice sheet starts to arise. The initial crack appears at the ship shoulder zone and propagates towards outboard. The irregular icebreaking channel and large ice cusps are generated due to ice bending failure, which can be clearly observed in Fig. 14b–c. The large ice cusps rotate and collide against the hull at the waterline zone under the action of buoyancy and gravity (see Fig. 14e–f). Fig. 15 shows the ice failure patterns observed in the model test for Case 205. The similar phenomenon of ice cusps rotation was also observed in the model test. From Fig. 15c, it is shown that the shape of icebreaking channel in the model test is irregular, which is caused by the asymmetric and non-simultaneous ice failure patterns at the
two sides of hull. A good agreement is achieved between simulated results and experimental results.

Fig. 14 Snapshots of simulated continuous icebreaking process for Case 205 (a. \( t = 50s \), b. \( t = 350s \), c. \( t = 700s \), top view; d. \( t = 50s \), e. \( t = 350s \), f. \( t = 700s \), front view).

Fig. 15 Snapshots of the ice failure patterns in the model test for Case 205 (a. failure patterns at the starboard side; b. failure patterns at the port side; c. outline of icebreaking channel)

Fig. 16 shows the side view of simulated icebreaking process for Case 205 at \( t = 700s \). It can be seen that the broken ice cusps slide backwards along the hull surface. There is not
much rubble accumulation around the ship bow due to the smooth shape of bow. The ice accumulation mainly distributes at the both sides and bottom of hull. As the ship moves forward, ice cusps slide along the hull surface to the stern. During the ice sliding process, the friction will be imposed on the hull. From Fig. 16, it can also be seen the generated ice cusps with various sizes and shapes are rather similar to those observed in Fig. 15c. However, the width of large ice cusps generated in the numerical simulation are slightly larger than those observed in the model test, which indicates that the simulated ice breaking length is slightly larger than the experimental result. Therefore, when the ice sheet experiences bending failure in the numerical simulation, greater loads fluctuation will occur. This is the reason that the fluctuation amplitudes of ice resistance obtained in the numerical simulations are larger than the experimental results as shown in Fig. 12.

4.3 Effect of ship velocity

The validation of the numerical method has been performed by comparing with the model test results in the last section. The influence of ship velocity $V$ on ice resistance and ice failure patterns will be studied in this section. Five ship velocities, 0.2m/s, 0.35m/s, 0.5m/s, 0.65m/s and 0.8m/s, are selected and the thickness of ice sheet are 760mm and 950mm. Since the model tests only contain the scenarios with ship velocities of 0.2m/s and 0.5m/s, the semi-empirical formulas proposed by Lindqvist [5] and Riska et al. [7] are introduced for the comparison purpose.

The Lindqvist semi-empirical formulas are expressed as:

\[
R_i = (R_c + R_b) \left( \frac{1 + 1.4V}{\sqrt{gh}} \right) + R_s \left( \frac{1 + 9.4V}{\sqrt{gL}} \right)
\]

(1)

\[
R_c = 0.5\sigma_b h^2 \left( \tan \phi + \mu \frac{\cos \phi}{\cos \psi} \right) \left( 1 - \mu \frac{\sin \phi}{\cos \psi} \right)
\]

(2)

\[
R_b = \frac{27}{64} \sigma_B B \frac{h^{1.5}}{E} \left( \frac{1}{12(1 - v^2)} \right) \left( \tan \psi + \mu \frac{\cos \phi}{\cos \psi \sin \alpha} \right) \left( 1 + \frac{1}{\cos \psi} \right)
\]

(3)

\[
R_s = (\rho_w - \rho) ghB \left( T \frac{B + T}{B + 2T} + \mu \left( 0.7L - \frac{T}{\tan \phi} - \frac{B}{4\tan \alpha} \right) + T \cos \phi \cos \psi \sqrt{\frac{1}{\sin \phi^2} + \frac{1}{\tan \alpha^2}} \right)
\]

(4)

where $R_i$, $R_c$, $R_b$, and $R_s$ are the total ice resistance, crushing resistance, bending resistance and submersion resistance, respectively. $\sigma_b$ is the ice bending strength, which are 828kPa for 760mm thick ice and 920kPa for 950mm thick ice, based on the measurements in the model tests by Zhou et al. [29]. $\mu$ is the friction coefficient between the ship and the ice, and $\rho_w$ is the water density. $\phi$ is the stem angle, $\alpha$ is the waterline entrance angle, and $\psi$ is the normal angle...
which is defined by $\psi = \tan(\tan\phi/\sin\alpha)$. The illustration of the hull angles in Lindqvist formulas is shown by Fig. 17.

![Illustration of hull angles](image)

**Fig. 17** Illustration of the hull angles in Lindqvist formula

The Riska semi-empirical formulas are given as:

$$R_i = C_1 + C_iV$$  

(5)

$$C_1 = \left(\frac{f_1}{2T_B + 1}\right)B L_P h + (1 + 0.021\phi)(f_2 Bh^2 + f_3 L_n h^3 + f_4 L_0 h)$$  

(6)

$$C_2 = (1 + 0.063\phi)(g_1 h^{1.5} + g_2 Bh) + g_3 h(1 + \frac{1.2T_B}{B})\frac{B^2}{\sqrt{T}}$$  

(7)

where $L_P$ and $L_B$ are length of parallel middle body and length of bow, which are 65m and 39m respectively. The other parameters are empirical constants, as given in Table 7.

<table>
<thead>
<tr>
<th>Items</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$f_1$ (kN/m³)</td>
<td>0.23</td>
</tr>
<tr>
<td>$f_2$ (kN/m³)</td>
<td>4.58</td>
</tr>
<tr>
<td>$f_3$ (kN/m³)</td>
<td>1.47</td>
</tr>
<tr>
<td>$f_4$ (kN/m³)</td>
<td>0.29</td>
</tr>
<tr>
<td>$g_1$ (m/s·m¹·⁵)</td>
<td>18.9</td>
</tr>
<tr>
<td>$g_2$ (m/s·m)</td>
<td>0.67</td>
</tr>
<tr>
<td>$g_3$ (m/s·m²·⁵)</td>
<td>1.55</td>
</tr>
</tbody>
</table>

The ice resistances versus ship velocity obtained by numerical simulation and the semi-empirical formulas are shown in Fig. 18, and the experimental results are also presented for the comparison. The ice resistance calculated by the two semi-empirical formulas increases linearly with the ship velocity, and the ice resistance calculated numerically also increases approximately linearly with the ship velocity. The ice resistance calculated by Riska formulas are significantly higher than the experimental results and other calculation results. This is because the influence of various ice properties on ice resistance is not considered in Riska formulas, which may influence the calculation accuracy of Riska formulas. The ice resistance calculated by Lindqvist formulas is smaller than the experimental results, especially for the scenarios with 950mm thick ice. The similar result is also obtained by the experimental research by Myland and Ehlers [31]. The reason may be that the components of ice resistance due to the rotation of ice cusps is not taken into account in Lindqvist formulas. The ice resistance calculated by numerical simulation is closest to the experimental results.
Fig. 19 shows the relationship between the standard deviation of ice resistance and ship velocity calculated by the numerical simulations. It can be seen that the standard deviations of ice resistance present obvious decreasing trend with the increase of ship velocity. Especially when the ship velocity is greater than 0.5 m/s, the standard deviations of ice resistance decrease rapidly. Fig. 20 shows the time histories of longitudinal forces on ship with different velocities under two thick ice sheet. The force levels of ice resistance are higher at high velocity, which is because the impact between the ship and ice sheet at higher velocity will generate greater crushing contact force. However, the fluctuation amplitudes of ice resistance at high velocity are obviously smaller than those at low velocity. Fig. 21 shows the simulated results of continuous icebreaking process with different ship velocities. It can be observed that the width of broken ice channel at low velocity is obviously wider than that at high velocity. The width of broken ice channel at the velocity of 0.8 m/s is only slightly wider than the ship breadth. This indicates that ice breaking length decreases rapidly at high velocity, which results in smaller fluctuation amplitude of ice resistance at high velocity.
Study of continuous icebreaking process with cohesive element method

Feng Wang, Li Zhou, Zao-Jian Zou
Ming Song, Yang Wang, Yi Liu

Fig. 20 The influence of ship velocity on longitudinal force

Fig. 21 The simulated results of continuous icebreaking process with different ship velocities
5. Conclusion

In this paper, the continuous icebreaking process in level ice for the icebreaking tanker - MT Uikku is simulated by combining an elastoplastic softening constitutive model with cohesive element method (CEM). The numerical models are established in commercial finite element software LS-DYNA. Firstly, the elastoplastic softening constitutive model is calibrated by simulating ice cone crushing tests, and the effects of different softening laws on simulation results are evaluated. Then, the continuous icebreaking process is simulated by numerical methods. Mesh dependency study is performed and the simulated results are compared with the experimental results in terms of ice resistance and ice failure patterns. Finally, the effect of ship velocity on ice resistance and ice failure patterns are also analyzed and discussed. The main conclusions can be drawn as follows:

(1) Among three softening laws for the ice elastoplastic softening constitutive model, the numerical results calculated by linear softening law fits best with the experimental results in terms of force-displacement curves and pressure-area curves for the ice cone crushing tests.

(2) The correct selection of mesh size is crucial for the computational accuracy of the cohesive element method. With a mesh size similar to ice breaking length in the radial direction, the CEM simulation can provide a good estimation of ice resistance and modelling of ice failure patterns. The calculated longitudinal forces on the ship agree well with the measured ones in both of the time domain and frequency domain. The power spectral peaks of ice resistance obtained by numerical simulations and model tests are below 2Hz, which indicates that the ice bending-induced force with low frequency is dominant in the total ice resistance.

(3) The numerical method achieves good agreement with the experimental observations in terms of ice failure patterns. The simulations well reproduce the ice local crushing and bending failure, as well as the rotation, accumulation and sliding process of ice cusps during the continuous icebreaking process. The asymmetric and non-simultaneous failure patterns happened at the both sides of the hull cause an irregular icebreaking channel, which reflects the random and uncertain nature inherent in the continuous icebreaking process.

(4) The mean ice resistance calculated by simulations presents approximately linear increasing tendency with the increase of ship velocity. The numerical simulation results agree well with the experimental results. Riska formulas give the highest ice resistance prediction, whereas Lindqvist formulas underestimate ice resistance. The ice breaking length decreases rapidly with the increase of ship velocity, which results in the fluctuation amplitude of ice resistance to be obviously smaller at higher velocity.

It is shown that the presented numerical methods capture well the main features of the continuous icebreaking process and have a good prospect in modeling ship-ice interactions. In order to facilitate the comparison between numerical results and available experimental results, two kinds of ice with different properties are established in the calibration of ice constitutive model and validation of numerical methods to simulate ship icebreaking process. Different ice properties may generate deviations on the numerical results. However, limited by the available experimental data, the possible deviation arisen from this could not be taken into consideration. In addition, the above conclusions are drawn based on the present simulations on the continuous icebreaking process of the icebreaking tanker - MT Uikku. The reliability and generality of the numerical results still need to be proved by more investigations on other ships in the future.
ACKNOWLEDGEMENTS

The corresponding author (second author) greatly acknowledges the supports of the National Natural Science Foundation of China (Grant No. 51809124, 5181102016, 51709136), Natural Science Foundation of Jiangsu Province of China (Grant No. BK20170576), the Natural Science Foundation of the Higher Education Institutions of Jiangsu Province of China (Grant No. 17KJB580006) and State Key Laboratory of Ocean Engineering (Shanghai Jiao Tong University) (Grant No. 1704).

REFERENCES

Study of continuous icebreaking process with cohesive element method


VEHICLE SECURING SAFETY ASSESSMENTS OF A KOREAN COASTAL CAR FERRY ACCORDING TO ACCELERATION PREDICTION APPROACHES

UDC 629.541.2:629.331:629.5.065.2
Professional paper

Summary

The capsize and subsequent sinking of a coastal car ferry occurred along the Korean coast, resulting in hundreds of casualties. The rapid course change of the ship might have forced improperly secured cargoes to rush to one side and accelerated the capsizing event. This paper provides a comparative study of vehicle securing safety assessments composed of evaluations of the external inertia forces and lashing strengths for a car and a truck. The external inertia forces were evaluated based on the IMO CSS (CSS approach) and rule-based maximum motion angles (RULE approach). Being a car ferry as a target ship, the sea states were collected along the most frequent seagoing routes of the target ship. Frequency domain seakeeping analyses (FSA approach) were carried out and then the long-term motion components were derived using the collected sea state data. The long-term motion components were put forward based on time domain seakeeping analyses (TSA approach). The TSA approach estimated the most optimistic external forces, while the CSS approach provided the most conservative external forces. Assuming the vehicles were secured symmetrically with four steel wires, the lashing strengths were derived. More numbers of lashing cables were required for the heavy vehicles when the CSS approach was applied, while other approaches predicted sufficient lashing strengths compared to exerted forces.

Key words: car ferry; acceleration; inertia force; vehicle securing; lashing

1. Introduction

The sinking of a Korean coastal car ferry in 2014 caused more than 300 deaths. Cargoes including vehicles were not secured properly in the car ferry and a sudden course change might have induced a rush of general cargoes and cars. This motivated a review of the importance of vehicle stowage and securing in a car ferry.

The lashing rules on container cargoes are well defined in references such as a ship rule [1]. Hwang et al. [2] examined the container lashing technique, in which different types of lashing equipment were used. Shin and Hwang [3] performed the container stowage optimization based on a genetic algorithm.
On the other hand, there have been few studies on the securing of general cargoes and vehicles. Turnbull and Dawson [4] suggested a mathematical model for calculating the trailer lashing forces. A classification society, DNV, developed and distributed an Excel macro called LashCon [5] that made it possible to estimate the cargo securing safety based on International Maritime Organization (IMO): Code of safe practice for cargo stowage and securing (CSS, [6]).

IMO [6] suggested a systematic procedure to assess the cargo securing safety in terms of the external inertia forces and lashing strengths. The IMO CSS suggested the tabulated acceleration components of ocean-going vessels when calculating the external inertia force components. Considering coastal liners are subjected to less motion-induced acceleration than ocean-going ones, IMO CSS is expected to provide more conservative inertia forces for car ferries sailing within the coastal area. This is the fundamental motivation for carrying out this study. The so-called direct load approach (DLA) were applied to a Korean coastal car ferry to evaluate conservancy of IMO CSS code.

In this paper, a car ferry with the displacement of 1,633 tonf, which has been a coastal liner, was selected as the target vessel. The main voyage routes of Korean coastal car ferries were also investigated to collect sea state data. The short-term sea data were collected from the sea observation buoys and stations close to main voyage routes for 64 months and a long-term wave scatter diagram was newly constructed. To predict the motion and hydrodynamic forces, which are necessary to estimate lashing strength, of a floating body in waves, experimental and numerical analysis can be applied. Among them, the numerical simulations based on the potential theory have been generally performed in frequency domain [7] or time domain [8] since they are less expensive than other methods but give proper results expect for the cases when non-linear viscous effects are important. In this study, both frequency and time domain hydrodynamic analyses were carried out to determine how much long-term motion components would be developed in the vehicles loaded in the car ferry. These approaches are called the frequency domain seakeeping analysis (FSA approach) and the time domain seakeeping analysis (TSA approach), respectively. The FSA- and TSA-based long-term motion components for the car ferry were derived using the wave scatter diagram.

Some ship rules, such as KR-Rules [9], suggest the maximum roll and pitch angles which can be used for calculating the motion-induced acceleration components. External force components can be predicted using these long-term acceleration components; hence, this approach is called the RULE approach. The RULE-based long-term acceleration components were also provided in this study.

The lashing safeties of a 0.96 tonf car and a 39 tonf truck were evaluated. The car and truck are believed to experience the largest roll acceleration because they were stowed on the farthest side of the ship. This paper calculates the external force components according to the four approaches. The lashing strengths were evaluated using LashCon [5] and LashingSafety by Jo et al. [10]).

2. Target vessel and vehicles

2.1 Target vessel

The target ship was a coastal car ferry built in a Korean shipyard. Table 1 lists the principal dimensions. In this paper, the full load condition was taken into account for seakeeping analyses, since a survey on the navigation records showed that the full load condition shares a large portion between two typical loading conditions: full load and ballast conditions.
Table 2 provides information on the mass and center of mass (COM), where the longitudinal center of mass (LCOM) and vertical center of mass (VCOM) were measured from after the perpendicular (AP) and baseline (BL).

**Table 1 Principal dimensions of the coastal car ferry**

<table>
<thead>
<tr>
<th>Item</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length overall, $L_{OA}$ (m)</td>
<td>69.76</td>
</tr>
<tr>
<td>Length between perpendiculars, $L_{BP}$ (m)</td>
<td>56.00</td>
</tr>
<tr>
<td>Breadth molded, $B_m$ (m)</td>
<td>14.0</td>
</tr>
<tr>
<td>Depth molded, $D_m$ (m)</td>
<td>3.50</td>
</tr>
<tr>
<td>Mean draught at full load, $T_f$ (m)</td>
<td>2.65</td>
</tr>
<tr>
<td>Full load displacement, $\Delta_f$ (tonf)</td>
<td>1632.90</td>
</tr>
</tbody>
</table>

**Table 2 Principal dimensions of the coastal car ferry**

<table>
<thead>
<tr>
<th>Item</th>
<th>Ballast</th>
<th>Full load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of lightship, $W_L$ (ton)</td>
<td>1111.50</td>
<td>1111.50</td>
</tr>
<tr>
<td>Mass of deadweight, $W_D$ (ton)</td>
<td>282.95</td>
<td>521.40</td>
</tr>
<tr>
<td>$2^{nd}$ moment of mass, $I_x$ (ton- m$^2$)</td>
<td>3.57E10</td>
<td>3.97E10</td>
</tr>
<tr>
<td>$2^{nd}$ moment of mass, $I_y$ (ton- m$^2$)</td>
<td>3.56E11</td>
<td>4.16E11</td>
</tr>
<tr>
<td>$2^{nd}$ moment of mass, $I_z$ (ton- m$^2$)</td>
<td>3.25E11</td>
<td>3.76E11</td>
</tr>
<tr>
<td>LCOM(m)</td>
<td>26.30</td>
<td>27.3</td>
</tr>
<tr>
<td>VCOM(m)</td>
<td>5.18</td>
<td>5.30</td>
</tr>
</tbody>
</table>

### 2.2 Target vehicles

The ferry can load 5 heavy trucks and 22 cars for which the masses are 39 tonf and 0.96 tonf, respectively. Fig. 1 shows the main deck plan with the loaded heavy trucks and cars in the fore and after parts, respectively.

Two vehicles, a car and a truck, in the red rectangle lines in Fig. 1 were taken into account because they are located on the farthest port side and expected to be subjected to most extreme roll acceleration.

The spatial locations can be defined using the distance components of $r_x$, $r_y$, and $r_z$ from the center of ship mass $G$, as delineated in Fig. 2. Table 3 provides detailed information on the two vehicles including the masses, sizes and locations.

**Fig. 1 Upper deck plan with cars and trucks loaded for full load condition**
Table 3 Information on the two vehicles

<table>
<thead>
<tr>
<th>Item</th>
<th>Car</th>
<th>Truck</th>
</tr>
</thead>
<tbody>
<tr>
<td>m (ton)</td>
<td>0.91</td>
<td>39.00</td>
</tr>
<tr>
<td>C_x (m)</td>
<td>3.60</td>
<td>12.62</td>
</tr>
<tr>
<td>C_y (m)</td>
<td>1.60</td>
<td>2.50</td>
</tr>
<tr>
<td>G_cz (m)</td>
<td>0.74</td>
<td>1.57</td>
</tr>
<tr>
<td>r_x (m)</td>
<td>7.27</td>
<td>3.47</td>
</tr>
<tr>
<td>r_y (m)</td>
<td>5.49</td>
<td>3.09</td>
</tr>
<tr>
<td>r_z (m)</td>
<td>0.24</td>
<td>0.24</td>
</tr>
</tbody>
</table>

3. COMPARISON OF THE EXTERNAL FORCES

3.1 CSS Approach

Neglecting wind pressure and sea water sloshing pressure, the IMO CSS suggests translational force components of \( F_x \), \( F_y \) and \( F_z \) as delineated in Eqs. (1)-(3), where \( f_{VL} \) and \( f_{BGM} \) are the acceleration correction factors as functions of the ship speed to length ratio and ship breadth to metacentric height ratio, respectively. \( a'_x \), \( a'_y \), and \( a'_z \) are the tangential acceleration components in longitudinal, transverse, and vertical directions, respectively (see Fig. 4). \( a'_x \) and \( a'_y \) include the gravitational acceleration component, while \( a'_z \) is the pure motion-induced acceleration component. \( F_z \) in Eq. (3) is a vertical force component. The transverse force, \( F_y \), with a combination of moment arm, \( G_{cz} \), can induce the tipping moment, \( M_x \), regarding the tipping line, as shown in Fig. 3, where \( x' \), \( y' \), and \( z' \) are the local coordinate directions with the origin at the rear wheel axis, centerline, and the bottom of a vehicle. In addition, \( m \), \( G_c \), \( C_x \), and \( C_y \) imply the mass, vertical center of mass, wheelbase, and tread of a vehicle.
\[ F_x = ma_x = m \left( f_{VL} a'_x \right) \]  \hspace{2cm} (1)

\[ F_y = ma_y = m \left( f_{VL} f_{BGM} a'_y \right) \]  \hspace{2cm} (2)

\[ F_z = ma_z = m \left( f_{VL} a'_z \right) \]  \hspace{2cm} (3)

\[ M_x = F_y G_{ez} \]  \hspace{2cm} (4)

Fig. 3 Sketch for a vehicle secured by a wire

Fig. 4 shows the translational acceleration components defined in IMO CSS code [6]. Longitudinal locations of the car and truck were assumed to be 0.4 and 0.6 in Fig. 4, respectively. In addition, the vertical locations were thought to be at the tween-deck in Fig. 4. \( f_{BGM} \), which is presented in tabular form can also be expressed by polynomials, as shown in Fig. 5.

<table>
<thead>
<tr>
<th>Transverse acceleration ( a'_y ) in m/s(^2)</th>
<th>Longitudinal acceleration ( a'_x ) in m/s(^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>on deck, high</td>
<td>0  1  2  3</td>
</tr>
<tr>
<td>on deck, low</td>
<td>4  5  6  7</td>
</tr>
<tr>
<td>tween-deck</td>
<td>8  9 10 11</td>
</tr>
<tr>
<td>lower hold</td>
<td>11 12 13 14</td>
</tr>
<tr>
<td>Vertical acceleration ( a'_z ) in m/s(^2)</td>
<td>---------------------------------------------</td>
</tr>
<tr>
<td>on deck, high</td>
<td>0  1  2  3</td>
</tr>
<tr>
<td>on deck, low</td>
<td>4  5  6  7</td>
</tr>
<tr>
<td>tween-deck</td>
<td>8  9 10 11</td>
</tr>
<tr>
<td>lower hold</td>
<td>11 12 13 14</td>
</tr>
</tbody>
</table>

Fig. 4 Translational acceleration data defined in IMO CSS(IMO, 2011)

Fig. 5 Polynomial regression of \( f_{BGM} \)
3.2 RULE Approach

According to IMO CSS, \( F_x \) and \( F_y \) are induced mainly by pitch and roll motion components, respectively, but the gravitational acceleration components of \( g_x \) and \( g_y \) can contribute to increasing \( F_x \) and \( F_y \), as shown in Eqs. (5)-(6). The vertical force component, \( F_z \), is caused by the heave-, pitch- and roll-induced acceleration components \((a_{h_z}, a_{p_z} \text{ and } a_{r_z})\), respectively, as delineated in Eq. (7). \( \theta_r \) and \( T_r \) are the roll angle and period, while \( \theta_p \) and \( T_p \) are the pitch angle and period, respectively. As depicted in Fig. 2, \( r_x, r_y, \) and \( r_z \) are the longitudinal, transverse, and vertical distances from ship mass center \((G)\) to vehicle mass center \((G_c)\). Common acceleration parameter, \( a_0 \), in Eq. (8) is taken from the classification society rules (DNV-GL, [5]). The RULE approach uses the same tipping moment, as shown in Eq. (4).

\[
F_x = m a_x = m \left( g_x + a_{p_x} \right) = m \left( g \sin \theta_p + \frac{2 \pi}{T_p} \theta_p r_z \right) a_{p_x} \tag{5}
\]

\[
F_y = m a_y = m \left( g_y + a_{r_y} \right) = m \left( g \sin \theta_p + \frac{2 \pi}{T_r} \theta_p r_z \right) a_{r_y} \tag{6}
\]

\[
F_z = m a_z = m \left( a_{h_z} + a_{p_z} + a_{r_z} \right) = m \left[ 0.8 \left( 1.15 - \frac{6.5}{\sqrt{g L_{BP}}} \right) a_0 g + \frac{2 \pi}{T_p} \theta_p r_x + \frac{2 \pi}{T_r} \theta_p r_y \right] \tag{7}
\]

\[
a_0 = (1.58 - 0.47C_b) \left( \frac{2.4}{\sqrt{L_{BP}}} + \frac{34}{L_{BP}} - \frac{600}{L_{BP}^2} \right) \tag{8}
\]

According to the ship rules of a classification society (KR, 2016), the maximum angles for roll and pitch motions \((\theta_r \text{ and } \theta_p)\) should be 25° and 5°, respectively. The maximum periods for roll and pitch motions \((T_r \text{ and } T_p)\) are also expressed by Eqs. (9) and (10), where \( GM \) is the metacentric height. The translational acceleration components can be derived by substituting the determined \( \theta_r, \theta_p, T_r, \) and \( T_p \) into Eqs. (5)-(7). The tipping moment component can be determined easily using Eq. (4). The RULE approach calculations were carried out using LashingSafety [10].
Vehicle securing safety assessments of a Korean coastal car ferry according to acceleration prediction approaches

\[ T_r = \frac{0.7 B_m}{\sqrt{GM}} \]  \hspace{1cm} (9)

\[ T_p = \frac{1}{2} \sqrt{L_{BP}} \]  \hspace{1cm} (10)

3.3 FSA Approach

As shown in Fig. 6, Incheon to Jeju, Busan to Jeju, Donghae to Ulung, and Pohang to Ulung are the main voyage routes in Korea [9]. Four ocean stations of Boksacho, Gyoboncho, Wangdolcho, and Ssangjeongcho are the closest ones from the main voyage routes. Therefore, the sea states at the locations were collected from the Korea Hydrographic and Oceanographic Agency (KHOA) for longer than 5 years from January in 2010 to April in 2015. Each dataset consists of a significant wave height \( H_s \) and significant wave period \( T_s \) for one hour measurement.

![Fig. 6 Main voyage routes with the locations of the ocean stations](image)

The significant wave period can be converted to zero up-crossing period \( T_z \) using Eq. \((11)\) by Kim [11]. Table 4 lists a newly constructed wave scatter diagram (WSD) according to \( H_s \) and \( T_z \), where sea states less than an occurrence probability of 1% were discarded.

\[ T_z = (4/5)^{0.25} \cdot T_s \]  \hspace{1cm} (11)

Considering the computing limits, the increment of wave incident angles was determined to be 22.5°. Each incident angle was assumed to have the same occurrence probability. The forward speed of 10 knots (5.144 m/s) was also applied to frequency response analyses considering the normal continuation rate (NCR) of the car ferry. The forward speeds for the following and quartering seas were reduced so that the encounter frequencies were kept positive. The range and number of frequencies can determine how well the frequencies describe the real wave spectrum in terms of the 0th order spectral moments (area of spectrum) and spectrum shapes for two extreme sea states of #1 with the shortest \( T_z \) (3.0s) and #11 with the longest \( T_z \) (11.0s). Fifty frequencies (49 increments) in total were applied to frequency response analyses, where the minimum and maximum frequencies were 0.1rad/s and 4.850rad/s. Table 5 lists the drafts at after perpendicular (AP) and fore
perpendicular (FP) and number of panels for the full loading condition. The two panel models for the loading condition were produced for frequency response analysis (refer to Fig. 7). A large sized-commercial vessel has a bilge keel to mitigate the roll motion so the viscous roll damping ratio, $\zeta$, is usually larger than 5%. Because there is no information on the roll damping ratio of the car ferry, a viscous roll damping ratio of 2.5% was assumed in this paper. As shown in Eq. (12), the roll damping constant $b_{roll}$ was calculated using the roll damping ratio. In equation (12), $I_x$ and $I_{xa}$ mean second moments of masses about roll axis corresponding to initial displacement and added mass, respectively. $k_x$ means rotational stiffness with respect to roll axis. $I_x$, $I_{xa}$, and $k_x$ are functions of frequency, so values at the roll resonance frequency were used.

With the panel models and environmental data described above, frequency response analyses were conducted to calculate the radiation and wave excitation coefficients $[12]$. 

$$b_{roll} = 2\zeta \sqrt{(I_x + I_{xa}) \times k_x}$$  \hspace{1cm} (12)

**Table 4** Newly constructed wave scatter diagram

<table>
<thead>
<tr>
<th>Sea state</th>
<th>$H_s$</th>
<th>$T_s$</th>
<th>Prob.</th>
</tr>
</thead>
<tbody>
<tr>
<td>01</td>
<td>0.5</td>
<td>3.0</td>
<td>0.1123</td>
</tr>
<tr>
<td>02</td>
<td>0.5</td>
<td>5.0</td>
<td>0.2839</td>
</tr>
<tr>
<td>03</td>
<td>0.5</td>
<td>7.0</td>
<td>0.2231</td>
</tr>
<tr>
<td>04</td>
<td>0.5</td>
<td>9.0</td>
<td>0.0644</td>
</tr>
<tr>
<td>05</td>
<td>0.5</td>
<td>11.0</td>
<td>0.0215</td>
</tr>
<tr>
<td>06</td>
<td>1.5</td>
<td>5.0</td>
<td>0.0392</td>
</tr>
<tr>
<td>07</td>
<td>1.5</td>
<td>7.0</td>
<td>0.1237</td>
</tr>
<tr>
<td>08</td>
<td>1.5</td>
<td>9.0</td>
<td>0.0777</td>
</tr>
<tr>
<td>09</td>
<td>1.5</td>
<td>11.0</td>
<td>0.0153</td>
</tr>
<tr>
<td>10</td>
<td>2.5</td>
<td>9.0</td>
<td>0.0272</td>
</tr>
<tr>
<td>11</td>
<td>2.5</td>
<td>11.0</td>
<td>0.0117</td>
</tr>
</tbody>
</table>

**Table 5** Information on the panels for the full loading condition

<table>
<thead>
<tr>
<th>Information</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of panels:</td>
<td></td>
</tr>
<tr>
<td>Wet part</td>
<td>6699</td>
</tr>
<tr>
<td>Dry part</td>
<td>124</td>
</tr>
<tr>
<td>Draft(m)</td>
<td></td>
</tr>
<tr>
<td>at after perpendicular</td>
<td>2.646</td>
</tr>
<tr>
<td>at fore perpendicular</td>
<td>2.646</td>
</tr>
</tbody>
</table>
Fig. 7 Panels for two loading conditions

Fig. 8 presents the roll and pitch motion RAOs for full load conditions. These RAOs were used to calculate the long-term motion components. Using the linear order Volterra series shown in Eq. (13), a wave spectrum $S_w(\omega)$ can be the motion spectrum $S_m(\omega)$ for each wave incident angle. After deriving the zero order spectral moment $m_0$ of a motion component from $S_m(\omega)$, the effect of short crested waves was taken into account using the cosine square spreading function, $f(\phi)$, as delineated in Eq. (14). Let the long-term probability level be $1.0 \times 10^{-8}$ corresponding to 20 years, then the long-term motion components can be derived using Eq. (15), where $p$, $x$, and $x_0$ are the probability of exceedance, motion component, and target motion component. The predicted long-term roll and pitch components were 41.07° and 12.26°, respectively.

\[ S_m(\omega) = RAO^2 \cdot S_w(\omega) \]  \hfill (13)

\[ f(\phi) = K \cdot \cos^2 \phi \]  \hfill (14)

\[ p(x \geq x_0) = e^{-\frac{x_0^2}{2m_0}} \]  \hfill (15)

where $\phi$ and $K$ are the wave incident angles around the central incident angle and spreading coefficient.
3.4 TSA Approach

The Cummins equation (Cummins, 1962) provides a numerical technique to solve the equations of motion for a floating body in the time domain with radiation coefficients and wave excitations determined from frequency response analyses. This means that any probable nonlinear effect from wave excitation forces cannot be taken into account, but the nonlinear effect due to the wave amplitude in way of mean water level is considered [12].

The maximum frequency of 4.585rad/s forces the time increment to be 0.2s. The time duration for each analysis case was decided to be one hour after comparing the statistical properties, such as the average and variance of peak distribution of a motion component between the one hour and two and half hour durations.

The ISSC standard wave spectrum was introduced to produce random wave excitation in the time domain seakeeping analyses. Considering the 11 sea states and 9 wave directions, 99 time domain analyses were conducted for each loading condition.

Fig. 9 shows the time response analysis results for sea state #10, which has the highest significant wave height. Time processes and spectra of the pitch motion component are shown for direction of 180° while the roll component results are depicted for the direction of 45°. To verify the validity of the time processes, the motion RAOs and wave spectrum were plotted together with the motion spectra. The resonance frequencies for the roll and pitch RAOs coincided relatively well with the response spectra, as delineated in Fig. 9 (a) and (b).

(a) Roll history, roll spectrum, wave spectrum, and roll RAO for a wave direction of 45°

(b) Pitch history, pitch spectrum, wave spectrum, and pitch RAO for a wave direction of 180°

Fig. 9 Time histories, motion spectra, wave spectra, and motion RAOs
A first step to predict the long-term extreme values of the motion components is to collect the peak and valley data from the motion processes. Then, distribution fitting based on Anderson-Darling test were conducted with a couple of PDFs in which seven different PDFs of generalized extreme, normal, log-normal (two and three parameters, respectively), Weibull (two and three parameters, respectively), and Gumbel distributions were used. All distributions proved to obey three parameter log-normal PDF shown in Eq. (16). Table 6 summarizes all the parameters derived. A second step for the long-term prediction is to calculate the probability of exceedance. Assuming that the probability of exceedance is $1.0 \times 10^{-8}$, the probability values corresponding to the accumulated probability of $1-(1.0 \times 10^{-8})$ become the long-term motion components as listed in Table 6. Considering the wave scatter diagram was based on 64 months, insufficient data collection period may not be suitable with the probability of exceedance $1.0 \times 10^{-8}$. The long-term motion components are less conservative than those obtained from the FSA approach.

$$f(x) = \exp \left[ -\frac{1}{2} \left( \frac{\ln(x-\gamma) - \mu}{\sigma} \right)^2 \right] f \left( (x-\gamma) \sigma \sqrt{2\pi} \right)$$

(16)

where $x$ implies a motion component and $\mu$, $\sigma$, and $\gamma$ are the mean, standard deviation, and location parameters.

### Table 6 Parameters of the log-normal PDFs and long-term values

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Roll</th>
<th>Pitch</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu$ (deg)</td>
<td>5.3251°</td>
<td>3.6670°</td>
</tr>
<tr>
<td>$\sigma$ (deg)</td>
<td>0.0206°</td>
<td>0.0227°</td>
</tr>
<tr>
<td>$\gamma$ (deg)</td>
<td>-205.470°</td>
<td>-38.917°</td>
</tr>
<tr>
<td>Long-term(deg)</td>
<td>25.10°</td>
<td>5.54°</td>
</tr>
</tbody>
</table>

#### 3.5 Comparison of the External Forces

LashCon [5], which was developed by Det Norske Veritas (DNV) and distributed free, was used to calculate the acceleration and force components based on IMO CSS code [6]. This is denoted by CSS-LC in Table 7. The other results based on the CSS, RULE, FSA, and TSA approaches were calculated using LashingSafety [10]. The tangential acceleration and load components were determined using the derived rotational acceleration component as the input data of LashingSafety.

### Table 7 Comparison of the acceleration and force components

<table>
<thead>
<tr>
<th>Case</th>
<th>Item</th>
<th>CSS</th>
<th>CSS-LC</th>
<th>RULE</th>
<th>FSA</th>
<th>TSA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Car</td>
<td>$a_x$ (m/s²)</td>
<td>2.36</td>
<td>2.36</td>
<td>0.91</td>
<td>2.23</td>
<td>1.01</td>
</tr>
<tr>
<td></td>
<td>$a_y$ (m/s²)</td>
<td>9.49</td>
<td>9.28</td>
<td>4.28</td>
<td>6.66</td>
<td>4.29</td>
</tr>
<tr>
<td></td>
<td>$a_z$ (m/s²)</td>
<td>5.07</td>
<td>5.06</td>
<td>10.85</td>
<td>15.35</td>
<td>11.01</td>
</tr>
<tr>
<td></td>
<td>$F_x$ (KN)</td>
<td>2.14</td>
<td>2.10</td>
<td>0.83</td>
<td>2.03</td>
<td>0.92</td>
</tr>
<tr>
<td></td>
<td>$F_y$ (KN)</td>
<td>8.64</td>
<td>8.40</td>
<td>3.89</td>
<td>6.06</td>
<td>3.91</td>
</tr>
<tr>
<td></td>
<td>$F_z$ (KN)</td>
<td>4.61</td>
<td>n/a</td>
<td>9.87</td>
<td>13.97</td>
<td>10.06</td>
</tr>
</tbody>
</table>
Joonmo Choung, Se-Min Jeong  

Vehicle securing safety assessments of a Korean coastal car ferry according to acceleration prediction approaches

<table>
<thead>
<tr>
<th></th>
<th>$M_x$ (KN-m)</th>
<th>6.39</th>
<th>6.20</th>
<th>2.88</th>
<th>4.48</th>
<th>2.89</th>
</tr>
</thead>
<tbody>
<tr>
<td>Truck</td>
<td>$a_x$ (m/s$^2$)</td>
<td>2.36</td>
<td>2.36</td>
<td>0.91</td>
<td>2.23</td>
<td>1.01</td>
</tr>
<tr>
<td></td>
<td>$a_y$ (m/s$^2$)</td>
<td>9.67</td>
<td>9.45</td>
<td>4.28</td>
<td>6.66</td>
<td>4.29</td>
</tr>
<tr>
<td></td>
<td>$a_z$ (m/s$^2$)</td>
<td>5.89</td>
<td>5.89</td>
<td>8.63</td>
<td>10.93</td>
<td>8.72</td>
</tr>
<tr>
<td></td>
<td>$F_x$ (KN)</td>
<td>91.90</td>
<td>91.90</td>
<td>35.66</td>
<td>86.91</td>
<td>39.50</td>
</tr>
<tr>
<td></td>
<td>$F_y$ (KN)</td>
<td>377.16</td>
<td>368.70</td>
<td>166.77</td>
<td>259.69</td>
<td>167.39</td>
</tr>
<tr>
<td></td>
<td>$F_z$ (KN)</td>
<td>229.76</td>
<td>n/a</td>
<td>336.39</td>
<td>426.45</td>
<td>340.25</td>
</tr>
<tr>
<td></td>
<td>$M_x$ (KN-m)</td>
<td>592.14</td>
<td>578.90</td>
<td>261.83</td>
<td>407.72</td>
<td>262.81</td>
</tr>
</tbody>
</table>

The difference in $f_{BGM}$ between the two cases of CSS and CSS-LC may induce a slight difference in the transverse acceleration components ($a_y$). This also slightly affects the tipping moments ($M_x$) between the two cases of CSS and CSS-LC. $a_x$ and $a_y$ (or $F_x$ and $F_y$) by CSS or CSS-LC were predicted conservatively compared to the other cases, whereas CSS or CSS-LC predicted a smaller $a_z$ (or $F_z$). The results by the FSA approach are similar to those by the CSS approach, but the TSA approach estimates $a_x$ and $a_y$ (or $F_x$ and $F_y$), which are close to ones by the RULE approach.

Considering that $a_z$ or $F_z$ are not used for the lashing strength components, the CSS approach predicts more conservative external force components than the other cases. This is why the IMO CSS acceleration data might be suitable for ocean-going vessels.

The differences in acceleration between those obtained by TSA and FSA would be due to four reasons. First, whether TSA or FSA is used, we tried to capture the acceleration components using Eqs. (5)-(7), which makes the acceleration calculation process consistent for the approaches of RULE, TSA, and FSA. This means that the long-term roll and pitch angles should be obtained to determine each acceleration component. However, if we directly calculate the long-term roll- and pitch-induced acceleration components from each distribution, they are significantly different from the present approaches. Second, considering FSA depends on linear extrapolation to predict long-terms values, conservancy of the FSA-based results appears to be very natural. Third, we used Ansys Aqwa [12] for TSA and FSA and especially, Aqwa/Naut module was used for TSA in which non-linear Froude-Krylov and hydrostatic forces were estimated under instantaneous incident wave elevation. This is one of the result differences between TSA and FSA. The last cause may be the time increment of TSA. The maximum frequency applied in this study was 4.85rad/s, which corresponds to a period of about 1.3s. Depending on the method of numerical integration scheme, the magnitude of the time increment may be different. However in order to minimize the possibility of divergence, a time increment of less than 0.2 seconds needs to be applied.
4. LASHING STRENGTH

4.1 Lashing Strength Formulas

Fig. 3 presents a sketch for a vehicle secured by a wire. The wire forms two angles of $\alpha$ and $\beta$ on the $y'z'$-plane and $x'y'$-plane, respectively. Using the given lashing strength or wire tension, $T$, the longitudinal and transverse lashing strengths of $F_{cx}$ and $F_{cy}$ can be expressed as Eqs. (17)-(18). Let $l$ be a moment arm measured from the tipping point, then the tipping strength becomes Eq. (19), where $FS$ is the factor of safety and 0.9 is suggested by IMO CSS.

\[
F_{cx} = \mu (mg - F_z) + T \sum_{i=1}^{n} \left( \mu \sin \alpha_i + \cos \alpha_i \sin \beta_i \right) \tag{17}
\]

\[
F_{cy} = \mu mg + T \sum_{i=1}^{n} \left( \mu \sin \alpha_i + \cos \alpha_i \cos \beta_i \right) \tag{18}
\]

\[
M_{cx} = \frac{1}{2} C_x mg + FS \sum_{i=1}^{n} (T_i \cdot l_i) \tag{19}
\]

where $\mu$ is the friction coefficient and $n$ is the number of lashing lines on one side.

4.2 Lashing Strengths of a Car and a Truck

In this study, it was assumed that four steel cables secure the car and truck, as shown in Fig. 10; the lashing positions are listed in Table 8.
Table 8 Lashing points on vehicle and deck (unit: m)

<table>
<thead>
<tr>
<th>Case</th>
<th>Lash.</th>
<th>(x_1)</th>
<th>(y_1)</th>
<th>(z_1)</th>
<th>(x_2)</th>
<th>(y_2)</th>
<th>(z_2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Car</td>
<td>L1</td>
<td>0.50</td>
<td>0.80</td>
<td>1.00</td>
<td>-0.50</td>
<td>1.80</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>L2</td>
<td>0.50</td>
<td>-0.80</td>
<td>1.00</td>
<td>-0.50</td>
<td>-1.80</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>L3</td>
<td>3.10</td>
<td>0.80</td>
<td>1.00</td>
<td>4.10</td>
<td>1.80</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>L4</td>
<td>3.10</td>
<td>-0.80</td>
<td>1.00</td>
<td>4.10</td>
<td>-1.80</td>
<td>0.00</td>
</tr>
<tr>
<td>Truck</td>
<td>L1</td>
<td>1.00</td>
<td>1.25</td>
<td>2.00</td>
<td>-1.00</td>
<td>3.25</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>L2</td>
<td>1.00</td>
<td>-1.25</td>
<td>2.00</td>
<td>-1.00</td>
<td>-3.25</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>L3</td>
<td>11.00</td>
<td>1.25</td>
<td>2.00</td>
<td>13.00</td>
<td>3.25</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>L4</td>
<td>11.00</td>
<td>-1.25</td>
<td>2.00</td>
<td>13.00</td>
<td>-3.25</td>
<td>0.00</td>
</tr>
</tbody>
</table>

This securing arrangement makes the lashing angles of \(\alpha\) and \(\beta\) to be 45°. Each cable was also assumed to be under a tension of 110kN (\(T=110\text{kN}\)), friction coefficient of \(\mu=0.3\), and FS of 0.9.

The lashing strengths should be independent of the external force calculation approaches. On the other hand, longitudinal lashing strength, \(F_{cx}\), includes the vertical translational force, as delineated in Eq. (17); thus, slight differences in \(F_{cx}\) were found according to the external force estimation approaches.

According to external force calculation cases, Table 9 lists the sum of the securing strengths on one side. One side means that the lashing strengths should be collected at one side because the sum of the lashing strengths on both sides must always be zero. In addition, the load to strength ratios as shown in Table 9 exceed unity, it means failed cargo securing.

Table 9 Comparison of the lashing strengths

<table>
<thead>
<tr>
<th>Case</th>
<th>Item</th>
<th>CSS</th>
<th>CSS-LC</th>
<th>RULE</th>
<th>FSA</th>
<th>TSA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Car</td>
<td>(F_{cx}) (kN)</td>
<td>158.64</td>
<td>158.00</td>
<td>156.39</td>
<td>155.16</td>
<td>156.33</td>
</tr>
<tr>
<td></td>
<td>(F_{cx}/F_{cx})</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
</tr>
<tr>
<td></td>
<td>(F_{cy}) (kN)</td>
<td>159.35</td>
<td>159.00</td>
<td>159.35</td>
<td>159.35</td>
<td>159.35</td>
</tr>
<tr>
<td></td>
<td>(F_{cy}/F_{cy})</td>
<td>0.05</td>
<td>0.05</td>
<td>0.02</td>
<td>0.04</td>
<td>0.02</td>
</tr>
<tr>
<td></td>
<td>(M_{cx}) (kN)</td>
<td>637.17</td>
<td>427.00</td>
<td>411.61</td>
<td>411.61</td>
<td>411.61</td>
</tr>
<tr>
<td></td>
<td>(M_{cx}/M_{cx})</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
<td>0.01</td>
</tr>
<tr>
<td>Truck</td>
<td>(F_{cx}) (kN)</td>
<td>270.77</td>
<td>203.00</td>
<td>170.56</td>
<td>143.54</td>
<td>169.40</td>
</tr>
<tr>
<td></td>
<td>(F_{cx}/F_{cx})</td>
<td>0.34</td>
<td>0.45</td>
<td>0.21</td>
<td>0.61</td>
<td>0.23</td>
</tr>
<tr>
<td></td>
<td>(F_{cx}) (kN)</td>
<td>271.48</td>
<td>271.00</td>
<td>271.48</td>
<td>271.48</td>
<td>271.48</td>
</tr>
<tr>
<td></td>
<td>(F_{cx}/F_{cy})</td>
<td>1.39</td>
<td>1.36</td>
<td>0.61</td>
<td>0.96</td>
<td>0.62</td>
</tr>
<tr>
<td></td>
<td>(F_{cx}) (kN)</td>
<td>1108.39</td>
<td>1204.00</td>
<td>1178.40</td>
<td>1178.40</td>
<td>1178.40</td>
</tr>
<tr>
<td></td>
<td>(M_{cx}/M_{cx})</td>
<td>0.53</td>
<td>0.48</td>
<td>0.22</td>
<td>0.35</td>
<td>0.22</td>
</tr>
</tbody>
</table>
Because $F_{cy}$ does not include an external force term, all approaches provide a similar $F_{cy}$, as shown in Table 9. $F_{cx}$ including the external force term, $F_z$, varies according to the approaches. $F_z$ of the truck was considerably larger than $F_z$ of the car, and the difference appears to be very large. When calculating $F_z$, Eq. (7) includes only the kinetic components, but CSS-LC showed such a difference including the self-weight term. In the future, IMO CSS should be able to prevent this confusion by providing a guide to the $F_z$ calculation method.

Although $M_{cx}$ does not include an external force term, $M_{cx}$ values by CSS and CSS-LC are different from each other. As a result of the rigorous analysis of the root causes through manual calculations, there was an error in calculating the moment arm in LashCon [5]. These errors were presumably caused by IMO CSS, providing incorrect figure information. Fig. 11 shows two $\alpha$ values, where $\alpha$ by IMO CSS is not correct so a physically correct $\alpha$ should be used.

![Fig. 11 Correct definition of $\alpha$](image)

4.3 Discussion on the Lashing Safety

As depicted in Eqs. (20)-(22), the translational inertia force components ($F_x$ and $F_y$) and a tipping moment component ($M_x$), which correspond to the longitudinal and transverse translation and roll directions, respectively, must be less than the cargo securing strengths of $F_{cx}$, $F_{cy}$, and $M_{cx}$.

\[
F_x < F_{cx} \quad (20)
\]
\[
F_y < F_{cy} \quad (21)
\]
\[
M_x < M_{cx} \quad (22)
\]

The plan for the car securing was determined to be safe, regardless of the external force approaches. On the other hand, when evaluating the lashing safety of the truck using the CSS approach, it is not safe anymore because $F_y$ is larger than $F_{cy}$ or $F_y/F_{cy}$ is larger than unity as delineated in Table 9. An additional number of lashing wires or an increase in the nominal size of the lashing wire is required in this case.

The acceleration components proposed by IMO CSS is suitable for ocean-going vessels. The RULE or TSA approach is considered to be a more realistic choice because the TSA
approach used the sea states collected around the Korean coast and level of acceleration based on the TSA approach is similar to that of the RULE approach.

5. CONCLUSIONS

A benchmark study on the vehicle securing safety was conducted for a Korean coastal car ferry with a full load displacement of 1,633 tonf in which a car and a truck were stowed at the port-most locations. The acceleration and force components acting on the secured vehicles were derived based on the four different approaches: IMO CSS approach (CSS), rule-based approach (RULE), frequency domain seakeeping analysis approach (FSA), and time domain seakeeping analysis approach (TSA).

The CSS approach uses the acceleration components proposed by IMO CSS. The commonality of the RULE, FSA, and TSA approaches is that they use the same formulae to predict the acceleration components, and the difference is that they apply different roll and pitch angles to these formulae.

In the CSS approach, the acceleration components were taken directly from IMO CSS, and three force components and a tipping moment component were determined.

The FSA approach calculates the motion RAOs directly after frequency response analysis is completed with the ship panel model and sea states collected around Korean coast area. The long-term roll and pitch components for the 20 years design period were 41.07° and 12.26°, respectively.

The long-term roll and pitch components corresponding to the probability of occurrence of 20 years were predicted using the TSA approach. The motion component histories were obtained from time domain simulations, and the three parameters for log-normal probability density function were captured by collecting the peaks and valleys of the roll and pitch motion components. The long-term values were 25.10° and 5.54° for the roll and pitch components, respectively.

To elevate the reliability of CSS approach, LashCon was also used for external force calculations. CSS and CSS-LS showed similar results in all acceleration components. IMO CSS is intended for ocean-going vessels, so the results by the CSS approach were much more conservative than by other approaches. The results by the FSA approach were close to those by the CSS approach, but the results by the TSA approach were close to those by the RULE approach. The RULE and TSA approaches can provide more reasonable force components for coastal car ferries than IMO CSS.

The lashing strength evaluation formulae were taken from IMO CSS. For the car and truck secured symmetrically by four steel wires, the longitudinal, transverse, and tipping strengths were calculated for the approaches. There were the difference between $F_c$ by cases of CSS and CSS-LS, because IMO CSS did not provide a firmly established procedure to predict $F_c$.

Further in-depth research will be needed to understand why the FSA and TSA approaches gave different results. In addition, it is necessary to perform FSA and TSA for various car ferries to determine the appropriate roll and pitch angles for Korean coastal waters.
6. ACKNOWLEDGMENTS

This study was financially supported by the Korea Ship Safety Technology Authority and was a part of the project titled ‘Manpower training program for ocean energy’, funded by the Ministry of Oceans and Fisheries, Korea.

REFERENCES

https://doi.org/10.3744/SNAK.2014.51.5.362
VALIDATION OF OPTIMALY DESIGNED STATOR-PROPELLER SYSTEM BY EFD AND CFD

UDC 629.5.016.7:629.5.018.71:629.544
Original scientific paper

Summary

The development of energy-saving devices to lower the energy efficiency design index (EEDI) of ships has been actively researched worldwide. One such device is an asymmetric pre-swirl stator, which helps to improve the propulsion efficiency by recovering the rotational energy generated during propeller rotation. Determining the pitch angle is the most important factor in the design of an efficient asymmetric pre-swirl stator. To optimize the pitch angle of an asymmetric pre-swirl stator, this study deals with potential-flow, computational fluid dynamics, and model tests. The model delivered power at a design speed of 24 kt was compared by changing the pitch angle by ±2° with respect to the reference angle designed using a potential-flow program. The commercial code Star-CCM+ was used for the numerical analysis, and the model was also tested in a towing tank at Pusan National University. This study proposes an effective method for determining and verifying the optimal pitch angle of an asymmetric pre-swirl stator.

Key words: asymmetric pre-swirl stator; pitch angle; optimization; model test; computational fluid dynamics (CFD);

1. Introduction

International environmental regulations are important to preventing climate change, and regulations against emissions of ships are increasing. The International Maritime Organization is reinforcing regulations that it initiated against CO₂, NOₓ, and SOₓ, which are representative ship exhaust gases, as the Energy Efficiency Design Index (EEDI), Tier III, and Emission Control Area (ECA) [1]. Accordingly, energy-saving devices have been developed to reduce the EEDI. Energy-saving device development is critical for reducing not only the energy
consumption of ships, but also CO₂ emissions. Energy-saving devices are defined for use in three zones, namely, Zones I, II, and III, as shown Fig. 1 [2].

![Fig. 1 Zones for the classification of energy-saving devices [2]](image)

These three zones are: I) pre-devices, which are installed toward the bow of the ship, II) main devices, such as propellers, and III) post-propeller devices installed toward the stern of the ship.

Currently, various energy-saving devices are being actively researched. The contra-rotating propeller (CRP), a main device, improves propulsion efficiency through recovering rotational energy by installing two propellers rotating in opposite directions on the drive shaft, one facing the front and the other facing the rear. Boucetta and Imine (2019) [3] have been carried out to determine the hydrodynamic characteristics in cavitating viscous flow of the co-rotating tandem propellers. This paper focuses on the asymmetric pre-swirl stator, because it currently seems to be the most effective device and has been frequently applied without any major problems. A pre-swirl stator, which is a pre-device, improves the propulsion efficiency by recovering the rotational energy generated during propeller rotation via the stator blades placed in front of the propeller. Thus, the pre-swirl stator imposes a flow velocity component against the tangential velocity lost by the propeller and reduces the energy consumption by approximately 4–5% [4–9]. Moreover, compared to that of a contra-rotating propeller, the shafting of a pre-swirl stator is less complicated, and it is also easier to install. Therefore, the initial installation costs are not only lower, but the risk of breakdown during operation is also reduced [10].

A pre-swirl stator was symmetric and has six blades, as shown in Fig. 2 [11]. Blades at the top and bottom positions appear to be missing; this is intended in consideration of the blockage effects against the on-coming flow and potential-flow docking problems, respectively.

![Fig. 2 Profile of Mitsubishi-style reaction fins](image)
Kim et al. [12] designed and developed an asymmetric pre-swirl stator with three blades on the starboard side and one blade on the port side, as shown in Fig. 3.

![Fig. 3 Profile of an asymmetric pre-swirl stator](image)

The proposed asymmetric pre-swirl stator has fewer blades on the starboard side, as the rotational flow component is different on each side because of the upward flow. The upward velocity is normally cancelled by the propeller rotational velocity on the starboard side, whereas the velocity on the port side is doubled. This phenomenon is evident in the typical velocity vector profile just behind the propeller, which is measured using a laser Doppler velocimeter (LDV) [13]. An asymmetric pre-swirl stator is advantageous compared with the symmetric 6 blades stator because it can not only increase the efficiency by evenly absorbing the rotational energy throughout but can also significantly reduce the manufacturing cost and the load on the stern by removing unnecessary blades. Kim et al. [10] showed that the application of an asymmetric pre-swirl stator to a 300 K very-large crude oil carrier (VLCC) helped increase the efficiency by approximately 5.6%. Lee et al. [14] applied pre-swirl stators of the constant-pitch-angle type and variable-pitch-angle type to a 160K LNG ship and improved its propulsion efficiency by approximately 6% and 8%, respectively.

With regard to asymmetric pre-swirl stators, if the pitch angle of the stator in non-uniform wake is identical for each stator, the axial components are non-uniform; moreover, the load on each stator varies significantly because the tangential velocities at the port and starboard sides are opposite to each other with respect to the stator. Therefore, attaining maximum efficiency is difficult unless the pitch angle of the stator at each stator position is adjusted appropriately. Lee et al. [4] used the iterative design method with the lifting surface theory considering the mutually induced velocity of the propeller and pre-swirl stator to determine the pitch angle of an asymmetric pre-swirl stator. They used the Preswirl Asymmetric Stator Analysis (PASTA) program to implement the calculation processes. The lifting-surface code which has been originally developed by Greely and Kerwin(1982) [14] (most popularly used code in propeller field) has been modified by including the stator modeling. The interaction between propeller and stator is considered by iterative scheme and in that case the treatment of stator’s wake has to be carefully done. Lee et al. [15] and Shin et al. [16] determined the pitch angle of an asymmetric pre-swirl stator using the PASTA program and found that the efficiency increased by 4–5%. However, the optimal pitch angle corresponding to maximum efficiency needs to be verified.

In the present study, the optimal pitch angle of an asymmetric pre-swirl stator was verified through computational fluid dynamics (CFD) and experimental fluid dynamics (EFD) model tests by changing the pitch angle by ±2° with respect to the pitch angle determined using the potential-flow program.
2. Design of an asymmetric pre-swirl stator

In the design of the asymmetric pre-swirl stator, a potential-flow-based analysis program was used for the propellers and pre-swirl stators. If the pitch angle of the stator blades is the same for all the blades during non-uniform countercurrents, it is difficult to achieve maximum efficiency without adjusting the pitch angle corresponding to the position of each blade. The stator design has been conducted to have an optimum loading (elliptic loading distribution) for each blade as shown Fig.4.

\[ \Gamma = \frac{\Gamma}{2\pi RV_s} \]  
\[ C_L = 2\pi \sin \alpha_{equiv}. \]

Here, \( L \), \( C_L \), \( c \), and \( V \) represent the lift force for the 2D cross section, the lift coefficient, the chord of the cross section, and the inflow velocity, respectively. Conversion of Eq.3 for equivalent incidence angle and substitution of Eq.1 results in the following Eq.4.

The computed circulation of radial loading in each blade is converted by the equivalent angle of attack (\( \alpha_{equiv} \)) as introduced in Eq.4.
Validation of Optimally Designed Stator-Propeller System by EFD and CFD

Yong Jin Shin, Moon Chan Kim, Jin Gu Kang, Hyeon Ung Kim, I Rok Shin

\[ \alpha_{equiv.} = \sin^{-1}\left( \frac{C_L}{2\pi} \right) \]

\[ = \sin^{-1}\left( \frac{2G}{(\frac{u}{V_s})(\frac{c}{R})} \right) \]

where \( G \) is non-dimensional circulation as given by \( G = \frac{\Gamma}{2\pi RV_s} \), \( c \) is local chord length at each radius, \( u \) local inflow velocity, \( C_L \) lift coefficient and \( V_s \) ship velocity at design condition. [10] The blade numbers are assigned based on the representation above, as shown in Fig. 5.

In the present study, the asymmetric pre-swirl stator proposed by Shin et al. [16] was used, as shown in Tables 1 and 2. The equivalent incident angle is verified, as shown in Fig. 6, using the potential-flow program, as explained above.

**Table 1**  Principle dimension of the asymmetric pre-swirl stator

<table>
<thead>
<tr>
<th>Section type</th>
<th>NACA 66</th>
</tr>
</thead>
<tbody>
<tr>
<td>Scale ratio</td>
<td>39.5</td>
</tr>
<tr>
<td>Model diameter (m)</td>
<td>0.2</td>
</tr>
<tr>
<td>Skew (°)</td>
<td>0.0</td>
</tr>
<tr>
<td>Rake (°)</td>
<td>0.0</td>
</tr>
<tr>
<td>No. of blades</td>
<td>4</td>
</tr>
</tbody>
</table>

Fig. 5 Definition of blade number (looking upstream)
If the pitch angle (equivalent angle) is designed largely until the stall point, the rotational energy of propeller can be recovered, however the drag of stator becomes large accordingly. Therefore, the reference of optimum equivalent angle is important in the design of pitch angle.

Table 2  Geometry of the asymmetric pre-swirl stator

<table>
<thead>
<tr>
<th>r/R</th>
<th>0.20</th>
<th>0.25</th>
<th>0.30</th>
<th>0.40</th>
<th>0.50</th>
<th>0.60</th>
<th>0.70</th>
<th>0.80</th>
<th>0.90</th>
<th>0.95</th>
<th>1.00</th>
</tr>
</thead>
<tbody>
<tr>
<td>C/D</td>
<td>0.177</td>
<td>0.174</td>
<td>0.171</td>
<td>0.165</td>
<td>0.159</td>
<td>0.153</td>
<td>0.147</td>
<td>0.142</td>
<td>0.136</td>
<td>0.133</td>
<td>0.130</td>
</tr>
<tr>
<td>f₀/C</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>t/D</td>
<td>0.037</td>
<td>0.035</td>
<td>0.033</td>
<td>0.030</td>
<td>0.026</td>
<td>0.023</td>
<td>0.019</td>
<td>0.016</td>
<td>0.012</td>
<td>0.011</td>
<td>0.009</td>
</tr>
</tbody>
</table>

Fig. 6  Radial circulation distribution on the stator blade

Kim et al. [12] confirmed that designing the pitch angle such that the maximum equivalent incidence angle is between −13° and −14° is generally most effective in improving the propulsion efficiency when a pre-swirl stator is installed on a low-speed full body ship. However, the drag of stator in case of container ships, which is the subject of this paper, increases significantly when large pitch angles are adopted. Therefore, the maximum angle is designated as −12°, which is slightly lower than the equivalent incidence angle used for low-speed full body ships [16]. Table 3 shows the pitch angles for each blade, and Fig. 7 shows the final 3D model.

Table 3  Comparison of pitch angle based on the stator type

<table>
<thead>
<tr>
<th>Blade No.</th>
<th>Position (°)</th>
<th>Pitch Angle (°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constant Type</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>45</td>
<td>14</td>
</tr>
<tr>
<td>2</td>
<td>90</td>
<td>19</td>
</tr>
<tr>
<td>3</td>
<td>135</td>
<td>12</td>
</tr>
<tr>
<td>4</td>
<td>270</td>
<td>2</td>
</tr>
</tbody>
</table>

138
In the present study, the model delivered power at a design speed of 24 kts is compared by changing the pitch angle of the stator by \( \pm 2^\circ \) with respect to the pitch angle determined using the potential-flow program, as shown in Table 3. The pitch angle variations of \( \pm 2^\circ \) are defined, as shown in Fig. 8. Table 4 shows each case according to the variation in the pitch angle.
Table 4 Different cases based on the pitch angle of different blades

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Additional pitch angle [°]</th>
<th>1st Blade</th>
<th>2nd Blade</th>
<th>3rd Blade</th>
<th>4th Blade</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>Optimally designed pitch angle using potential-flow program</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 2</td>
<td>+2</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Case 3</td>
<td>−2</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Case 4</td>
<td>0</td>
<td>+2</td>
<td>0</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Case 5</td>
<td>0</td>
<td>−2</td>
<td>0</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Case 6</td>
<td>0</td>
<td>0</td>
<td>+2</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Case 7</td>
<td>0</td>
<td>0</td>
<td>−2</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Case 8</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>+2</td>
<td></td>
</tr>
<tr>
<td>Case 9</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>−2</td>
<td></td>
</tr>
</tbody>
</table>

3. Materials and Methods

3.1 Target ship

The target ship used in the present study is a 3,600 TEU KRISO container ship (KCS), as shown in Fig. 9. The model ship was manufactured at a scale ratio of 39.5, considering the size of the towing tank at Pusan National University (PNU). The design speed of the model ship is 1.964 m/s. Table 5 shows the dimensions in detail.

![Fig. 9 3,600 TEU KRISO container ship model](image-url)
3.2 Computational setup

The commercial program STAR-CCM+ was used for the computations in this study. With the asymmetric pre-swirl stator attached to the target ship, a self-propulsion analysis was carried out at a design speed of 24 kt for each case, as shown Table 4.

3.2.1 Governing equations

The RANS equations were used for this study, considering unsteady, incompressible, and viscous flows, which can be expressed as follows.

$$\frac{\partial u_i}{\partial x_i} = 0$$

(5)

$$\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + \frac{\partial}{\partial x_i} \left( -\rho \overline{u_i u_j} \right)$$

(6)

where $u_i$, $p$, $t$, $\rho$, $\mu$, and $-\rho \overline{u_i u_j}$ correspond to the speed, pressure, time, density, dynamic viscosity coefficient, and Reynolds stress tensor, respectively.

3.2.2 Numerical method

To resolve the coupling of the velocity and pressure, the semi-implicit method for pressure-linked equations consistent (SIMPLE) algorithm was used, and the realizable $k$-$\varepsilon$ model was chosen as the turbulence model. This improved the performance for the boundary layer separation flow with an adverse pressure gradient. In addition, a sliding interface moving mesh for direct rotation was used to rotate the propeller during the numerical analysis of the self-propulsion. Table 6 shows the details of the analysis conditions.
Table 6 Analysis conditions

<table>
<thead>
<tr>
<th>Program</th>
<th>StarCCM+(Ver.9.04)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Governing equation</td>
<td>Incompressible RANS</td>
</tr>
<tr>
<td>Discretization</td>
<td>Cell-centered FVM</td>
</tr>
<tr>
<td>Turbulence model</td>
<td>Realizable k-ε model</td>
</tr>
<tr>
<td>Wall function</td>
<td>Non-equilibrium</td>
</tr>
<tr>
<td>Velocity–pressure coupling</td>
<td>SIMPLE algorithm</td>
</tr>
<tr>
<td>Rotation method</td>
<td>Sliding interface moving mesh</td>
</tr>
<tr>
<td>Yp+</td>
<td>100</td>
</tr>
<tr>
<td>Number of cells</td>
<td>approx. 5,000,000</td>
</tr>
<tr>
<td>Time step</td>
<td>0.002 s</td>
</tr>
<tr>
<td>Physical time</td>
<td>50 s</td>
</tr>
</tbody>
</table>

The analysis method proposed by Choi et al. [17] was used for the performance comparison. The analysis results corresponding to both the propeller rotation speeds were used to determine the model total resistance ($R_{TM}^{SP}$), thrust ($T_M$), torque ($Q_M$), and number of revolutions ($n_M$) at the self-propulsion point. The wetted surface area was considered to determine the towing force (TF) corresponding to stator. To compare the performance, the delivered power of the model ship ($P_{DM}$) was determined as follows.

$$\text{TF} = R_{TM}^{SP} - T_M$$

$$PD_M = 2\pi n_M Q_M$$

3.2.3 Boundary conditions and grid system

Fig. 10 shows the boundary conditions employed in the present study. We used a velocity inlet condition for the entrance and outer boundary regions and a pressure outlet condition for the exit region. The computational domain was from mid-ship to $-1.5 \leq x/L_{pp} \leq 3.0$ in the x direction, $-1.5 \leq Y/L \leq 1.5$ in the y direction, and $-1.5 \leq Z/L \leq 1.5$ in the z direction.
The grid system was constructed with trimmer grids using automatic grid generation, which is provided in Star CCM+. Approximately 3,300,000 propeller cells are obtained around the ship using approximately 1,700,000 grids, as shown in Fig. 11. The volume-of-fluid (VOF) method was employed, which is widely used to analyze free-surface flow. This method is known to be robust with high validity and reliability for the analysis of complex nonlinear free-surface flows in relation to marine structures and wave-making problems of ships. Also the pressure or axial velocity field along a y=0 cut on the ship stern as shown in Fig. 12.

Fig. 11 View of the generated grids for the hull, propeller and pre-swirl stator

Fig. 12 The pressure or axial velocity field on the ship stern
3.3 Experimental setup

Model testing was conducted in the towing tank at PNU. The length, width, and depth of the tank are 100 m, 8 m, and 3.5 m, respectively. The maximum speed of the towing tank carriage was 5 m/s.

![Towing carriage and tank at PNU](image)

Fig. 13 Towing carriage and tank at PNU

The model size used for the model test was identical to that used for the numerical analysis. To verify the optimum pitch angle of the asymmetric pre-swirl stator, a controllable-pitch-angle type is employed, making it possible to control the pitch angle, as shown in Fig. 14. Fig. 15 shows the experimental setup for the self-propulsion test.

![Controllable pitch angle stator](image)

Fig. 14 Controllable pitch angle stator
The model tests were conducted in accordance with Froude’s law of similarity. The measurement speed in the self-propulsion test was 24 kts.

4. Results and discussion

4.1 Validation of experiment

First, the bare hull experimental results have been compared with the Korea Research Institute of Ships & Ocean Engineering (KRISO) model test results. Fig. 16 shows the comparison of the total resistance coefficient (CTM) at model scale of the PNU and KRISO model tests. There is a good correspondence between PNU and KRISO.

![Comparison of CTM of model towing carriages and tanks at PNU and KRISO](image-url)
Table 7 Relative error of CTM between PNU and KRISO

<table>
<thead>
<tr>
<th></th>
<th>Bare Hull $R_{TM}(N)$</th>
<th>With Stator $R_{TM}(N)$</th>
<th>Diff. (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PNU</td>
<td>45.75</td>
<td>46.78</td>
<td>2.25</td>
</tr>
<tr>
<td>KRISO</td>
<td>45.45</td>
<td>46.30</td>
<td>1.87</td>
</tr>
<tr>
<td>Diff. (%)</td>
<td>-0.66</td>
<td>-1.03</td>
<td></td>
</tr>
</tbody>
</table>

The validation of the present model test has been conducted by comparing the measured wake field with the computed result as shown in Fig 17. The results were similar in terms of the axial average velocity distribution and tangential-direction vector.

![Numerical computation and Model test](image)

Fig. 17 Experimental and numerical wake distributions

The resistance and self-propulsion test result with and without stator were also compared with the CFD results whose results were very similar as shown in Table 8-9.

Table 8 Comparison of numerical and experimental resistance results

<table>
<thead>
<tr>
<th></th>
<th>Bare Hull $R_{TM}(N)$</th>
<th>With Stator $R_{TM}(N)$</th>
<th>Diff. (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CFD</td>
<td>45.75</td>
<td>46.78</td>
<td>2.25</td>
</tr>
<tr>
<td>EFD</td>
<td>45.45</td>
<td>46.30</td>
<td>1.87</td>
</tr>
<tr>
<td>Diff. (%)</td>
<td>-0.66</td>
<td>-1.03</td>
<td></td>
</tr>
</tbody>
</table>
Table 9 Comparison of numerical and experimental self-propulsion results

<table>
<thead>
<tr>
<th></th>
<th>$n_M (RPS)$</th>
<th>$Q_M (Nm)$</th>
<th>$2\pi n_M Q_M (W)$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Bare Hull</td>
<td>With Stator</td>
<td>Diff. (%)</td>
</tr>
<tr>
<td>CFD</td>
<td>11.219</td>
<td>11.071</td>
<td>-1.32</td>
</tr>
<tr>
<td>EFD</td>
<td>10.900</td>
<td>10.650</td>
<td>-2.29</td>
</tr>
<tr>
<td>Difference (%)</td>
<td>-2.84</td>
<td>-3.80</td>
<td>-</td>
</tr>
</tbody>
</table>

The optimally designed pitch angle of the asymmetric pre-swirl stator was also verified through a numerical analysis and a model test, and the results are shown in Table 10.

Table 10 Comparison of numerical and experimental delivered horsepower according to pitch angles

<table>
<thead>
<tr>
<th>Case No.</th>
<th>1st Blade</th>
<th>2nd Blade</th>
<th>3rd Blade</th>
<th>4th Blade</th>
<th>$2\pi n_M Q_m [W]$ Numerical</th>
<th>$2\pi n_M Q_m [W]$ Experimental</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Optimally designed pitch angle using potential-flow program</td>
<td>78.12</td>
<td>77.29</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>$+2^\circ$</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>81.35</td>
<td>79.29</td>
</tr>
<tr>
<td>3</td>
<td>$-2^\circ$</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>81.20</td>
<td>79.21</td>
</tr>
<tr>
<td>4</td>
<td>$0^\circ$</td>
<td>$+2^\circ$</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>81.64</td>
<td>79.62</td>
</tr>
<tr>
<td>5</td>
<td>$0^\circ$</td>
<td>$-2^\circ$</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>80.63</td>
<td>78.23</td>
</tr>
<tr>
<td>6</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>$+2^\circ$</td>
<td>$0^\circ$</td>
<td>81.87</td>
<td>80.54</td>
</tr>
<tr>
<td>7</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>$-2^\circ$</td>
<td>$0^\circ$</td>
<td>80.44</td>
<td>80.26</td>
</tr>
<tr>
<td>8</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>$+2^\circ$</td>
<td>79.84</td>
<td>78.16</td>
</tr>
<tr>
<td>9</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>$0^\circ$</td>
<td>$-2^\circ$</td>
<td>80.20</td>
<td>78.65</td>
</tr>
</tbody>
</table>

Fig. 18 Comparison of delivered power between CFD and EFD
We compared the self-propulsion performances as represented by the delivered horsepower according to the variation in the stator pitch angle as calculated by both the model test and numerical computation. As shown in Table 9, the optimally designed pitch angle was verified by both methods, and their trends in resistance are qualitatively and quantitatively similar, although the viscous effect was considered as a flat plate in the potential-flow code. The potential-flow design program seems sufficiently accurate because the tangential flow potential-flow behavior of the stator is dominant in the flow field. There is no difference in resistance for each case, as shown in Table 11. Accordingly, the efficiency difference can be thought as the difference of delivered power can be compared at the model scale.

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Numerical computation $R_{TM}(N)$</th>
<th>Diff. (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>46.78</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>46.90</td>
<td>0.26</td>
</tr>
<tr>
<td>3</td>
<td>46.85</td>
<td>0.15</td>
</tr>
<tr>
<td>4</td>
<td>46.91</td>
<td>0.28</td>
</tr>
<tr>
<td>5</td>
<td>46.84</td>
<td>0.13</td>
</tr>
<tr>
<td>6</td>
<td>46.91</td>
<td>0.28</td>
</tr>
<tr>
<td>7</td>
<td>46.83</td>
<td>0.11</td>
</tr>
<tr>
<td>8</td>
<td>46.80</td>
<td>0.04</td>
</tr>
<tr>
<td>9</td>
<td>46.82</td>
<td>0.09</td>
</tr>
</tbody>
</table>

We compared the wake distribution at a point ($x/L_{pp}=0.993$) between the propeller and the rudder in Cases 1 and 6, as shown in Fig. 19. Overall, the rotational energy recovered in the hub area is higher in Case 1.

Additionally, We compared the axial and tangential average velocity in Bare(Case 1) and w/PSS(Case 6) as shown in Fig. 20. Overall, the rotational energy recovered over each $r/R$ is higher in bare hull(Case 1).
The streamline around the 2nd blade is visualized in Fig. 21, which demonstrates that the flow escapes smoothly in Case 1. Meanwhile, in Case 6, a separation is observed. The difference between the delivered power of Cases 1 and 6 is also understood by this phenomenon.

As the optimum pitch angle is rather sensitive to the propulsion efficiency, it should be carefully determined according to each ship wake and propeller.

5. Conclusions

An asymmetric pre-swirl stator is a device attached to the front of a propeller. It improves the efficiency by recovering the rotational energy and is a representative energy-saving device that can be attached to any type of ship. For a non-uniform flow, the determination of the pitch angle of each blade is critical in the design of an efficient asymmetric pre-swirl stator.

In the present study, to verify the pitch angle of an asymmetric pre-swirl stator designed by using the potential-flow program, the optimal pitch angle was verified through numerical analysis and a model test. The model delivered power at a design speed of 24 kts was compared by changing the pitch angle by ±2° with respect to previously designed angle.

In the numerical analysis, the model delivered power was compared through a self-propulsion analysis by using the commercial code Star-CCM+. For the model test, the ship and asymmetric pre-swirl stator used in the numerical analysis were manufactured at the same scale and were tested in a towing tank at PNU. The manufactured model stator was a controllable-pitch-angle type to verify the optimum angle in the self-propulsion test.
The results of both the numerical analysis and model tests showed that the pitch angle can be optimized using this model. The potential-flow design program seems sufficiently accurate because the tangential flow potential-flow behavior by the stator is dominant in the flow field.

In the near future, the full-scale performance with an asymmetric stator is expected to be investigated using CFD simulations. The difference between the optimum pitch angle of the model- and full-scale studies might not be so large that the tangential portion (potential-flow) becomes dominant, as shown in the present study.

ACKNOWLEDGMENTS

This work was supported by a National Research Foundation of Korea (NRF) grant funded by the Korea government (MSIP) through GCRC-SOP (No.2011-0030013).

REFERENCES


Validation of Optimally Designed Stator-Propeller System by EFD and CFD

Yong Jin Shin, Moon Chan Kim, Jin Gu Kang, Hyeon Ung Kim, I Rok Shin


Accepted: 22.08.2019.

Yong Jin Shin, dydwlstls@hanmail.net
Moon Chan Kim
Jin Gu Kang, Hyeon Ung Kim, I Rok Shin
Department of Naval Architecture and Ocean Engineering, Pusan National University, South Korea
A DECISION-SUPPORT TOOL FOR DEMOLITION SALE OF A VESSEL

UDC 339.165.4
Review paper

Summary

The fundamental goal of any business is to create value for its owners. In shipping, the value is not only created with freight income, but also with the trade of the vessel itself. A ship has a limited lifetime and can be traded in different markets. The lowest value it will ever receive is its scrap price. An owner may decide to sell a vessel to scrap due to various reasons together with her physical condition and age. In this paper, a fuzzy Analytic Hierarchy Process based decision model is used to provide practitioners with a decision support tool for demolition sale versus further trading of a vessel. The usage of the tool is further illustrated with five actual cases.

Key words: maritime management; demolition sale; ship management; shipping; decision support tools; fuzzy analytic hierarchy process; scoring guide; actual cases

1. Introduction

World seaborne trade accounts for more than 80% of the volume of the total world merchandise [1]. World seaborne trade reached to 10.7 billion tons and is transported by 1,746.4 million dead-weight tons (dwt) of world fleet [1]. Even though there are many factors that determine the market conditions at any given time, the supply and the demand have a big influence on the shipping markets [2].

Shipping is affected by almost every economic and geopolitical crisis in the world. As a result of the 2008 global financial crisis, the world layup capacity has reached its highest levels ever, and even newer vessels are being demolished [3].

It is a capital-intensive industry and a good knowledge of a ship’s life cycle and its management is crucial in avoiding financial losses or making financial gains that can be substantial. Shipping is one of the few industries where the main capital asset, the ship itself, is traded [4]. Shipowners are therefore constantly required to make the decision of ordering, buying, selling or scrapping a vessel, which needs a close market follow up and forecasting. However, as Kou and Luo [3] stated, supported with a review of the existing literature, there is only a limited number of studies on modelling the strategic decisions behind ship investments.
A shipowner in the global shipping industry operates in four closely associated markets: the newbuilding market where new vessels are traded, the sale and purchase market for trading second-hand ships, the freight market for trading sea transport and the demolition market for trading scrap vessels [5], [6]. An alternative categorization for shipping markets is given by Wijnolst and Wergeland [7] where they distinguish between ‘real’ markets for ships, which are the newbuilding market, the demolition market and the spot freight market; and ‘auxiliary’ markets for time charters and second-hand vessels. Since the same shipowners are trading in all of the shipping markets they are interrelated and any change in one will affect the others [8]. Ship demolition is a strategic decision used to balance the fleet capacity in the shipping industry [9].

Considering all these, this study aims to provide the decision makers with a decision support tool for making demolition decisions. The tool is built on a hierarchical model having 4 main and 13 sub-criteria. Three international experts, Turkey Country Manager of RINA Group, Chief Shipping Analyst of BIMCO (Baltic and International Maritime Council), and Director of Braemar Acm Shipbroking Group Limited, filled in pairwise comparisons for those criteria. Buckley’s fuzzy AHP method is then used for prioritization yielding the model with weights. A scoring guide for the decision makers is added to the model to help them in evaluating the demolition candidates against aforementioned 13 sub-criteria. To illustrate the practical usage of the decision support tool, and testing the reliability of it, demolition scores are calculated for 5 actual scenarios. The results are compared with the compromised decisions of a board of shipowners giving a perfect match in the final decisions.

It should be noted that this decision support tool cannot be used in situations where rules and regulations restrict the vessels from further trading.

The rest of the study is organized as follows: Section 2 presents a literature review. In Section 3, methodology of this study is defined. In Section 4, the decision-support tool for demolition decision is introduced. It includes detailed information about the proposed model, weights of the criteria, a scoring guide for the decision makers, and explains how a demolition decision can be given for a ship. Section 5 illustrates the usage of the tool with five actual scenarios. At the end, a conclusion is given in Section 6.

2. Literature Review

Shipping is a cyclical industry and speed reduction, lay-up and scrapping are the main three methods of capacity adjustment [10]. The world fleet growth is established by the delivery of new ships and scrapping the old ones. Shipowners have to continuously make a capacity adjustment decision in this very volatile industry. Historically, high lay-up volumes had been observed during the 1930s world economic depression and shorter but deeper recessions of 1958 and early 1980s. However; in the long lasting depression the shipping industry is in since 2008 financial crises, there has been historically low lay-up rates but high scrapping rates.

There are many factors in deciding the scrapping decision. Hess et al. [11], in their report conducted for the US Navy, explain that the decision whether a ship lives or dies is economical. Yin and Fan [9] underline the effects of ship obsolescence, technological changes and environmental regulations on demolition decision. They also consider the significant effects of operating costs and the state of the shipping market on that decision [9]. Buxton [12] argues that scrap sale decision of an owner is due to the state of both second-hand and freight markets. When both of the markets are in recession and there is no operating income, the owner has to either lay-up or scrap his/her vessel. Mikelis [13] and Kagkarakis et al. [14] point out a positive correlation between the demolition prices and freight markets and claim that demolition price a vessel gets is not only determined by the shipping markets but mostly due to demand for the steel itself.
The price offers the shipowner will receive for an end-of-life (EOL) ship will be affected by many factors like the geographical position of the ship, her physical condition, items remaining on board such as bunkers, and deal terms such as ‘on delivery’, ‘as-is, where-is’ [15]. However, the main determinant of the price will be based on the type of the vessel and the quantity of the steel available on the vessel since the major portion of the weight of the ship (60% to 80%) is steel [16]. Therefore, the scrap price is sensitive to changes in the steel price [17].

The world fleet increased from 766 billion dwt to 1,428 billion dwt between the years 2002-2012 [1]. The increase in the fleet (supply) occurred as a result of the growth in the world trade [18]. The financial crises of 2008 caused rapid decline in the world trade which led to millions of tons of decrease in the world trade (demand) causing imbalance in the world fleet (supply and demand) which then led to a sharp decline in both asset prices and the freight rates.

Despite of the fact that an average shipping cycle is considered to be around eight years [6], it is not easy to forecast the bottom or peek of the market. Although there exist various types of vessels, due to the cargo they transport are employed in different types of trade, shipping cycles affect them all [19].

3. Research Methodology

In this study, a hierarchical model consisting of four main criteria and thirteen sub-criteria is presented. The model is expected to help practitioners as the main part of a ready-to-use decision support tool. Hence, to proceed with the tool, it is needed to define the priorities of those main and sub-criteria.

In the evaluation process of these components, available information is mostly subjective and imprecise. Hence, experts prefer to use natural language expressions rather than sharp numerical values in their evaluations. Modelling using fuzzy logic offers a preferred systematic approach in such situations [20], [21]. Moreover, advantages of using fuzzy numbers in dealing with the inexact information inherent in transportation problems have already been highlighted [22].

Fuzzy AHP is a widely used technique especially for calculating the relative weights of some criteria [15]. Aydin et al. [23], for example, use it to weigh the customer satisfaction criteria while they are studying the problems of rail transit customers in Istanbul. Çakıroğlu et al. [24] use it to select a suitable tugboat alternative given the type of a propulsion system. Li et al. [25] use it for developing a composite business efficiency index score and a composite service effectiveness index score for urbanized areas to compare transit efficiency and effectiveness. Demirel et al. [26] use a hybrid method composed of fuzzy AHP and ELECTRE to select the most effective roll stabilizing system to be used in a trawler type fishing boat. Although they do not use a fuzzy method, in the further research part of their conclusion, Özceylan et al. [27] mention some shortcomings of their research, and advise researchers to adopt fuzzy multi criteria decision making approaches like fuzzy AHP to cope with fuzziness.

Considering these, fuzzy AHP is decided to be the weighing approach for the criteria in this study. There are several fuzzy AHP methods offered in the literature out of which Buckley's [28] was preferred. It is an easy extension of classical AHP [29] to the fuzzy case. Moreover, it guarantees a unique solution to the reciprocal comparison matrix [30].

The steps of the methodology can be summarized as follows:

**Step 1.** To proceed with the method, firstly, a questionnaire is formed to gather the preferences of experts via pairwise comparisons done in linguistic expressions. These expressions and corresponding triangular fuzzy numbers \( \tilde{a}_{ij} = (l, m, u) \) are shown in Table 1.
Table 1 Linguistic Scale and Corresponding TFNs for Pairwise Comparisons

<table>
<thead>
<tr>
<th>Linguistic scale</th>
<th>Triangular fuzzy number (Where criterion $i$ is preferred to criterion $j$)</th>
<th>Triangular fuzzy number (Where criterion $j$ is preferred to criterion $i$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equally important</td>
<td>(1, 1, 3)</td>
<td>(1/3, 1, 1)</td>
</tr>
<tr>
<td>Weakly important</td>
<td>(1, 3, 5)</td>
<td>(1/5, 1/3, 1)</td>
</tr>
<tr>
<td>Essentially important</td>
<td>(3, 5, 7)</td>
<td>(1/7, 1/5, 1/3)</td>
</tr>
<tr>
<td>Very strongly important</td>
<td>(5, 7, 9)</td>
<td>(1/9, 1/7, 1/5)</td>
</tr>
<tr>
<td>Absolutely important</td>
<td>(7, 9, 9)</td>
<td>(1/9, 1/9, 1/7)</td>
</tr>
</tbody>
</table>

If two criteria are perfectly indistinguishable, only then they are considered “just equal”, and the corresponding TFN for this case is (1, 1, 1).

**Step 2.** The pairwise comparisons are then placed in decision matrices such as

$$
\tilde{A} = \begin{bmatrix}
(1,1,1) & \tilde{a}_{12} & \ldots & \tilde{a}_{1n} \\
\tilde{a}_{21} & (1,1,1) & \ldots & \tilde{a}_{2n} \\
\vdots & \vdots & \ddots & \vdots \\
\tilde{a}_{n1} & \tilde{a}_{n2} & \ldots & (1,1,1)
\end{bmatrix}
$$

(1)

where $\tilde{a}_{ij}$ represents the triangular fuzzy number corresponding to the comparison of criteria $i$ and $j$ by the decision maker. The items below the diagonal line are calculated such as $\tilde{a}_{ji} = \tilde{a}_{ij}^{-1}$ or, with a clearer notation,

for $\forall \tilde{a}_{ij} = (l, m, u)$, $\tilde{a}_{ji} = \left(\frac{1}{u}, \frac{1}{m}, \frac{1}{l}\right)$.

(2)

**Step 3.** All the matrices are checked for consistency. To check the individual decision matrices for consistency, the values within them are defuzzified first [31], [32]. Given the triangular fuzzy number $\tilde{d} = (l, m, u)$, the crisp number ($\mu_{\tilde{d}}$) representing this fuzzy number is calculated by using Eq. (3).

$$
\mu_{\tilde{d}} = \frac{l + 2m + u}{4}
$$

(3)

Consistency ratio (CR) for the matrix at hand is then calculated using Eqs. (4), and (5);

$$
CI = \frac{\lambda_{\text{max}} - n}{n - 1}
$$

(4)

$$
CR = \frac{CI}{RI}
$$

(5)

where $CI$ is the consistency index, $\lambda_{\text{max}}$ refers to the largest eigenvector of the matrix, $n$ represents the number of criteria, and $RI$ is the corresponding random index for $n$ criteria. CR values lower than 0.1 are accepted satisfactory indicating a consistent evaluation. If any of the matrices is found inconsistent, initial linguistic evaluations need to be revised until a consistent decision matrix is reached.

**Step 4.** Next step is the aggregation of the individual matrices as in Eq (6);
\[
\tilde{a}_{ij}^{Aggregate} = \sqrt{\prod_{k=1}^{m} \tilde{a}_{ij}^{DM_k}}
\]

where \(\tilde{a}_{ij}^{Aggregate}\) denotes the aggregated fuzzy comparison value of criterion \(i\) to criterion \(j\) in the aggregated matrix, \(m\) is the number of decision makers involved in the decision, \(\tilde{a}_{ij}^{DM_k}\) is the \(\tilde{a}_{ij}\) value for \(k^{th}\) decision maker (\(DM_k\)) for \(k = 1, 2, \ldots, m\), and \(\otimes\) is the fuzzy multiplication sign.

**Step 5.** Then, the fuzzy weight matrix is calculated by Buckley’s Method as adapted from [33] as follows:

\[
\tilde{r}_i = \sqrt{\tilde{a}_{i1} \otimes \tilde{a}_{i2} \otimes \cdots \otimes \tilde{a}_{in}}
\]

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

where \(\tilde{r}_i\) is the geometric mean of fuzzy comparison value of criterion \(i\) to each criterion, \(\tilde{w}_i\) is the weight of criterion \(i\), and \(\oplus\) is the fuzzy addition sign.

**Step 6.** After the calculation of the fuzzy weights, defuzzification and normalization are applied simultaneously by using Eq (9):

\[
W_r = \frac{w_{r_l}^n + 2w_{r_m}^n + w_{r_u}^n}{\sum_{i=1}^{n} (w_{r_l}^i + 2w_{r_m}^i + w_{r_u}^i)}
\]

where the importance weight of \(r^{th}\) criterion, \(w_r\), is a crisp number, \(n\) is the total number of the criteria, and \(w_{r_l}, w_{r_m}\) and \(w_{r_u}\) are the lower (smallest likely value), medium (the most probable value), and upper (the largest possible value) weight values of the triangular fuzzy number \(\tilde{w}_r\), respectively.

4. Decision-support Tool for Demolition Decision

The proposed decision support tool consists of a decision model having prioritized decision criteria, a scoring guide, and a calculation table for overall demolition scores for vessels under focus.

4.1 Proposed Model

All ships have a trading life, and the product life cycle for a commercial ship is about 30 years [13]. However, there are many determinants for the decision of the timing of the scrap sale of a vessel other than age. General economic aspects, current conditions in shipping market, physical condition of the vessel (ship specific issues) and new rules and regulations make up the four main criteria that affect the demolition decision of a shipowner. Those main criteria can further be broken down into sub-criteria as illustrated in Figure 1.
For constituting a common understanding, the criteria and sub-criteria mentioned in the model can be explained as below:

**Economic Aspects**

Like in every major investment decision, market conditions such as the condition of the freight markets, commodity prices and finance markets affect all the investment decisions of a shipowner.

*Scraper Steel Price:*

The scrap value of a vessel is a direct function of the demand for scrap steel which in turn defines the price of it, and scrapping costs [34], [35]. Since the main aim of a shipowner is to get the highest price possible for his asset, the ship, an owner with a vessel close to EOL needs to follow scrap steel price and trends closely for the best possible timing of the scrap sale.

*Bunker Price:*

The owners’ decision to scrap a vessel mainly depends on the difference between expected future income of the vessel and the cost of maintaining her. Bunker costs account for the largest part of the voyage costs; therefore, high bunker costs can easily force the shipowners towards a scrap sale of a vessel making it infeasible to trade it further [36].

*Finance Cost:*

Finance cost is an important determinant in a capital-intensive industry like shipping. Higher interest rates (libor) increase the cost of shipping loans and increased finance cost might weaken the owner’s cash flow position leading the owner towards the scraping decision [10].

**Market conditions**

Freight market conditions also have an important impact on the decision of scrapping a vessel. A synthetic analysis of the world trade shows that world seaborne trade is strongly linked to the world economy [37]. Also, supply-demand balance within the industry has a direct impact on the demolition decisions. The shipping trade primarily depends on the cargo that needs to be...
transported (i.e. demand) and ships available to transport the cargo (i.e. supply) [38]. Demand for shipping (freight rates) is dependent on the world fleet size (supply). Excess supply leads to lower freight market, which pushes the older vessels out of business leading the owner towards a scrapping decision.

**Orderbook:**

Orderbook provides the shipowners with information on the future supply of vessels which is used for the decision making process for further trading and sale and purchase decisions for the existing vessels in their fleet. High level of deliveries, ceteris paribus, is considered to be an indication of lower future freight rates which again leads the shipowners to make decision regarding older tonnage vessels that they possess [39]. An owner would be reluctant to pass necessary surveys for further trading of the vessel if there is an excess amount of new vessels coming to the market.

**Total tonnage scrapped:**

The second element in the supply side of the supply-demand balance is total tonnage scrapped. This variable should be scored considering the total dwt of vessels scrapped a year ago. A high tonnage here indicates a shrink in the supply, thus it decreases the probability of further demolition.

**Freight Market:**

One of the major factors that drive the market for scrapping ships is the freight market since higher the spot rates lower the probability of scrapping as the shipowners would like to make the most out of their assets. So, the freight market can be considered as a market where the shipowner trades his risk. A shipowner would trade his vessel in voyage charter (spot) if he were optimistic about the future, which would then allow him to earn higher freight rates; and fix his vessel in a long-term time charter to secure his future earnings if he were pessimistic about future market rates. In one hand, during a market boom, the shipowner would be willing to trade his vessel until the last day her survey allows and would even be willing to pass the necessary intermediate or special survey as long as he has positive cash flow. On the other, in a depressed, low freight market, vessels as young as 15 years can go to scrap depending not only on the scrap metal prices but also on future expectations of the freight markets [19].

**Cargo Availability:**

Cargo volume is one of the major determinants of the shipping market as it has huge and immediate effect on demand for vessels. When there is an increase in cargo volume, demand for vessels increase, which then affect the freight markets immediately in favour of the owners. A shipowner would be willing to continue trading his old vessel only if he has access to cargo that brings positive cash flow.

**Charterers’ Vessel Selection Criteria**

Charterers have their own criteria for selecting a vessel to transport their cargoes. In the good freight market, they are ready to take the most competitively priced vessel whereas if the freight market is down and is in favour of the charterers they tend to have more criteria for selecting a vessel. If the charterers’ selection criteria no longer allow employments for an older vessel, the shipowner might be forced to a scrap sale.

**Ship Specific Issues**

Although the UNCTAD review of maritime transport categorizes vessels under basic categories such as tankers, bulk carriers, general cargo ships, container ships, and other ships, every ship has her own characteristics besides their type, tonnage and age [40]. Those ship specific issues influencing the demolition sale decision can be explained as below.
Age Restrictions for Further Trading:

An older vessel would have less debt and when she is competing for a cargo she can create a positive cash flow with relatively lower freight rate than a new vessel which usually has a higher income requirement. Nevertheless, an older vessel can trade longer only if there are no surveys due and is creating positive cash flow for her owner. Even the insurance cost alone is significantly higher for an older vessel compared to a new one.

Technical Condition of the Vessel:

Age is not necessarily the only sign of the technical condition of the vessel. Regardless of her age, every vessel has a different technical condition. The shipyard the vessel was built, maintenance, the cargoes transported, and the accidents occurred all have an effect on the technical condition of the vessel. Also, some owners are known for operating older tonnage and since this is their expertise, they are more economical and better in taking care of older vessels.

Bunker Consumption:

Bunker consumption of a vessel depends not only on the sailing speed but also on the design and structure of the ship [41]. In a competitive and relatively low freight market, a vessel that consumes more bunker will require higher freight rate for brake-even.

Regulations

Regulations are one of the major forces that push shipowners sell their vessels to scrap instead of second-hand sale or further trading.

Newly Imposed Rules and Regulations by IMO:

IMO is the global standard-setting authority for the safety, security and environmental performance of international shipping. Therefore, all the players of the shipping industry must follow any new rules and regulations set by IMO. Sometimes these new rules and regulations require a massive investment, which might require many years to compensate and cannot be justified for older tonnage. The shipowners would then have to sell their vessels to scrap instead of making these large investments for further trading.

Flag State Requirements:

Every vessel has a flag state and each flag state has its own rules and regulations. The flag state choice is influenced by several factors like hostile trade partners, strict regulations, competitive advantages, and national taxation [42]. If there is a new rule and regulation by the flag state of a vessel, which cannot justify the investment to comply with the regulation, the shipowner will again be forced to sell his vessel for scrap.

4.2 Weights of the Criteria

After building the model, a questionnaire was prepared for pairwise comparisons. The questionnaire was sent to three international experts, Turkey Country Manager of RINA Group, Chief Shipping Analyst of BIMCO (Baltic and International Maritime Council), and Director of Braemar Acm Shipbroking Group Limited. Via an interactive process, experts filled in the questionnaires. The weights of the main and sub-criteria are then calculated following the steps explained in the methodology. The calculation of the weights of main criteria, as an example, are given together with the results below to numerically illustrate the process. Readers could also refer to [43] and [44] for some other numerical examples on the same methodology.

Step 1 and 2: Individual pairwise comparison values from the experts are collected via surveys. The linguistic variables are then converted into triangular fuzzy numbers as given for main criteria in Table 2.
A Decision-Support Tool for Demolition Sale of a Vessel

Basak Akdemir, Ahmet Beskese

Table 2. Pairwise comparisons of the experts with regard to main criteria

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Economic Aspects</th>
<th>Market Conditions</th>
<th>Ship Specific Issues</th>
<th>Regulations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economic Aspects</td>
<td>1.000</td>
<td>1.000</td>
<td>0.143</td>
<td>0.333</td>
</tr>
<tr>
<td>Market Conditions</td>
<td>3.000</td>
<td>7.000</td>
<td>1.000</td>
<td>1.000</td>
</tr>
<tr>
<td>Ship Specific Issues</td>
<td>0.143</td>
<td>0.200</td>
<td>0.111</td>
<td>0.143</td>
</tr>
<tr>
<td>Regulations</td>
<td>5.000</td>
<td>7.000</td>
<td>9.000</td>
<td>1.000</td>
</tr>
</tbody>
</table>

Step 3. All the matrices are defuzzified and checked for consistency.

The defuzzified evaluation of Expert 1 for the pairwise comparison of Economic Aspects and Ship Specifications, for example, is calculated as $\frac{3+2\times 5+7}{4} = 5$

The complete defuzzified matrix for Expert #1 for the main criteria, as an example, is given in Table 3.

Table 3. Defuzzified evaluation matrices of the experts with regard to main criteria

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Economic Aspects</th>
<th>Market Conditions</th>
<th>Ship Specific Issues</th>
<th>Regulations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economic Aspects</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
<td>0.111</td>
</tr>
<tr>
<td>Market Conditions</td>
<td>5.000</td>
<td>7.000</td>
<td>9.000</td>
<td>1.000</td>
</tr>
<tr>
<td>Ship Specific Issues</td>
<td>0.111</td>
<td>0.143</td>
<td>0.200</td>
<td>0.333</td>
</tr>
<tr>
<td>Regulations</td>
<td>3.000</td>
<td>7.000</td>
<td>9.000</td>
<td>1.000</td>
</tr>
</tbody>
</table>

The inconsistency levels of the evaluation matrices for the main criteria are then calculated as 0.096 for Expert #1, 0.098 for Expert #2, and 0.085 for Expert #3 proving that the evaluations are consistent since each value is lower than 0.1.

This step is applied to every single evaluation matrix of each expert. Once a result greater than 0.1 is found, the evaluations are revised by the experts until an acceptable consistency level is reached.

Step 4. To be able to end up with a group decision on the weights of the main and sub-criteria, the matrices are aggregated using geometric mean as in Eq. (6). The aggregated decision matrix for the main criteria is given in Table 4.

The fuzzy pairwise comparison of Economic Aspects and Ship Specific Issues within Table 4, for example, can be calculated as:

$a_{13} = (\sqrt[3]{3.000 \times 0.333 \times 0.200}, \sqrt[3]{5.000 \times 1.000 \times 0.333}, \sqrt[3]{7.000 \times 1.000 \times 1.000})$

$= (0.585, 1.186, 1.913)$
Table 4. Aggregated evaluation (decision) matrix for main criteria

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Economic Aspects</th>
<th>Market Conditions</th>
<th>Ship Specific Issues</th>
<th>Regulations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economic Aspects</td>
<td>1.000</td>
<td>1.000</td>
<td>1.000</td>
<td>0.251</td>
</tr>
<tr>
<td>Market Conditions</td>
<td>1.442</td>
<td>2.268</td>
<td>3.979</td>
<td>1.000</td>
</tr>
<tr>
<td>Ship Specific Issues</td>
<td>0.523</td>
<td>0.643</td>
<td>1.710</td>
<td>0.251</td>
</tr>
<tr>
<td>Regulations</td>
<td>1.442</td>
<td>2.268</td>
<td>3.979</td>
<td>1.000</td>
</tr>
</tbody>
</table>

Step 5. The fuzzy weights are calculated for main and sub-criteria using Eqs. (7) and (8).

To numerically illustrate the calculation process, calculation of the weights of the main criteria is given as follows:

Considering $\tilde{r}_i(r_{il}, r_{im}, r_{iu})$:

$r_{1l} = \sqrt[4]{a_{11l} \times a_{12l} \times a_{13l} \times a_{14l}} = \sqrt[4]{1.000 \times 0.251 \times 0.585 \times 0.251} = 0.4384$

$r_{1m} = \sqrt[4]{a_{11m} \times a_{12m} \times a_{13m} \times a_{14m}} = \sqrt[4]{1.000 \times 0.441 \times 1.186 \times 0.441} = 0.6929$

$r_{1u} = \sqrt[4]{a_{11u} \times a_{12u} \times a_{13u} \times a_{14u}} = \sqrt[4]{1.000 \times 0.693 \times 1.913 \times 0.693} = 0.9793$

$\tilde{r}_1 = (0.4384, 0.6929, 0.9793)$.

After following the same process, all $\tilde{r}_i$ values for the main criteria are calculated as follows:

$\tilde{r}_2 = (1.2009, 1.7711, 2.7394)$,

$\tilde{r}_3 = (0.4480, 0.6929, 0.9724)$,

$\tilde{r}_4 = (0.8380, 1.1760, 1.9396)$.

Using these $\tilde{r}_i$ values, fuzzy weights ($\tilde{w}_i(w_{il}, w_{im}, w_{iu})$) of the main criteria can be calculated such that:

$w_{1l} = \frac{r_{1l}}{\sum_{i=1}^{4} r_{iu}} = \frac{0.4384}{0.9793 + 2.7394 + 0.9724 + 1.9396} = 0.0661$,

$w_{1m} = \frac{r_{1m}}{\sum_{i=1}^{4} r_{im}} = \frac{0.6929}{0.6929 + 1.7711 + 0.6929 + 1.1760} = 0.1599$,

$w_{1u} = \frac{r_{1u}}{\sum_{i=1}^{4} r_{ir}} = \frac{0.9793}{0.4384 + 1.2009 + 0.4480 + 0.8380} = 0.3348$.

Thus, $\tilde{w}_1 = (0.0661, 0.1599, 0.3348)$.

Following the explained operations, fuzzy weights of the main criteria can be found as:

$\tilde{w}_1 = (0.0661, 0.1599, 0.3348)$,

$\tilde{w}_2 = (0.1811, 0.4088, 0.9364)$,

$\tilde{w}_3 = (0.0676, 0.1599, 0.3324)$,

$\tilde{w}_4 = (0.1264, 0.2714, 0.6630)$.

Step 6. After the calculation of the fuzzy weights, the crisp weights for the main criteria are calculated as given below via defuzzification and normalization as in Eq. (9):

$w_1 = 0.153$,  $w_2 = 0.411$,  $w_3 = 0.153$,  $w_4 = 0.283$.

To illustrate the calculation process, $w_1$, for example, can be calculated such that:

$w_1 = \frac{w_{1l} + 2w_{1m} + w_{1u}}{\sum_{i=1}^{4} (w_{il} + 2w_{im} + w_{iu})} = \frac{0.0661 + 2 \times 0.1599 + 0.3348}{0.0661 + 2 \times 0.1599 + 0.3348 + \ldots + 0.1264 + 2 \times 0.2714 + 0.6630} = 0.153$
For the sub-criteria, calculated local weights are multiplied by the weight of the parent main criteria to yield the global weights. The weights of all the criteria within the model are presented in Table 5.

**Table 5.** Weights of the criteria

<table>
<thead>
<tr>
<th>Main Criteria</th>
<th>Weights</th>
<th>Sub-criteria</th>
<th>Local Weights</th>
<th>Global Weights</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economic Aspects</td>
<td>0.153</td>
<td>Scrap Steel Price</td>
<td>0.363</td>
<td>0.056</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Bunker Price</td>
<td>0.277</td>
<td>0.042</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Finance Cost</td>
<td>0.360</td>
<td>0.055</td>
</tr>
<tr>
<td>Market Conditions</td>
<td>0.411</td>
<td>Orderbook</td>
<td>0.051</td>
<td>0.085</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Total tonnage scrapped</td>
<td>0.207</td>
<td>0.021</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Freight Market</td>
<td>0.314</td>
<td>0.129</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Cargo Availability</td>
<td>0.334</td>
<td>0.137</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Charterers’ Vessel Selection Criteria</td>
<td>0.094</td>
<td>0.039</td>
</tr>
<tr>
<td>Ship Specific Issues</td>
<td>0.153</td>
<td>Age Restrictions for Further Trading</td>
<td>0.574</td>
<td>0.088</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Technical Condition of the Vessel</td>
<td>0.216</td>
<td>0.033</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Bunker Consumption</td>
<td>0.210</td>
<td>0.032</td>
</tr>
<tr>
<td>Regulations</td>
<td>0.283</td>
<td>Newly Imposed Rules and Regulations by IMO</td>
<td>0.553</td>
<td>0.156</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Flag State Requirements</td>
<td>0.447</td>
<td>0.127</td>
</tr>
</tbody>
</table>

4.3 Demolition Decision for a Ship

Given the aforementioned model, the score levels for all the sub-criteria those would be used in the evaluation of any demolition sale decision (i.e. the scoring guide) are given in Table 6. In case of a demolition evaluation, decision makers should easily evaluate their own ship under focus using the score guide and assign a score (\(S_i\); for \(i=1, 2, 3, \ldots, 13\), indicating the criterion number) from 1 to 5 for each criterion.

Weighted score for each criterion (\(WS_i\)), and total demolition score (TDS) can then be calculated by using Eqs. (10) and (11), consequently, as shown in Table 7.

\[
WS_i = w_i \times S_i \quad (10)
\]

\[
Total \ Demolition \ Score \ (TDS) = \sum_{i=1}^{13} WS_i \quad (11)
\]

where \(i\) represents the number of each sub-criterion.

It should be noted that a higher score in TDS would lead to demolition sale.

To validate the scoring scheme and threshold score, highly experienced decision makers are told some cases. They were asked to evaluate the situations at hand in a panel discussion, and come up with one single decision: Demolition or further trading. During the panel, they were not equipped with the tool. Same cases were evaluated simultaneously with the tool and the results were compared.

During this session, the threshold for demolition score is decided to be 2.5. It means that if the calculate demolition score is below 2.5, the final decision will be further trading the ship. Scores equal to or larger than 2.5 will lead to a demolition decision.
Table 6. The scoring guide

<table>
<thead>
<tr>
<th>Main Criteria</th>
<th>Sub-Criteria</th>
<th>Conditions</th>
<th>Score</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Economic Aspects</strong></td>
<td>Scrap Steel Price (SSP)</td>
<td>SSP &lt; $200 per LWT</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$200 ≤ SSP &lt; $300 per LWT</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$300 ≤ SSP &lt; $400 per LWT</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$400 ≤ SSP &lt; $500 per LWT</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>SSP ≥ $500 per LWT</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Bunker Price</td>
<td>IFO price &lt; $400 per m/ton</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$400 ≤ IFO price &lt; 500 per LWT per m/ton</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$500 ≤ IFO price &lt; 600 per LWT per m/ton</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$600 ≤ IFO price &lt; 700 per LWT per m/ton</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>IFO price ≥ $700 per m/ton</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Finance Cost</td>
<td>Libor &lt; 2%</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>2% ≤ Libor &lt; 3%</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3% ≤ Libor &lt; 4%</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4% ≤ Libor &lt; 5%</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Libor ≥ 5%</td>
<td>5</td>
</tr>
<tr>
<td><strong>Market Conditions</strong></td>
<td>Orderbook</td>
<td>Increase in total dwt ordered &lt; 12%</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>12% ≤ Increase in total dwt ordered &lt; 14%</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>14% ≤ Increase in total dwt ordered &lt; 16%</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>16% ≤ Increase in total dwt ordered &lt; 18%</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Increase in total dwt ordered ≥ 18%</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Total tonnage scrapped</td>
<td>Total dwt vessels scrapped &gt; 40,000,000</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>40,000,000 ≥ Total dwt vessels scrapped &gt; 30,000,000</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>30,000,000 ≥ Total dwt vessels scrapped &gt; 20,000,000</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>20,000,000 ≥ Total dwt vessels scrapped &gt; 10,000,000</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Total dwt vessels scrapped ≤ 10,000,000</td>
<td>5</td>
</tr>
<tr>
<td><strong>Freight Market</strong></td>
<td>Baltic Dry Index &gt; 2,000</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>2,000 ≥ Baltic Dry Index &gt; 1,500</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1,500 ≥ Baltic Dry Index &gt; 1,000</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1,000 ≥ Baltic Dry Index &gt; 500</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Baltic Dry Index ≤ 500</td>
<td>5</td>
</tr>
<tr>
<td><strong>Cargo Availability</strong></td>
<td>Total cargo increase &gt; 3%</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>3% ≥ Total cargo increase &gt; 2%</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>2% ≥ Total cargo increase &gt; 1.5%</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1.5% ≥ Total cargo increase &gt; 1%</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Total cargo increase is ≤ 1%</td>
<td>5</td>
</tr>
<tr>
<td><strong>Charterers' Vessel Selection Criteria</strong></td>
<td>It is never a problem</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>It creates very little trouble</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>It creates problem occasionally</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>It creates problem in some regions</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>It creates problem in every fixture</td>
<td>5</td>
</tr>
</tbody>
</table>
### Table 6. The scoring guide (continued)

<table>
<thead>
<tr>
<th>Main Criteria</th>
<th>Sub-Criteria</th>
<th>Conditions</th>
<th>Score</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ship Specific</td>
<td>Age Restrictions for Further Trading</td>
<td>Vessels’ age &lt; 10</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>10 ≥ Vessels’ age &gt; 15</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>15 ≥ Vessels’ age &gt; 20</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>20 ≥ Vessels’ age &gt; 25</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Vessels’ age ≥ 25</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Technical Condition of the Vessel</td>
<td>Technical condition of the vessel is very good</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Technical condition of the vessel is good</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Technical condition of the vessel is mediocre</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Technical condition of the vessel is bad</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Technical condition of the vessel is very bad</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Bunker Consumption</td>
<td>Bunker consumption &lt; 4 ton/day</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>4% ≤ Bunker consumption &lt; 6%</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6% ≤ Bunker consumption &lt; 8%</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8% ≤ Bunker consumption &lt; 10%</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Bunker consumption ≥ 10 ton/day</td>
<td>5</td>
</tr>
<tr>
<td>Regulatons</td>
<td>Newly Imposed Rules and Regulations by IMO</td>
<td>Cost of implementation &lt; $250,000</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$250,000 ≤ Cost of implementation &lt; $500,000</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$500,000 ≤ Cost of implementation &lt; $750,000</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$750,000 ≤ Cost of implementation &lt; $1,000,000</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Cost of implementation ≥ $1,000,000</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Flag State Requirements</td>
<td>Cost of implementation &lt; $250,000</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$250,000 ≤ Cost of implementation &lt; $500,000</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$500,000 ≤ Cost of implementation &lt; $750,000</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$750,000 ≤ Cost of implementation &lt; $1,000,000</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Cost of implementation ≥ $1,000,000</td>
<td>5</td>
</tr>
</tbody>
</table>

### Table 7. Demolition Score Calculation Table

<table>
<thead>
<tr>
<th>Sub-Criteria</th>
<th>Weight ($w_i$)</th>
<th>Score ($S_i$)</th>
<th>Weighted Score ($WS_i$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Scrap Steel Price</td>
<td>0.056</td>
<td>$S_1$</td>
<td>$WS_1=0.056* S_1$</td>
</tr>
<tr>
<td>2. Bunker Price</td>
<td>0.042</td>
<td>$S_2$</td>
<td>$WS_2=0.042* S_2$</td>
</tr>
<tr>
<td>3. Finance Cost</td>
<td>0.055</td>
<td>$S_3$</td>
<td>$WS_3=0.055* S_3$</td>
</tr>
<tr>
<td>4. Orderbook</td>
<td>0.085</td>
<td>$S_4$</td>
<td>$WS_4=0.085* S_4$</td>
</tr>
<tr>
<td>5. Total tonnage scrapped</td>
<td>0.021</td>
<td>$S_5$</td>
<td>$WS_5=0.021* S_5$</td>
</tr>
<tr>
<td>6. Freight Market</td>
<td>0.129</td>
<td>$S_6$</td>
<td>$WS_6=0.129* S_6$</td>
</tr>
<tr>
<td>7. Cargo Availability</td>
<td>0.137</td>
<td>$S_7$</td>
<td>$WS_7=0.137* S_7$</td>
</tr>
<tr>
<td>8. Charterers' Vessel Selection Criteria</td>
<td>0.039</td>
<td>$S_8$</td>
<td>$WS_8=0.039* S_8$</td>
</tr>
<tr>
<td>9. Age Restriction for Further Trading</td>
<td>0.088</td>
<td>$S_9$</td>
<td>$WS_9=0.088* S_9$</td>
</tr>
<tr>
<td>10. Technical Condition of the Vessel</td>
<td>0.033</td>
<td>$S_{10}$</td>
<td>$WS_{10}=0.033* S_{10}$</td>
</tr>
<tr>
<td>11. Bunker Consumption</td>
<td>0.032</td>
<td>$S_{11}$</td>
<td>$WS_{11}=0.032* S_{11}$</td>
</tr>
<tr>
<td>12. Newly Imposed Rules and Regulations by IMO</td>
<td>0.156</td>
<td>$S_{12}$</td>
<td>$WS_{12}=0.156* S_{12}$</td>
</tr>
<tr>
<td>13. Flag State Requirements</td>
<td>0.127</td>
<td>$S_{13}$</td>
<td>$WS_{13}=0.127* S_{13}$</td>
</tr>
</tbody>
</table>

Total Demolition Score = $\sum_{i=1}^{13} WS_i$
5. Application of the decision-support tool

To illustrate the decision-support tool and verifying its validity, five actual cases (evaluations for five different vessels) are given as below.

Vessel # 1

The first case scenario takes place in August 2009. The vessel analysed is a 1984 built, 17,850 dwt multi-purpose roll on roll off vessel (MPP). It was purchased and passed her special survey and was fixed for a long term time charter to a first class Far Eastern charterer right before the Lehman Brothers financial crises of 2008. After her first voyage, the market collapsed. First class Charterers immediately cancelled the contract and started arbitration proceedings in London. The shipowner was left with an old vessel, which had just passed her special survey in a good trading condition. Shipowner had three choices: further trading, second-hand sale or scrap sale. After five spot shipments the vessel was still earning far below her operating expenses. The scrap price was $260 and the bunker price at Fujairah was $ 372.80. The USD libor was 1%. 14,630,000 dwt of the total world fleet was scrapped in 2008. There was 55.5% increase in the total world fleet and Baltic Dry Index was 2,685. Total seaborne cargo had decreased 4% and every fixture was a problem since there was no cargo for her specific trade. She was 25 years old at the time of decision and was in a good condition. She used to consume 8 tons/day more than a new vessel and did not require any additional cost for neither IMO nor Flag State requirements.

Considering these, scores for Vessel #1 can be defined as in Table 8.

<table>
<thead>
<tr>
<th>Evaluation sub-criteria</th>
<th>Actual value</th>
<th>Score</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Scrap Steel Price</td>
<td>$ 260.00</td>
<td>2</td>
</tr>
<tr>
<td>2. Bunker Price</td>
<td>$ 372.80</td>
<td>1</td>
</tr>
<tr>
<td>3. Finance Cost</td>
<td>1%</td>
<td>1</td>
</tr>
<tr>
<td>4. Orderbook</td>
<td>55.5%</td>
<td>5</td>
</tr>
<tr>
<td>5. Total tonnage scrapped</td>
<td>14,630,000 dwt</td>
<td>4</td>
</tr>
<tr>
<td>6. Freight Market</td>
<td>2,685</td>
<td>1</td>
</tr>
<tr>
<td>7. Cargo Availability</td>
<td>- 4%</td>
<td>5</td>
</tr>
<tr>
<td>8. Charterers' Vessel Selection Criteria</td>
<td>It creates a problem in every fixture</td>
<td>5</td>
</tr>
<tr>
<td>9. Age Restriction for Further Trading</td>
<td>25</td>
<td>5</td>
</tr>
<tr>
<td>10. Technical Condition of the Vessel</td>
<td>Good</td>
<td>2</td>
</tr>
<tr>
<td>11. Bunker Consumption</td>
<td>8 tons/day</td>
<td>4</td>
</tr>
<tr>
<td>12. Newly Imposed Rules and Regulations by IMO</td>
<td>None</td>
<td>1</td>
</tr>
<tr>
<td>13. Flag State Requirements</td>
<td>None</td>
<td>1</td>
</tr>
</tbody>
</table>

Vessel # 2

In 2006, the owner had a 25-year old Panamax (75,000dwt) vessel with special survey due. The scrap steel price was $345 and the bunker was trading at $310. The libor was 6%. 6,100,000 dwt of total world fleet was scrapped in 2005. There was 26.91% increase in the total world fleet and Baltic Dry Index was 2,718. There was 4.0% increase in the total seaborne cargo. The vessel was in a bad condition. There was never a problem fixing her. She used to consume no more than 2 tons/day extra than a new vessel and did not require any additional cost for neither IMO nor Flag State requirements.

Considering these, scores for Vessel #2 can be defined as in Table 9.
Table 9. Scores for Vessel #2

<table>
<thead>
<tr>
<th>Evaluation sub-criteria</th>
<th>Actual value</th>
<th>Score</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Scrap Steel Price</td>
<td>$ 345.00</td>
<td>3</td>
</tr>
<tr>
<td>2. Bunker Price</td>
<td>$ 310.00</td>
<td>1</td>
</tr>
<tr>
<td>3. Finance Cost</td>
<td>6%</td>
<td>5</td>
</tr>
<tr>
<td>4. Orderbook</td>
<td>26.91%</td>
<td>5</td>
</tr>
<tr>
<td>5. Total tonnage scrapped</td>
<td>6,100,000 dwt</td>
<td>5</td>
</tr>
<tr>
<td>6. Freight Market</td>
<td>2,718</td>
<td>1</td>
</tr>
<tr>
<td>7. Cargo Availability</td>
<td>4%</td>
<td>1</td>
</tr>
<tr>
<td>8. Charterers' Vessel Selection Criteria</td>
<td>There was never a problem fixing her</td>
<td>1</td>
</tr>
<tr>
<td>9. Age Restriction for Further Trading</td>
<td>25</td>
<td>5</td>
</tr>
<tr>
<td>10. Technical Condition of the Vessel</td>
<td>Bad</td>
<td>4</td>
</tr>
<tr>
<td>11. Bunker Consumption</td>
<td>2 tons/day</td>
<td>1</td>
</tr>
<tr>
<td>12. Newly Imposed Rules and Regulations by IMO</td>
<td>None</td>
<td>1</td>
</tr>
<tr>
<td>13. Flag State Requirements</td>
<td>None</td>
<td>1</td>
</tr>
</tbody>
</table>

Vessel #3

In 2008 the owner had a 1983 built product tankers, (25,000 dwt). Special survey of the vessels was due. The scrap steel price was $630 and the bunker was trading at $509. The libor was 3%. 6,400,000 dwt of total world fleet was scrapped in 2007. There was 52.11% increase in the total world fleet and Baltic Dry Index was 8,053. There was 4.3% increase in the total seaborne cargo and it was impossible to fix her. She was 25 years old and was in a bad condition. She used to consume no more than 2 tons/day extra than a new vessel. There was no steel renewal requirement. The cost of newly imposed rules and regulations by IMO was expected to be around $1.5 million. Flag state requirements were estimated as $1.2 million.

Considering these, scores for Vessel #3 can be defined as in Table 10.

Table 10. Scores for Vessel #3

<table>
<thead>
<tr>
<th>Evaluation sub-criteria</th>
<th>Actual value</th>
<th>Score</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Scrap Steel Price</td>
<td>$ 630.00</td>
<td>5</td>
</tr>
<tr>
<td>2. Bunker Price</td>
<td>$ 509.00</td>
<td>3</td>
</tr>
<tr>
<td>3. Finance Cost</td>
<td>3%</td>
<td>3</td>
</tr>
<tr>
<td>4. Orderbook</td>
<td>52.11%</td>
<td>5</td>
</tr>
<tr>
<td>5. Total tonnage scrapped</td>
<td>6,400,000 dwt</td>
<td>5</td>
</tr>
<tr>
<td>6. Freight Market</td>
<td>8,053</td>
<td>1</td>
</tr>
<tr>
<td>7. Cargo Availability</td>
<td>4.3%</td>
<td>1</td>
</tr>
<tr>
<td>8. Charterers' Vessel Selection Criteria</td>
<td>It was impossible to fix her</td>
<td>5</td>
</tr>
<tr>
<td>9. Age Restriction for Further Trading</td>
<td>23</td>
<td>4</td>
</tr>
<tr>
<td>10. Technical Condition of the Vessel</td>
<td>Very bad</td>
<td>5</td>
</tr>
<tr>
<td>11. Bunker Consumption</td>
<td>2 tons/day</td>
<td>1</td>
</tr>
<tr>
<td>12. Newly Imposed Rules and Regulations by IMO</td>
<td>$1.5 million</td>
<td>5</td>
</tr>
<tr>
<td>13. Flag State Requirements</td>
<td>$1.2 million</td>
<td>5</td>
</tr>
</tbody>
</table>

Vessel #4

In 2017, the owner had a 1999 built 46,000 dwt dry bulk vessel with intermediate survey due. The scrap steel price was $325 and the bunker was trading at $357. The libor was 1%.
44,400,000 dwt of total world fleet was scrapped in 2016. There was 6.5% increase in the total world fleet and Baltic Dry Index was 740. There was 2.4% increase in the total seaborne cargo. Charterers never had a problem fixing her. She was 18 years old and was in a good condition. The market was slightly improving. Orderbook was coming down in favour of the owner. She used to consume no more than 2 tons/day extra than a new vessel. IMO regulations would not be due until next special survey, which would be due in 2020 which meant no incurring costs for IMO or Flag State requirements.

Considering these, scores for Vessel #4 can be defined as in Table 11.

<table>
<thead>
<tr>
<th>Evaluation sub-criteria</th>
<th>Actual value</th>
<th>Score</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Scrap Steel Price</td>
<td>$325.00</td>
<td>3</td>
</tr>
<tr>
<td>2. Bunker Price</td>
<td>$357.00</td>
<td>1</td>
</tr>
<tr>
<td>3. Finance Cost</td>
<td>1%</td>
<td>1</td>
</tr>
<tr>
<td>4. Orderbook</td>
<td>6.5%</td>
<td>1</td>
</tr>
<tr>
<td>5. Total tonnage scrapped</td>
<td>44,400,000 dwt</td>
<td>1</td>
</tr>
<tr>
<td>6. Freight Market</td>
<td>740</td>
<td>4</td>
</tr>
<tr>
<td>7. Cargo Availability</td>
<td>2.4%</td>
<td>2</td>
</tr>
<tr>
<td>8. Charterers’ Vessel Selection Criteria</td>
<td>It is never a problem</td>
<td>1</td>
</tr>
<tr>
<td>9. Age Restriction for Further Trading</td>
<td>18</td>
<td>3</td>
</tr>
<tr>
<td>10. Technical Condition of the Vessel</td>
<td>Good</td>
<td>2</td>
</tr>
<tr>
<td>11. Bunker Consumption</td>
<td>2 tons/day</td>
<td>1</td>
</tr>
<tr>
<td>12. Newly Imposed Rules and Regulations by IMO</td>
<td>None</td>
<td>1</td>
</tr>
<tr>
<td>13. Flag State Requirements</td>
<td>None</td>
<td>1</td>
</tr>
</tbody>
</table>

**Vessel #5**

In 2003 the owner had a 1981 built Panamax combination carrier (75,000 dwt) vessel, which had a major accident in the British Channel. The vessel was due to pass intermediate survey and the cost of repair was $5,000,000 and would take 3 months to repair. The scrap steel price was $178 and the bunker was trading at $166. The libor was 1%. 29,470,000 dwt of total world fleet was scrapped in 2002. There was 14.32% increase in the total world fleet and Baltic Dry Index was 1694. There was 3.0% increase in the total seaborne cargo. Charterers never had a problem fixing her. She was 22 years old and was in a very bad condition. She used to consume no more than 2 tons/day extra than a new vessel. The value of the vessel due to the accident decreased to scrap value. However, there were no additional costs due to newly imposed rules and regulations.

Considering these, scores for Vessel #5 can be defined as in Table 12.

All those scores are then fed in Table 13 for further calculations as defined in Section 4.3.

**Decisions about the vessels**

**Vessel #1:** Demolition score is 2.644. This score means that the owner needs to give a scrap decision. This decision is parallel with the actual decision given by the owner. In reality, there was no second-hand buyer and the owner was left with the only alternative, scrap sale.

**Vessel #2:** Demolition score is 2.207. This score means that the owner can further trade the ship. In reality, the owner had the following options: spend an important sum of money to pass special survey for further trading, second-hand sale or scrap sale. He decided to pass her special survey even though the survey required 800 tons of steel renewal because the freight market was at record high due to supply shortage in the dry bulk fleet. As a result of a good decision, he benefited for a couple more years from the good freight market.
Table 12. Scores for Vessel #5

<table>
<thead>
<tr>
<th>Evaluation sub-criteria</th>
<th>Actual value</th>
<th>Score</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Scrap Steel Price</td>
<td>$178.00</td>
<td>1</td>
</tr>
<tr>
<td>2. Bunker Price</td>
<td>$166.00</td>
<td>1</td>
</tr>
<tr>
<td>3. Finance Cost</td>
<td>1%</td>
<td>1</td>
</tr>
<tr>
<td>4. Orderbook</td>
<td>14.32%</td>
<td>3</td>
</tr>
<tr>
<td>5. Total tonnage scrapped</td>
<td>29,470,000 dwt</td>
<td>3</td>
</tr>
<tr>
<td>6. Freight Market</td>
<td>1,694</td>
<td>2</td>
</tr>
<tr>
<td>7. Cargo Availability</td>
<td>3%</td>
<td>2</td>
</tr>
<tr>
<td>8. Charterers’ Vessel Selection Criteria</td>
<td>It is never a problem</td>
<td>1</td>
</tr>
<tr>
<td>9. Age Restriction for Further Trading</td>
<td>22</td>
<td>4</td>
</tr>
<tr>
<td>10. Technical Condition of the Vessel</td>
<td>Very bad</td>
<td>5</td>
</tr>
<tr>
<td>11. Bunker Consumption</td>
<td>2 tons/day</td>
<td>1</td>
</tr>
<tr>
<td>12. Newly Imposed Rules and Regulations by IMO</td>
<td>None</td>
<td>1</td>
</tr>
<tr>
<td>13. Flag State Requirements</td>
<td>None</td>
<td>1</td>
</tr>
</tbody>
</table>

Vessel #3: Demolition score is 3.526. This score means the owner should give a scrap decision right away. Since the score is way too high, the decision seems an easy one.

Vessel #4: Demolition score is 1.845. The owner should continue trading the vessel until the next special survey.

Vessel #5: Demolition score is 1.874. The score means that the owner should continue trading the ship. In reality, the owner decided to spend the money and repair the vessel because he saw the opportunity in the coming freight market due to the unbalance in supply and demand.

6. Conclusion

Shipowners’ income arises from their vessels and they naturally aim to receive maximum possible income from their assets. However, shipping is a cyclical business, which is very difficult to forecast, and investments do not always perform as initially planned. There are many factors that need to be considered when making a shipping investment decision and in this paper a decision support tool has been developed to aid the shipowners in making a scrap sale decision.

To build the tool, first, a model having four main and thirteen sub-criteria was constituted. During the process, literature review and actual expertise of some shipowners were utilized. Then the criteria and sub-criteria were weighed via pairwise comparisons of three well known international experts. Also, the calculation process for the weights of the criteria was explained in detail illustrated with some numerical examples to let practitioners use their own preference evaluations, if they desire to do so. Then, a scoring guide was prepared in line with the criteria. The tool was completed with a Demolition Score Calculation Table (see Table 13).

A shipowner is only expected to know the actual values/facts for his vessel and the environment. Thus, he can score the case at hand against thirteen criteria using the scoring guide. Finally, with simple mathematical operations such as addition and multiplication in Table 13, he can come up with a single demolition score for his vessel. If the score is less than 2.5, the decision will be further trading the ship, else the vessel needs to be sold for scrap. To help the practitioners with scoring, 5 actual cases were given in detail, and results were briefly discussed.

For future research, the model can further be customized for specific types of vessels (e.g. ro-ro, container, etc.).
Table 1: Scoring matrix for 5 ships

<table>
<thead>
<tr>
<th></th>
<th>Ship #1</th>
<th></th>
<th></th>
<th>Ship #2</th>
<th></th>
<th></th>
<th>Ship #3</th>
<th></th>
<th></th>
<th>Ship #4</th>
<th></th>
<th></th>
<th>Ship #5</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Weight</td>
<td>Score</td>
<td>Weighted Score</td>
<td>Score</td>
<td>Weighted Score</td>
<td>Score</td>
<td>Weighted Score</td>
<td>Score</td>
<td>Weighted Score</td>
<td>Score</td>
<td>Weighted Score</td>
<td>Score</td>
<td>Weighted Score</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Scrap Steel Price</td>
<td>0.056</td>
<td>2</td>
<td>0.112</td>
<td>3</td>
<td>0.168</td>
<td>5</td>
<td>0.280</td>
<td>3</td>
<td>0.168</td>
<td>1</td>
<td>0.056</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bunker Price</td>
<td>0.042</td>
<td>1</td>
<td>0.042</td>
<td>1</td>
<td>0.042</td>
<td>3</td>
<td>0.126</td>
<td>1</td>
<td>0.042</td>
<td>1</td>
<td>0.042</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Finance Cost</td>
<td>0.055</td>
<td>1</td>
<td>0.055</td>
<td>5</td>
<td>0.275</td>
<td>3</td>
<td>0.165</td>
<td>1</td>
<td>0.055</td>
<td>1</td>
<td>0.055</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Orderbook</td>
<td>0.085</td>
<td>5</td>
<td>0.425</td>
<td>5</td>
<td>0.425</td>
<td>5</td>
<td>0.425</td>
<td>1</td>
<td>0.085</td>
<td>3</td>
<td>0.255</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total Tonnage Scrapped</td>
<td>0.021</td>
<td>4</td>
<td>0.084</td>
<td>5</td>
<td>0.105</td>
<td>5</td>
<td>0.105</td>
<td>1</td>
<td>0.021</td>
<td>3</td>
<td>0.063</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Freight Market</td>
<td>0.129</td>
<td>1</td>
<td>0.129</td>
<td>1</td>
<td>0.129</td>
<td>1</td>
<td>0.129</td>
<td>4</td>
<td>0.516</td>
<td>2</td>
<td>0.258</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cargo Availability</td>
<td>0.137</td>
<td>5</td>
<td>0.685</td>
<td>1</td>
<td>0.137</td>
<td>1</td>
<td>0.137</td>
<td>2</td>
<td>0.274</td>
<td>2</td>
<td>0.274</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Charterers’ Vessel Selection Criteria</td>
<td>0.039</td>
<td>5</td>
<td>0.195</td>
<td>1</td>
<td>0.039</td>
<td>5</td>
<td>0.195</td>
<td>1</td>
<td>0.039</td>
<td>1</td>
<td>0.039</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Age Restriction for Further Trading</td>
<td>0.088</td>
<td>5</td>
<td>0.440</td>
<td>5</td>
<td>0.440</td>
<td>4</td>
<td>0.352</td>
<td>3</td>
<td>0.264</td>
<td>4</td>
<td>0.352</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Technical Condition of the Vessel</td>
<td>0.033</td>
<td>2</td>
<td>0.066</td>
<td>4</td>
<td>0.132</td>
<td>5</td>
<td>0.165</td>
<td>2</td>
<td>0.066</td>
<td>5</td>
<td>0.165</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bunker Consumption</td>
<td>0.032</td>
<td>4</td>
<td>0.128</td>
<td>1</td>
<td>0.032</td>
<td>1</td>
<td>0.032</td>
<td>1</td>
<td>0.032</td>
<td>1</td>
<td>0.032</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Newly Imposed Rules and Regulations by IMO</td>
<td>0.156</td>
<td>1</td>
<td>0.156</td>
<td>1</td>
<td>0.156</td>
<td>5</td>
<td>0.780</td>
<td>1</td>
<td>0.156</td>
<td>1</td>
<td>0.156</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Flag State Requirements</td>
<td>0.127</td>
<td>1</td>
<td>0.127</td>
<td>1</td>
<td>0.127</td>
<td>5</td>
<td>0.635</td>
<td>1</td>
<td>0.127</td>
<td>1</td>
<td>0.127</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

2.644  2.207  3.526  1.845  1.874
REFERENCES


PSO-BASED PID CONTROLLER DESIGN FOR SHIP COURSE-KEEPING AUTOPILOT (str.1-15)

Tatijana Dlabač, Martin Ćalasan, Maja Krčum, Nikola Marvučić
Original scientific paper

DETERMINATION OF LINEAR AND NONLINEAR ROLL DAMPING COEFFICIENTS OF A SHIP SECTION USING CFD (str.17-33)

Soon-Seok Song, Sang-Hyun Kim, Kwang-Jun Paik
Original scientific paper

CONTRIBUTION TO THE SEAKEEPING ANALYSIS OF MULTIHULL WARSHIPS (str.35-50)

Rodrigo Perez, Jose M. Riola
Review article

THE TURBULENT BOUNDARY LAYER AND FRICTIONAL DRAG CHARACTERISTICS OF NEW GENERATION MARINE FOULING CONTROL COATINGS (str.51-65)

Burcu Erbas
Original scientific paper
PSO-BASED PID CONTROLLER DESIGN FOR SHIP COURSE-KEEPING AUTOPilot

UDC 629.5.017.3:629.3.027.2
Original scientific paper

Summary

This paper deals with autopilot proportional-integral-derivative (PID) controller design. In the literature, various available methods for PID controller have been presented. Based on the fact that the existing methods do not guarantee the optimal response of the system on the input step change, in this paper we used a valuable technique called Particle Swarm Optimization (PSO) in order to optimally design PID controller taking into account the system limitation such as the value of the rudder angle saturation. Furthermore, we have compared system response on input step and ramp change of input signal for a few PID controller parameters values obtained by using different methods known in literature. It has been proven that by applying the PSO method, it is possible to determine the optimal PID controller parameters which guarantee fast and proper response from the aspect of the minimal overshoot and the minimal settling time. The obtained results confirm the applicability and efficiency of using PSO method for optimal autopilot PID controller design.

Key words: Autopilot; PID controller; Particle Swarm Optimisation (PSO); Rudder angle

1. Introduction

During its movement, vessel is steering a certain course. Operator monitors movement and position of the ship with his senses (sight, feeling for tilt and acceleration, etc.). The status of ship's movement and position is obtained depending on the situation based on the visual perception of certain fixed points as landmarks or based on readings on instruments. Based on obtained information, a person forms a control signal and gives commands from the bridge control panel which are transmitted to the steering gear mechanism. By using steering gear mechanism, helmsman maintains existing course or changes the course (maneuvering) [1]. This type of control is based on the difference between the existing and the desired course, and the rudder mechanism is a part of the automatic control closed-loop [2].
A ship, being an autonomous and highly complex dynamic system, is composed of a series of different processes, machines and devices that lend themselves to automation. The automation of a ship’s processes contributes to its higher efficiency, cutting maintenance and crew costs, prolonging its lifespan and bringing a host of other advantages.

The automatic steering system’s basic task is to maintain the ship’s course, i.e. to maintain a current navigational trajectory. The autopilot is a higher level of control in which the ship’s course is controlled without the participation of a helmsman [2]. By constantly monitoring the actual against the desired course, an error signal is determined, and the microcomputer formulates a steering algorithm that puts the ship back on its desired course.

Proportional-integral-derivative (PID) controllers or proportional-derivative (PD) controllers are usually used in ships autopilot design [3,4]. The PID autopilot was developed with the intention of enabling a vessel to follow course as accurately as possible by decreasing the error caused by excessive deviations of the helm and by simultaneously limiting the rudder deviation in order to minimize rudder skid [5]. As an addition to the fact that a straight course is not the most economical option, it has been decided that helm control must always be optimized relative to the prevailing state of the environment and that a small bandwidth should be used in order to minimize losses. There are different designs of ship course-keeping autopilots. For example, the steering parameters for normal adaptable PID autopilots have been developed during the last three decades as specified in [6-8], and the most important among them is the performance index regarding added resistance due to imperfect steering control. Also, designs based on neural network [9,10], fuzzy logic [11-12], backstepping [13], self-tuning control [14], pole placement technique (PPT) [15], extended state observer technique (ESO) [16] and similar are widely used.

As already mentioned, in [15] an analytical method for determining the PID controller parameters was presented. This method is based on the use of a symmetrical optimum and provides very simple formulation for determining PID controller parameter values. Moreover, the authors were analyzing the effect of step and ramp change of input signal and they obtained a good system response in terms of eliminating the errors in steady/stationary state. On the other hand, in [17], the application of the optimization method to determine the optimal values of PID parameters was obtained. It was considered that optimum values of the PID parameters are inside the range ±10% of the parameter values compared to analytical method given in [15]. Considering the mentioned references [15] and [17], it is important to point out that the steering machine limitation was not taken into account in either study.

On the other hand, a detailed description of the ship’s mathematical model and steering machine limitation is presented in [18] where authors analyzed the application of extended state observer for yaw control, which as output does not give certain values of the PID parameters of the autopilot. In addition to the mentioned works, the use of Lyapunov’s theory in PID controller design is reported in [19]. Specifically, using Lyapunov’s theory in determining the PID controller parameters of autopilot, a very slow system response is obtained, what can be inferred from the results shown in [18]. From the comprehensive literature review in the area, it can be concluded that the above-mentioned works either use complex mathematical apparatus or provide analytical solutions that do not lead to the optimal results. Moreover, according to the researches done in [15,17] no real model of the ship has been taken into account including the steering machine limitation.

However, this paper presents an upgraded investigation on PID controller design that was previously analyzed in [15,17]. The comparison of the system response to the step change in input signal (yaw angle step change), by different methods is presented in [18]. In the available works, the authors did not compare the values of overshoot, rise time, settling time and delay time. Otherwise, these transition process parameters define the efficiency, stability and
response speed of the regulator. Based on the previous analyses, the aim of this paper is to improve the PID controller design from the perspective of better response quality with respect to limitations dictated by the components of the regulation loop (for example, hydraulic pumps, rudder angle etc.). This improvement was achieved by using the optimization technique called Particle Swarm Optimization (PSO), which is a very powerful technique, whose application can be found in a number of areas such as power converters [20], solar cells [21], electrical machines [22], power network [23], etc.

This paper is organized as follows. In Section 2 a mathematical model of a ship and PID controller is proposed. PSO technique with corresponding algorithm and its explanation is described in Section 3. Simulation results with specific values of parameters including robustness analysis are presented in Section 4. In addition, comparative analysis that includes methods from the literature and novel ones provided in this paper are given in Section 5.

2. Mathematical model

It is well known that ships are equipped with autopilots consisting of PID controllers among others, which make part of the automatic control system and their purpose is to maintain a given course of the ship. During the design of the ship, it is very important to install a control circuit and an automatic control system, i.e. autopilot, in order to relieve the helmsman.

PID-based steering course autopilot is one of the most commonly used types of autopilot for navigating the course of the ship. The course autopilot usually contains a basic algorithm for course maintenance, with or without adaptation to navigation conditions, and a maneuver controller. Setting up the parameters of the PID controller is extremely important since the parameters of the ship represent an object of the steering, i.e. the parameters of the ship dynamics change with the speed, position of the rudder, load, etc. The same is also significant due to disturbances such as winds, waves, currents, etc.

Block diagram of overall structure of ship autopilot is shown in Figure 1. In this figure, the rudder angle (or rudder deflection) is denoted with δ and ψ represents ship heading angle which is closed to its desired value ψd. Based on the value of heading angle error ε which appears due to external disturbances, the autopilot generates the input signal for rudder actuator [12].

![Fig. 1 Overall structure of ship autopilot](image)

The rudder actuator represents the actuation mechanism that moves the rudder to the controller commanded angle. However, the response of the rudder actuator is defined by the speed of the rudder, whereas the rudder movement is mechanically limited. The rudder is moved by hydraulic pumps, the speed of which is governed by the pump capacity and by opening the valve. Hydraulic fluid flow regulated by the swash plate of the steering machine is controlled by the telemotor system which receives signals from the autopilot controller. Detailed
description of rudder actuator can be found in [18]. According to the SOLAS Convention [24], the power of the rudder actuator must be sufficient to shift the rudder from 35° on one side, to 30° on the other side in 28 seconds.

As proposed in [15], a PID controller with an additional degree of freedom should be used to control the autopilot. Its mathematical equation is:

\[ H_c = \frac{k_c}{sT_c} \left( sT_c + \frac{sT'_c + 1}{sT_1 + 1} \right) \]  

where: \( k_c \) is PID gain coefficient, \( T_c \) is PID main time constant, and \( T'_c \) and \( T_1 \) are time constants where \( T_1 < T'_c < T_c \). By using this equation, the PID controller is designed by combining the pole placement method with the symmetric optimal criterion [15]. If the natural frequency \( \omega_0 \) and the attenuation coefficient \( \xi \) are known, the unknown parameters of the PID can be obtained by using the following equations:

\[ k_c = \frac{\omega_0}{k_p}, \]  
\[ T_c = \frac{2\xi + 1}{\omega_0}, \]  
\[ T'_c = T_p, \]  
\[ T_1 = \frac{1}{(2\xi + 1)\omega_0}. \]  

Finally, the most widely used model of the ship is based on the Nomoto linear model [18] whose differential equation is as follows

\[ \ddot{\psi}(t) + \frac{1}{T} \dot{\psi}(t) = \frac{k}{T} \delta(t), \]  

where \( \psi \) and \( \delta \) are earlier defined, \( k \) represents the static yaw rate gain and \( T \) is the effective yaw rate time constant. The corresponding transfer function is given by

\[ H_p = \frac{k_p}{s(sT_p + 1)} \]  

Therefore, the plant has a low-order model, which contains a pure integrator, and which is characterized by a dominant time constant \( T_p \) and a gain coefficient \( k_p \).

3. PSO algorithm

An algorithm that is based on the PSO metaheuristic belongs to the category of algorithms inspired by the swarm intelligence. Similar to bird flocking, this method is originally based on a group of particles that are flying among the search space in order to find the best position. Generally speaking, PSO algorithm represents an optimization tool that finds its application in the investigations of solar cells, electrical machines, electronic systems etc. [20-23]. The PSO algorithm is established on the population (swarm) of candidate solutions. Also, each particle represents one candidate solution to the problem and moves around in the search spaces by using its experience, as well as the experience of other particles. The movement of each candidate solution (particle) is defined by the speed that is constantly changing in order to find a better feasible solution. Therefore, each particle is flying through n-dimensional search space in finding the right position according to the mathematical formulation. The aim of the iterative procedure is to enable the particles to find better positions (Figure 2).
For solving different optimization problems, an objective function $J$ should be proposed. In this study, we also define the PSO parameters and variables. The objective function is defined in each particle of search space $A$ as follows:

$$J: A \rightarrow R, \ A \subset R^n$$

This function shows that for each particle of search space $A$ an appropriate value of function $J$ is assigned. The value of variable (position of particle with its velocity) is limited due to constraints in the search space that can be found in each iteration. Since we treat the minimization problem, it means that while the value of objective function is lower, than the position of particle is better. In this phase, we have personal best and global best minimum (value and position). It is worth mentioning that the value of global minimum is common for each particle and very close to the minimum of the objective function.

If we define $x_i$ to be position vector and $v_i$ to be velocity vector of particle $i$, $P_i$ represents its best position while $g$ is a current global optimum, then moving among the search space can be formulated as

$$v_i(t+1) = w \cdot v_i(t) + c_1 r_1 (P_i(t) - x_i(t)) + c_2 r_2 (g(t) - x_i(t))$$

$$x_i(t+1) = x_i(t) + v_i(t+1)$$

\[(8)\]

\[(9)\]
where \( r_1 \) and \( r_2 \) are arbitrary positive numbers between 0 and 1, \( c_1 \) and \( c_2 \) represent accelerate constants, \( w \) is an inertia weight coefficient and \( t \) is a current iteration number. Accelerate constants \( c_1 \) and \( c_2 \) have a huge impact on the convergent speed because in the case that the values of constants are small, the particle swarm slowly converges to the solution. Otherwise, in a situation where the values of constants are relatively high, the whole optimization process may become unstable.

In this study, we estimated the values of four parameters while the objective function is mean square error of the reference signal and estimated signal value. Therefore, the ISE (Integral Square Error) function has been selected as the optimum function, with its mathematic form:

\[
OF_1 = \int_0^\infty e^2(t)dt
\]  

(10)

where \( t \) – is time, and \( e(t) \) – is the difference between the desired ship heading angle \((\psi_d)\) and actual ship heading angle \((\psi)\). In this paper we determined the PID parameters for step change of desired ship heading angle of \(\psi_d=10\) [deg].

However, beside the proposed optimum function used in [17], here we propose the following objective function:

\[
OF_2 = \int_0^\infty e^2(t)dt + G \cdot max(y)
\]  

(11)

that deals with the maximal value of rise. So, the objective and priority are to minimize the value of rise. In this paper, the value of coefficient \( G \) is set to be 10000.

The constraints of the used optimization technique in the paper are \((k_c, T_c, T_c', \text{ and } T_1)\) which must be set within some pre-specified limits. These limits may be bounded by

\[
k_c^{min} \leq k_c \leq k_c^{max}, T_c^{min} \leq T_c \leq T_c^{max}
\]  

(12)

\[
T_c'^{min} \leq T_c' \leq T_c'^{max}, T_1^{min} \leq T_1 \leq T_1^{max}
\]  

(13)

where the superscripts \(min\) and \(max\) speak for the minimum and the maximum values of the respective variables.

The procedure of determining the optimal values of PID controller parameters is described as follows. In its basis, PSO algorithm generates the values of PID controller parameters. Upon that, the value of objective function is reported for the input signal. In the next step (iteration), PSO algorithm generates new values of PID controller parameters with the new value of objective function. In the case that this objective function value is lesser, the algorithm is taking the corresponding PID controller parameter values. The procedure is repeated until the best value of objective function is obtained or upon the total number of reported iterations is finished.

However, during the optimization process, we tested the value of rudder angle \((\delta)\). Namely, if its value is greater than 35, the obtained combination of parameters is rejected. In that manner the contribution of the methodology provided in this paper differs from those proposed in [17] which gives an added value in the research area.
4. Simulation results

The proposed method (PSO algorithm, with limitation of rudder angle $\delta$, together with novel optimization function) is used for PID parameters determination of a ship model whose parameters are as follows: $k_p=-0.0834$, $T_p=5.98$ [15]. In addition, it is assumed that the natural frequency $\omega_0$ is 0.1 rad/s while damping coefficient $\xi$ is 0.9 [15].

The estimated PID parameters values, determined by using several methods, are presented in Table I. Namely, it shows the results obtained by using Nicolau [15], Calasan [17], as well as by using the proposed method based on the use of PSO algorithm together with objective functions OF1 and OF2.

Table 1 Comparison of results in terms of parameters value

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_c$</td>
<td>-1.2</td>
<td>-1.1102</td>
<td>-3.2715</td>
<td>-1.5606</td>
</tr>
<tr>
<td>$T_c$</td>
<td>28</td>
<td>30.8</td>
<td>71.9739</td>
<td>40.1966</td>
</tr>
<tr>
<td>$T'_c$</td>
<td>5.98</td>
<td>6.4569</td>
<td>6.5431</td>
<td>18.7858</td>
</tr>
<tr>
<td>$T_1$</td>
<td>3.57</td>
<td>3.3035</td>
<td>6.1581</td>
<td>8.3948</td>
</tr>
</tbody>
</table>

The step responses of the closed-loop transfer function for all four cases are illustrated in Figure 3a. The corresponding the rudder angle responses are presented in Figure 3b.
Fig. 3  a) Step response of the closed-loop system, b) Corresponding rudder angle responses.

The ramp responses of the proposed system are illustrated in Figure 4a. The corresponding the rudder angle responses are presented in Figure 4b.
As it can be seen, in both cases (step and ramp input signal), PID autopilot assures null stationary error for both step and ramp variations on reference inputs. However, it can be seen that the OF2 enables obtaining much better signal in terms of overshoot. Also, it can be seen that in all analyzed cases the maximal value of rudder angle is less than the prescribed value. Likewise, the higher change of the desired course angle value leads to higher changes of rudder angle.

Table 2 Characteristics of obtained signals

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Overshoot [%]</th>
<th>Time of overshoot [s]</th>
<th>Delay time [s]</th>
<th>Settling time [s]</th>
<th>Rise time [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>ESO [18]</td>
<td>0</td>
<td>-</td>
<td>12.82</td>
<td>&gt; 50</td>
<td>&gt; 25</td>
</tr>
<tr>
<td>Lyapunov [19]</td>
<td>22</td>
<td>25</td>
<td>8.7</td>
<td>&gt; 60</td>
<td>&gt; 10</td>
</tr>
<tr>
<td>IOL [18]</td>
<td>0</td>
<td>-</td>
<td>13</td>
<td>&gt; 49</td>
<td>&gt; 25</td>
</tr>
<tr>
<td>Nicolau [15]</td>
<td>27.3</td>
<td>29.5</td>
<td>8.37</td>
<td>&gt; 65</td>
<td>11</td>
</tr>
<tr>
<td>Calasan [17]</td>
<td>23</td>
<td>32.5</td>
<td>8.43</td>
<td>74</td>
<td>11.9</td>
</tr>
<tr>
<td>Proposed method – simple OF1</td>
<td>35</td>
<td>16.4</td>
<td>5.8</td>
<td>75</td>
<td>7.3</td>
</tr>
<tr>
<td>Proposed method – novel OF2</td>
<td>17</td>
<td>16.1</td>
<td>5.6</td>
<td>54</td>
<td>6.4</td>
</tr>
</tbody>
</table>
It is important to point out that in case when steering machine limitations (rudder angle saturation) is not applied, the following results will be obtained: $K_c = -85.8305$, $T_c = 20.6428$, $T_c' = 7.6052$ and $T_1 = 0$. Although the results guarantee the ideal response – almost minimal overshoot with a very short time-delay, they are not realistic, since the complex ship system dynamics cannot follow the change of the input – control signal with adequate speed.

4.1 Robustness analysis

The observed real system contains different kinds of uncertainties and various disturbances due to its complexity. For that reason, in this paper, the robustness analysis of the observed system with tuned parameters was carried out in three ways – by step changing of reference signal (desired ship heading angle), by combined step and ramp changing of reference signal and by adding certain disturbance signal on the output side of the diagram (on ship heading angle – see Figure 1).

In the first case, experiential values of step changes of input signal (desired ship heading angle) were made several times (Figure 5). First, the reference value of yaw angle was reduced by 30%, then after a certain time this value was increased by 50%, and finally the current value was decreased by 20%. As can be seen, for all given changes of reference values, both positive and negative, the best responses were provided by the proposed method OF2. It is clear that the speed of establishing a new stationary state is the highest, while the overshoot value for all the step changes of the input signal is the smallest. Also, the values of rudder angle changes are in permitted limits (see Figure 5b).

In the second case (Figure 6) the referent value of desired course is firstly changed with step signal and after that with ramp signal (see Figure 6a). Just like in the first case the best responses were provided by the proposed method OF2, while in all cases the rudder angle is within the permitted limits (see Figure 6b).

In the case of testing the effect of the step disturbance at the measured value of the actual course angle, the corresponding results are shown in Figure 7. In this case, the step disturbances were added at the output signal (see Fig. 1 – disturbance signal). As can be seen, the system closely follows all the changes at the output. It can be noticed that the fastest system response is achieved when the parameters used were determined by the proposed method OF2. Moreover, after the disturbance, the system quickly returns to the stationary state (see Figure 7a and 7b). Note, at starting time the step change of desired course angle is also realized.
Fig. 5 a) Heading course responses on step changes of desired course angle value, b) Corresponding rudder angle responses.
Therefore, based on all of the above, it can be concluded that the proposed method for PID parameters design enables very secure tracking of the reference signal as well as very secure disturbance attenuation, without an unallowed value of rudder angle.
Fig. 7 a) Additional signal and system responses, b) Corresponding rudder angle response
5. Conclusion

This paper deals with the design of the PID controllers for autopilots. For that purpose, a block diagram (of the ship and the autopilot controller) was observed, while the design was realized by using a PSO method.

However, unlike previous works in this field, when designing the parameters of the PID regulator, the importance of taking into account the dynamics (limitations) of the system is emphasized. Namely, by taking the steering dynamics limitations into account, the procedure for selecting the parameters of the PID controller is also defined. In addition to criterion functions for determining the parameters of the controller known from the literature, a new criterion function is proposed which takes into account the maximum overshoot value during the rapid change of the control signal. Moreover, the obtained response results for the step disturbance of the input signal are compared with the responses obtained by using several methods known from the literature. It has been shown that the selected parameters of the controller meet the stability criterion, while providing a fast and efficient system response to the effect of the input step and ram disturbances.

In the future work the authors plan to determine the optimal values of PID parameters for different desired values of ship heading angles. Also, we will test different optimization techniques for this purpose.

REFERENCES


Soon-Seok Song  
Sang-Hyun Kim*  
Kwang-Jun Paik

http://dx.doi.org/10.21278/brod70402  
ISSN 0007-215X  
eISSN 1845-5859

DETERMINATION OF LINEAR AND NONLINEAR ROLL DAMPING COEFFICIENTS OF A SHIP SECTION USING CFD

UDC 629.5.017.22  
Original scientific paper

Summary

The most prevalently used method to obtain the nonlinear roll damping coefficient is the free roll decay test. However, this method can only be conducted at the resonance frequency and thus cannot consider the effect of the frequency. This is a certain limitation as the resonance frequency can be changed at any time by the ship’s loading conditions. Therefore, it is worth investigating the frequency dependency of the nonlinear roll damping coefficients. In this study, a numerical method was proposed to derive the linear and nonlinear roll damping coefficients of ships at different frequencies. Fully nonlinear CFD simulations of forced harmonic roll motion were conducted and the roll damping coefficients were calculated. Then, the damping coefficients were decomposed into the linear and nonlinear components using the linear regression analysis. The linear roll damping coefficients were compared with potential coefficients and showed a good agreement, while the nonlinear roll damping coefficients were compared with the coefficients calculated using a semi-empirical method. The nonlinear roll damping coefficients calculated from the proposed method showed a strong frequency dependency. Finally, possible rationales for the frequency dependence of the nonlinear roll damping coefficient were investigated.

Key words: Nonlinear roll damping; computational fluid dynamics; ship motion; forced oscillation

1. Introduction

A ship’s motion in waves affects the habitability and safety of the ship, as well as the operation of equipment and various other activities. Especially roll motion, among the six degrees of freedom motions of ships, is critical as far as ship stability is concerned. The resonance phenomena of the roll motion are associated with problems of instabilities (e.g. synchronous rolling and parametric rolling). Synchronous rolling usually occurs at beam seas when the encounter wave period is close to the natural roll period. Parametric rolling, on the
other hand, often occurs on following and quartering seas. It is caused by inducing instability and an increase in rolling amplitude due to the periodic variation of stability characteristics [1]. For the many years, parametric rolling was considered as an unlikely occurrence, but its importance started to be highlighted after the incident from 1998 when a post-Panamax container ship experienced extreme rolling due to this phenomenon. Since that time, a large number of efforts have been devoted to investigating the features of parametric rolling [1-8].

These resonant phenomena can also occur in other modes if some specific conditions are met. Non-conventional vessels such as SWATH (small-waterplane-area twin hull), whose the restoring capabilities in pitch are not high as those of conventional mono-hulls, can experience resonant motions when the vessel is heaving and pitching [9, 10]. Resonant phenomena can also take place by trapped water between two vertical walls e.g. moonpools [11], ship moored by a bottom mounted terminal [12], or multi-hull configurations [13].

Predicting the ship motions using traditional linear potential flow based solutions can bring sufficient accuracies for most of the modes of motions, due to their weak nonlinearities. However, it is well known that roll damping has strong nonlinearity due to the high contribution of viscous roll damping [2]. Therefore, nonlinear roll damping coefficients should be taken into account for better predictions. Accordingly, a large number of studies have been devoted to the derivations of the nonlinear roll damping coefficients [2, 14-20].

The free roll decay test is one of the most prevalently used methods to determine the nonlinear roll damping coefficients. This method has a simple test procedure and thus simple test equipment. However, it can only be used at the natural frequency of the floating body and cannot provide information about the relation between the damping coefficients and the frequencies of roll motion. This can be a limitation considering that the ships’ natural frequency can always vary by the loading conditions.

On the other hand, the forced harmonic oscillation test can be used to derive damping coefficients and the added mass or added moment of inertia coefficients of all six degrees of freedom. This method can be performed at arbitrary frequencies and amplitudes. This method involves measurements of the hydrodynamic and hydrostatic loads acting on the body while it is forced to oscillate at a selected frequency and amplitude. The coefficients are obtained by decomposing the measured hydrodynamic forces into in-phase and out-phase.

However, there are critical difficulties for forced roll oscillations, as asserted by Vugts [21]. The pure sinusoidal oscillation is not achievable due to the interference of hydrodynamic loads induced by the motion itself. Second, the magnitude of the damping moment is relatively small, compared to other moment components measured together (i.e. the added moment of inertia, restoring moment, and the moment of inertia of the model itself), such that small measurement error can lead to large uncertainties in the calculated damping coefficients.

Recently, with the rapid growth of computational power, there have been increasing number of studies utilizing Computational Fluid Dynamics (CFD) for the derivations of roll damping coefficients. These studies involve either the simulations of free roll decay motions [22-26] or forced harmonic motions [27-36].

The studies using free roll decay simulations [22-26] show good agreements with the experimental results. However, these studies also have the same limitation that the simulation can only be performed at the natural frequencies of the ships, as they use the same mechanism as the physical free roll decay test.

On the other hand, the CFD simulations involving forced motions [27-36] are free from this limitation as the simulations can be performed at arbitrary frequencies. And also many of the above-mentioned difficulties of the forced oscillation tests can be omitted. That is to say, the pure sinusoidal oscillation is achievable in the simulations and the results are free from the
measurement errors. However, none of these studies investigate the frequency dependence of the nonlinear roll damping coefficients.

The literature suggests that there exists no specific study determining the nonlinear roll damping coefficient at various frequencies to investigate the frequency dependence. One may argue that the most critical motions occur at their resonance frequencies and thus it is enough using a constant nonlinear roll damping coefficient, which is obtained from a roll decay test. However, the resonance frequencies of ship roll motions can always vary by the loading conditions. That means the nonlinear roll damping coefficient obtained from the free roll decay test cannot necessarily represent the nonlinear roll damping coefficients at different loading conditions. Accordingly, it is worth investigating the frequency dependence of the nonlinear roll damping coefficients for better understanding of ship motions in waves.

Therefore, the aim of this study is to fill the literature gap by developing a CFD model to simulate forced harmonic roll motions and performing a comprehensive investigation into the behaviors of linear and nonlinear roll damping coefficients at different frequencies.

In this study, a series of CFD simulations of forced harmonic roll with varying roll amplitudes and frequencies. The roll damping coefficients including the viscous effects were obtained from the simulations. For validation purposes, the results were compared with experimental data from a free roll decay test and a forced oscillation test. Then, the obtained roll damping coefficients were decomposed into the linear and nonlinear (quadratic) components using the linear regression analysis. The linear roll damping coefficient showed good agreement with the potential roll damping coefficients calculated using the close-fit conformal mapping. The nonlinear roll damping coefficients showed a decreasing trend with the frequencies. A possible rationale behind the frequency dependency was presented.

2. Methodology

2.1 Motion equation

The roll motion equation with a quadratic damping term is,

\[(I_{xx} + A_{44}) \ddot{\phi} + B_{44i} \dot{\phi} + B_{44v}^{(2)} |\dot{\phi}| \dot{\phi} + C_{44} \phi = F_4\]  

(1)

where \(\ddot{\phi}\), \(\dot{\phi}\), and \(\phi\) are the angular acceleration, angular velocity, and angular displacement of roll motion, respectively. \(I_{xx}\) and \(A_{44}\) are the moment of inertia and the added moment of inertia in rolling. \(B_{44i}\) and \(B_{44v}^{(2)}\) are the linear and nonlinear (viscous) roll damping coefficients, and the superscript \(2\) indicate that his coefficient is used in a quadratic term \(B_{44v}^{(2)} |\dot{\phi}| \dot{\phi}\). \(C_{44}\) is the restoring coefficient, which can be calculated using Equation 2

\[C_{44} = \rho g \nabla \bar{G} M \tan \phi\]  

(2)

where, \(\rho\), \(g\), \(\nabla\), and \(\bar{G} M\) are the density of water, gravitational acceleration, the displacement volume, and the metacentric height of the body, respectively. The quadratic damping term in Equation 1 can be linearized by the following equations,

\[(I_{xx} + A_{44}) \ddot{\phi} + (B_{44i} + B_{44v}^{(1)}) \dot{\phi} + C_{44} \phi = M_{roll}\]  

(3)

\[B_{44v}^{(1)} = \frac{8}{3\pi} \phi_{a} \omega B_{44v}^{(2)}\]  

(4)
in which, the superscript (1) and (2) indicate the orders of the term, i.e. $B_{44}^{(1)}$ is the linearized viscous damping coefficient, whereas $B_{44}^{(2)}$ is the viscous damping in quadratic form. $\phi_a$ and $\omega$ are the roll amplitude and the roll frequency, respectively.

2.2 Forced harmonic roll motion

In the simulations, the body was forced to roll with the roll amplitude, $\phi_a$, and the frequency, $\omega$. The time-dependent forced roll motion is given as,

$$\phi = \phi_a \cos(\omega t)$$  \hspace{1cm} (5)

The hydrodynamic moment can be obtained by subtracting the restoring moment ($C_{44} \phi$) from the total moment acting on the body. The time-dependent hydrodynamic moment, $M$, can be written as,

$$M = M_a \cos(\omega t - \varepsilon)$$  \hspace{1cm} (6)

where, $M_a$ is the amplitude of the hydrodynamic moment and $\varepsilon$ phase shift with respect to the roll motion. Figure 1 schematically illustrates the hydrodynamic moment obtained from the forced roll simulations.

![Fig.1 Hydrodynamic moment acting on the body](image)

The obtained hydrodynamic moment can be divided by the in-phase, $M_{in}$, and out-phase, $M_{out}$, as follows.

$$M_a \cos(\omega t - \varepsilon) = M_{in} \cos(\omega t) + M_{out} \sin(\omega t)$$  \hspace{1cm} (7)

$$M_{in} = M_a \cos(\varepsilon)$$  \hspace{1cm} (8)

$$M_{out} = M_a \sin(\varepsilon)$$  \hspace{1cm} (9)

Then the added moment of inertia, $A_{44}$, and the roll damping coefficient, $B_{44}$, can be determined as

$$A_{44} = \frac{M_{in}}{\phi_a \omega^2}$$  \hspace{1cm} (10)
It is of note that the damping coefficients, $B_{44}$, obtained from the forced roll simulations include the viscous roll damping coefficients ($B_{44} = B_{44l} + B_{44v}^{(2)}$). Therefore, the magnitudes of $B_{44}$ obtained from the forced roll simulations vary with the amplitudes, due to the nonlinearity of the viscous roll damping.

2.3 Linear and Nonlinear Damping Components

In this study, the forced roll simulations were conducted at varying amplitudes, $\phi_a$, at each frequency. Then, the obtained roll damping coefficients, $B_{44}$, varies with the roll amplitudes due to the nonlinearity, as mentioned earlier. The obtained $B_{44}$ values can be divided into the linear and nonlinear components using a linear regression method, as schematically illustrated in Figure 2. Using the intercept, $\alpha$, and the slope, $\beta$, of the linear trend line of the $B_{44}$ values varying with the amplitudes, the linear and nonlinear roll damping coefficients are determined as,

$$B_{44} = \alpha + \beta \phi_a$$

(12)

$$B_{44l} = \alpha$$

(13)

$$B_{44v}^{(2)} = \frac{3\pi}{8\omega} \beta$$

(14)

Fig.2 Roll damping coefficients at different roll amplitudes of oscillation

2.4 Numerical modelling

The proposed CFD model was developed based on the unsteady Reynolds-averaged Navier-Stokes (URANS) method using a commercial CFD software package, STAR-CCM+. The simulations were modelled in two dimensions (2D) to minimise the computational cost. Mesh generation was performed using the built-in automated meshing tool of STAR-CCM+. Trimmed hexahedral meshes were used. A prism layer was used around the body surface as shown in Figure 3.
Figure 4 illustrates the computational domain used for the simulations. The overset-mesh technique was used to simulate the forced roll motions. Both the sides of the basin were modelled with slopes, such that they induce the waves to break. The boundary conditions for the rotating body and the bottom and sides of the tank were defined as no-slip walls, while the top of the computational domain was set to pressure outlet boundary condition.

The volume of fluid (VOF) method was used for the free surface, and the $k - \varepsilon$ model was selected as the turbulence model. The physics continua used in this study are given in Table 1.
Table 1 Physics continua used for the simulation

<table>
<thead>
<tr>
<th>Physics continua</th>
<th>Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time</td>
<td>Implicit Unsteady</td>
</tr>
<tr>
<td>Material</td>
<td>Eulerian Multiphase</td>
</tr>
<tr>
<td></td>
<td>Multiphase Interaction</td>
</tr>
<tr>
<td>Eulerian Multiphase Model</td>
<td>Volume of Fluid (VOF)</td>
</tr>
<tr>
<td></td>
<td>Multiphase Equation of state</td>
</tr>
<tr>
<td></td>
<td>Segregated Flow</td>
</tr>
<tr>
<td>Viscous Regime</td>
<td>Turbulent</td>
</tr>
<tr>
<td></td>
<td>Reynold-Averaged Navier-Stokes</td>
</tr>
<tr>
<td>Reynolds-Averaged Turbulence</td>
<td>K-Epsilon Turbulence</td>
</tr>
<tr>
<td></td>
<td>Realizable K-Epsilon Two-layer</td>
</tr>
</tbody>
</table>

3. Results

3.1 Validation study

3.1.1 Validation case 1

For the validation of the CFD model at different frequencies, the obtained damping coefficients were compared with the experimental results of forced harmonic oscillation tests performed by Vugts [21]. The heave and roll damping coefficients were compared for the validation. Figure 5 and Table 2 show the specifications of the model.

![Cross section of the model for validation case 1](image)

Figure 6 compares the heave damping coefficients obtained from the current CFD simulations and the experimental results of Vugts [21]. The simulations were conducted at the frequency range of \( \omega \sqrt{B/2g} = 0.3 - 1.4 \) with three different heave amplitudes. The
corresponding Reynolds number range for the simulations are $Re_B = 9.4 \times 10^3 - 1.1 \times 10^5$ (based on the peak velocity and the breadth of the ship section).

<table>
<thead>
<tr>
<th>Item</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, $L$</td>
<td>4.19 m</td>
</tr>
<tr>
<td>Breadth, $B$</td>
<td>400 mm</td>
</tr>
<tr>
<td>Depth, $D$</td>
<td>400 mm</td>
</tr>
<tr>
<td>Draft, $d$</td>
<td>200 mm</td>
</tr>
<tr>
<td>Bilge radius, $r$</td>
<td>12.5 mm</td>
</tr>
<tr>
<td>Displacement, $\Delta$</td>
<td>335.2 kg</td>
</tr>
<tr>
<td>$KB$</td>
<td>100 mm</td>
</tr>
<tr>
<td>$BM$</td>
<td>65.1 mm</td>
</tr>
<tr>
<td>$KG$</td>
<td>200 mm</td>
</tr>
<tr>
<td>$GM$</td>
<td>-33.33 mm</td>
</tr>
</tbody>
</table>

As shown in the figure, a good agreement achieved between the current CFD and the experimental data.

On the other hand, deviations were observed for the roll damping coefficients. Figure 7 shows the roll damping coefficients, obtained from the current CFD simulations and the Vugts’ forced harmonic oscillation test. The simulations were conducted at the frequency range of $\omega \sqrt{B/2g} = 0.3 - 1.5$ with three different roll amplitudes. The corresponding Reynolds number range for the simulations are $Re_B = 9.4 \times 10^3 - 1.1 \times 10^5$ (based on the peak peripheral velocity at the bilge and the breadth of the ship section). The roll damping coefficients show a fair agreement for low and moderate frequencies but the deviations increase with the frequency. Similar differences were also observed by other studies [18-21]. These discrepancies may be attributed to experimental inaccuracies at the high frequency of motions, as also questioned by Vugts [21], Bonfiglio [31] and Thilleul et al. [32]. However, further investigations are required in future studies for better understandings.

Fig.6 Heave damping coefficients at different frequencies and amplitudes
3.1.2 Validation case 2

An additional validation study was performed comparing the simulation results with the free roll decay test conducted by Kim et al. [20]. Figure 8 and Table 3 show the specifications of the model.

![Cross section of the model for validation case 2](image)

Figure 9 compares the roll damping coefficients calculated from the current simulations and the scatter diagrams obtained from the free roll decay test performed by Kim et al. [20]. The coefficients in the figure are non-dimensionalised as the ratios to the critical damping coefficients ($B_{44} = B_{44}/B_{cr}$). As can be seen in the figure, a good agreement was achieved between the roll damping coefficients obtained from the current simulations and the experimental results of Kim et al. [20].
Table 3 Principal dimensions of the model for validation case 2

<table>
<thead>
<tr>
<th>Item</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, $L$</td>
<td>1000 mm</td>
</tr>
<tr>
<td>Breadth, $B$</td>
<td>250 mm</td>
</tr>
<tr>
<td>Depth, $D$</td>
<td>160 mm</td>
</tr>
<tr>
<td>Draft, $d$</td>
<td>80 mm</td>
</tr>
<tr>
<td>Bilge radius, $r$</td>
<td>25 mm</td>
</tr>
<tr>
<td>Displacement, $\Delta$</td>
<td>19.73 kg</td>
</tr>
<tr>
<td>$KB$</td>
<td>41.5 mm</td>
</tr>
<tr>
<td>$25BM$</td>
<td>65.1 mm</td>
</tr>
<tr>
<td>$KG$</td>
<td>77 mm</td>
</tr>
<tr>
<td>$GM$</td>
<td>29.6 mm</td>
</tr>
<tr>
<td>Critical damping coefficient, $B_{cr}$</td>
<td>5.73 N·m·s</td>
</tr>
</tbody>
</table>

Fig. 9 Roll damping coefficients obtained in the present study and from a scatter diagram by Kim et al. [20]

3.2 Determination of the linear and nonlinear roll damping coefficients

To examine the effect of frequency on the nonlinear roll damping coefficient, the linear and nonlinear roll damping coefficients were calculated using the proposed method. For the calculations, the damping coefficients obtained for validation case 1 were used. The coefficients obtained at different amplitudes at each frequency were decomposed into linear and nonlinear damping coefficients using Equation 12-14. Figure 10 shows the calculated linear and nonlinear roll damping coefficients using the proposed method. Both the linear and nonlinear coefficients show strong frequency dependencies.
3.3 Linear roll damping coefficient

Figure 11 compares the linear roll damping coefficients calculated by the newly proposed method and the linear potential damping coefficients obtained by close-fit conformal mapping. As shown in the figure, a good agreement was achieved between the linear roll damping coefficients obtained from the different methods.

![Linear and nonlinear roll damping coefficients obtained from the present study](image1)

![Linear roll damping coefficients obtained from the present study and potential theory](image2)
3.4 Nonlinear roll damping coefficient

The nonlinear (quadratic) roll damping coefficients obtained at the different frequencies were compared with a semi-empirical method. Ikeda [17] described the nonlinear roll damping coefficient using four components, namely; frictional damping, eddy-making damping, bilge-keel damping, and lift damping. In this study, only the frictional and eddy-making components were considered as there is no bilge-keel or advance speed in this study. Equation 13 shows the quadratic roll damping coefficient calculated by Ikeda’s method as,

\[ B_{44q}^{(2)} = B_{44F}^{(2)} + B_{44E}^{(2)} \] (13)

Figure 12 compares the quadratic damping coefficients which were obtained from the proposed method and those calculated by Ikeda’s method. It can be seen from the figure that the quadratic damping coefficients obtained from the current study show a strong frequency dependence while those from Ikeda’s method remain relatively consistent. Considering that eddy-making damping has large contributions in the quadratic damping coefficients in Ikeda’s method, as shown in the figure, the difference between two methods can be attributed to eddy-making damping. That is to say, whereas Ikeda’s method does not consider the effect of frequency when calculating the eddy-making damping coefficients, if the eddy-making behavior of the current CFD simulation has frequency dependency, this can bring different results.

For confirmation purposes, the roll damping components of the CFD simulations were divided into frictional (shear) and pressure to investigate the contributions of the different damping components. Figure 13 and 14 compare the contributions of the shear and pressure components in the damping coefficients, at low (\( \omega' = 0.5 \)) and high (\( \omega' = 1.5 \)) frequencies respectively.

As illustrated in the figures, the pressure components were observed to be dominant. They are nonlinear with respect to the roll amplitude. The nonlinearity of the damping coefficients (i.e. changes with the roll amplitude) was observed to be larger at low frequencies (Figure 13) compared to them at high frequencies (Figure 14). As the nonlinearity of the wave making damping is expected to be minor [37], this frequency dependence can be mostly attributed by the eddy-making damping.

As a result, the quadratic roll damping coefficients calculated from the newly proposed method show disagreement with those obtained from Ikeda’s method, which does not consider the effect of frequency on the eddy-making damping.
Fig. 12 Quadratic roll damping coefficients obtained by the present study and Ikeda’s method

Fig. 13 Pressure and shear component of damping moment at $\omega' = 0.5$
Soon-Seok Song, Sang-Hyun Kim, Kwang-Jun Paik

Determination of linear and nonlinear roll damping coefficients of a ship section using CFD

In order to investigate the rationale behind the frequency dependency of eddy-making damping, the vortex field around the body was compared. Figures 15 (a) and (b) compares the vorticity around the body with respect to motion amplitudes of 0.05 and 0.15 rad at a frequency of $\omega = 3.5$ rad/s ($\omega \sqrt{B/2g} = 0.5$), while Figures 15 (c) and (d) show the those at $\omega = 10.5$ rad/s ($\omega \sqrt{B/2g} = 1.5$). As shown in the figure, the vorticity magnitude increase with both the roll amplitude and frequency, but the influence from the roll amplitude (a vs b, c vs d) is much greater than that on the frequency (a vs c, b vs d).

On the other hand, it should be borne in mind that the roll damping coefficient is the ratio between the radiation moment in out phase, $R_{out}$, and the roll rate, $\dot{\phi}_a = \phi_a \omega$. Therefore, comparing the vorticity magnitudes at the same roll rate, $\dot{\phi}_a = \phi_a \omega$ can be a good measure to investigate the frequency dependency on eddy-making damping. That is to say, Figure (b) and (c) have the same roll rate ($\dot{\phi}_a = 0.53 \text{ rad/s}$), and thus the vorticity magnitudes of them can represent the level of eddy-making damping at the frequencies of $\omega = 3.5$ rad/s and $\omega = 10.5$ rad/s, respectively. As can be seen in the figure, the vorticity in Figure (b) is much greater than that in Figure (c). It is thought that because the vorticity increases more with roll amplitude than with frequency, as mentioned earlier, (b) shows larger vorticity magnitude due to its larger amplitude although it has smaller oscillation frequency than (c).

Therefore, it can be concluded that at a constant roll rate, the corresponding roll amplitude decreases when the frequency increases ($\phi_0 = \dot{\phi}_a / \omega$). As a consequence, the vorticity around the body decreases and thus the eddy-making damping decreases, resulting in the decreasing trend of the quadratic roll damping coefficients. Since Ikeda’s method does not consider these influences of frequency, the Ikeda’s quadratic roll damping coefficients show disagreement with those obtained from the newly proposed method.
Determination of linear and nonlinear roll damping coefficients of a ship section using CFD

Soon-Seok Song, Sang-Hyun Kim, Kwang-Jun Paik

4. Conclusion

In this study, a new approach was proposed to determine the linear and nonlinear roll damping coefficients with varying frequencies. A fully nonlinear CFD simulation model for forced harmonic oscillations was developed. For validation purposes, the roll damping coefficients calculated from the proposed method were compared with the experimental data obtained from a forced oscillation test and a free roll decay test. Then, CFD simulations were conducted with varying roll amplitudes and frequencies. At each frequency, the roll damping coefficients were divided into linear and nonlinear components using the linear regression analysis.

The obtained linear roll damping coefficients were compared with potential damping coefficients and showed a good agreement. The nonlinear (quadratic) damping coefficients were compared with those obtained by Ikeda’s semi-empirical method and the two methods showed different trends. While the coefficients calculated using Ikeda’s method remain relatively consistent, those from the proposed method showed a decreasing trend with frequencies. It was observed that the difference between the two methods originates from the frequency dependence of the eddy-making damping. Whereas Ikeda’s method uses a constant eddy-making damping coefficient for different frequencies, the current CFD simulation showed decreases in vorticity with increasing frequency, which leads to decreasing eddy-making damping with frequency. Accordingly, the quadratic roll damping coefficients obtained from the proposed method show a decreasing trend as the frequency increases.

This study proposes a novel approach to determine the linear and nonlinear roll damping coefficients at varying frequencies. However, this study was only conducted for a simplified 2D section, therefore future pieces of work may be extending the proposed approach for a 3D ship, including other modes of motions.
Soon-Seok Song, Sang-Hyun Kim, Kwang-Jun Paik

Determination of linear and nonlinear roll damping coefficients of a ship section using CFD

Acknowledgement

This work was supported by INHA UNIVERSITY Research Grant.

References


Soon-Seok Song, Sang-Hyun Kim, Kwang-Jun Paik

Determination of linear and nonlinear roll damping coefficients of a ship section using CFD


Submitted: 11.09.2019. Soon-Seok Song
Department of Naval Architecture, Ocean & Marine Engineering, Faculty of Engineering, University of Strathclyde, Glasgow, UK

Accepted: 03.10.2019. Sang-Hyun Kim (corresponding author), kimsh@inha.ac.kr
Department of Naval Architecture and Ocean Engineering, Inha University, Incheon 22212, Republic of Korea

Kwang-Jun Paik
Department of Naval Architecture and Ocean Engineering, Inha University, Incheon 22212, Republic of Korea

33
CONTRIBUTION TO THE SEAKEEPING ANALYSIS OF MULTIHULL WARSHIPS

UDC 629.5.017.2:629.5.022.1:629.5.022.3

Summary

This paper analyses in the study of the factors that most affect multihull vessels, and thus determine the feasibility of a new class of ships. It focuses on the impact of service requirements (vessels are typically designed and built under national standards that take into account different factors such as strength, stability, buoyancy, etc.) in a way they are able to withstand extreme environmental conditions with minimal damage, essential for correct and responsible operation. In this paper it is compared, in terms of seakeeping prediction, the results of a monohull with a multihull. In recent years there has been a major breakthrough in this area and it should be noted that although a priori, it may seem that such structures have very good seakeeping characteristics and they provide superior operation compared to traditional monohull with equivalent displacement. They also have some disadvantages such as a higher drag because the wetted surface is higher than its equivalent monohull, and a large weight variation due to their low flotation area. Seakeeping optimization and improving on board comfort aspects are both aspects related to the development of reliable numerical tools as well as knowing a good statistical climatological description. It is crucial to conduct preliminary tests using scale models in a canal with the latest technology for wave generation, test instruments and devices for measuring the movements of the ship and the application of operational criteria. This ensures smaller amplitude movements, high coefficient of floating, sterns mirror, less draft, more beam and metacentric height provide the appropriate natural period.

Key words: Seakeeping; multihull vessels; warships; monohull vessels

1. Introduction

As the latest literature Nobel Prize in literature Bob Dylan’s song “Times are changing”. However, in the latest times, the changes go as fast as the technology changes, and actually, it changes very fast. Naval industry is a very conservative sector where the changes enter very slowly.

However, in order to evolve properly, it must to use technology tools incorporating current improvements of this year that become obsolete just a year later. Naval industry has
long life cycles and needs to face with many technology changes along the entire lifecycle. The naval industry needs to harmonize these two opposite poles: modern technology and the long building lifecycle of the specific products.

A warship, as shown in figure 1, must face different scenarios, more diverse every day, so it seems necessary to conduct detailed studies in multiple areas that could affect its design. New ship designs with non-standard hull forms are promoting large researches and development programs. As it cannot be otherwise, a warship should be designed according to a future scenario, but this scenario, as Bob Dylan could say, “are changing”, and here is where there are uncertainties that a Ministry of Defense (MoD) should anticipate, and prevent, in order to be able to transform the necessity in some appropriate operational requirements.

Fig. 1 Chinese multihull warship Tuo Chiang

It must not be forgotten that a single initial factor may completely affect the construction of the vessel, such as a request on the speed or its seakeeping. The concept of seakeeping must be understood from the very early stages of the ship design, looking for an optimal degree for the crew. “Seasickness is to the space, what impatience is at the time” as it was stated by Arthur Schnitzler during the XIX century. For this reason, it is necessary to define where the warship will operate, what kind of sea prevails, and what kind of operations will be carried out.

The platform of the future warship affects in a substantive way its operation and therefore it is necessary to have an expertise (know-how) that will help the MoD in order to evaluate several hydrodynamic alternatives of the project, in the search for a solution end that contributes not only for saving cost and optimization of the ship conditions, but also to increase the safety, fuel consumption, performance of the propeller, its conditions of navigation and movements and accelerations in the sea. The main characteristic to the advance of multihull ships, in terms of resistance, is more complex when it comes to characterize them due to the interaction between the hulls, consequence of the interference
between the flows surrounding each body, and those produced around a hull as a result of the presence of the other. In addition to the improvement in the seakeeping (behavior associated with the platform in the sea) analysis, there are other developments that allow to get the necessary stabilization during a certain ship operation, predicting and scheming the coming of a quiescent period in advance (Riola and Diaz, 2010) [22] and analyzing the impact on the ship behavior is a minor grade or rank (pitching, rolling and yawing).

Additionally, it must be noted, the comfort level in moderate waves and safety in very big waves are not “the different concepts“, because better seakeeping means higher comfort and safety too.

It is known that the rolling induced by the wave movement, produces lateral and vertical accelerations that could affect the crew (Çakici, 2017) [1], cause interruptions in their tasks or deterioration of the load, with serious implications in its stability. In addition, the rolling affects the maintenance of the course following the optimal route prioritizing fuel consumption (Pérez, 2013) [17].

Also the vertical accelerations (heaving) imply a rising of the seasickness rates and affect the effectiveness of the weapons shipped, while complicate the landing and takeoff of aircraft and unmanned aerial vehicles in frigates and aircraft carriers. Some of the most popular devices whose purpose is to reduce the movement of rolling produced by waves (Moaleji, 2007) [14] are anti-rolling tanks, preferably active ones which incorporate a control that alters the natural period of the tank in order to correspond with the natural period (Pérez, 2005) [21].

It must also be taken into consideration the balance keels welded to the hull and located in the bilge of the vessel, the stabilizing fins, as shown in figure 2, which tend to be more effective the greater is the speed of the ship and you exhibit them with mechanisms that allow them to be hidden inside the hull.

![Fig. 2 Anti-rolling devices](image-url)
As example of an innovative multihull warships, it could be mentioned the effort of the French Navy in its L-CAT (figure 5) as a program of amphibious forces, an idea of the 90s that it was materialized in 2008 with the construction of a first prototype merging the catamaran concept and landing craft, having in its central part of a lifting platform that allows to modify its configuration when boarding or landing, as well as having been designed to integrate on amphibious vessels of the French Navy.

Derived from this concept, its MoD carried out a contract in 2009 for the construction of four units named Engin de Débarquement Amphibie Rapide (EDAR), figure 3, which offer up to 5 times its shipping capacity.

This acronym represents another concept of multihull vessel whose origin dates back to the 19th century (Ohiales, 1880) [16], but which would not be until 1968 when the launch of “Duplus”, a SWATH, takes place in Holland, a 40 meters length and 1,200 tons ship whose mission was to act as auxiliary vessel to the Offshore Platforms (Kos et al, 2009) [13] and in 1973 when the “kimalino” was launched in San Diego and the SWATH acronym was created by the US Navy.

At the end of the XX century the design and construction of this type of vessels was generalized and its greatest design highlight was the Shadow IX-529, built by “Lockheed Missiles and Space Company” which had two twin hulls, each with a propeller.

The shown examples of multi-hulls seem more or less accidental; it seems, the author has not a general view on contemporary application of multi-hull ships. For example, the book of Victor Dubrovsky [4].
2. Designing the hull forms

When these type of ships (multihulls), are being designed, Naval Architects & Engineers focus on improving the main characteristics of these hull forms, increasing L/B-ratios, designing ships thinner and slender rather than the traditional equivalent monohull, in order to take advantage as regards the total resistance of the ship (i.e. Frictional resistance is increased in multi-hulls, while wave resistance is reduced).

These different ways of hull forms definition minimize the wave resistance, which highly depends on the speed (Holtrop and Mennen) [9], see equation 1, allowing multihull vessels to reach higher speeds.

\[
R_{\text{waves}} = k \cdot \nabla \cdot \rho \cdot g \cdot e^{(m_1F_N^d + m_2 \cos(\lambda F_N^{-2}))}
\]  

(1)

Another feature of the side hulls refers to their symmetry / asymmetry, understood as that which is observed from the division of the volume of the hull by the theoretical plane from the ship center line.

There is not a fixed rule on which hull forms are the most appropriate, this is why it is necessary to do an analysis of different alternatives in each project depending on the speed and separation or distance between hulls.

From a hydrodynamic point of view, there are software Computer Aided Design (CAD) tools, see figure 4, capable of incorporating different aspects of almost automatically.

![Fig. 4 Pressure and speed distribution on the hull of a catamaran vessel](image)
As few examples to mention: the FORAN System (Pérez et al., 2015) [18], probably the older shipbuilding CAD tool in the market; the FRIENDSHIP tool (Nowacki et al., 1995) [15] which incorporates a technique of modeling based on generation of parametric curves, whose first approximation was made at the end of the past century (Harries and Abt, 1998) [8]; and the (FRONTIER) [6] or (SHIPFLOW) [27].

When designing, there are today more references as (Grigoropoulos, 2004) [7] which proposed a dual scheme to optimize forms according not only to the resistance factor (see figure 5) but also to the seakeeping.

![Fig. 5 Structural integrity of the L-CAT project](image-url)

More contemporary application of multihull ships focused on resistance and seakeeping behavior can be obtained in (Dubrovsky and Lyakhovitsky, 2001) [3], (Dubrovsky, 2010) [4] and [5], (Katayama, et al, 2011) [11] or (Kim and Yang, 2011) [12].

3. Results and discussion

The seakeeping is a measure of the ability that has a floating structure to be adapted to the sea conditions.

In the famous engraving “The great wave of Kanagawa” by the Japanese artist Katsushika Hokusai, it appears a small fishing boat capable of surviving great waves, but it is far from being a comfortable condition for the fisherman who is on board.

These two different concepts, survival and comfort, are the critical ideas to keep in mind during the design of any floating structure when subjected to a certain type of sea (Perez and Lamas, 2011) [19].

To successfully perform a study of seakeeping, it is mandatory to model the behavior of sea in which it has to navigate the vessel, so the most common sea conditions, as the toughest which will be found to originate the spectrum of waves that will analyze their behavior. Below, in figure 6, it is shown the data that could be obtained through the official website of Puertos del Estado (www.puertos.es), which would present the annual significant height, the
annual peak period histogram, as well as the rest of necessary values to identify the sea condition.

![Seakeeping Charts](image)

**Fig. 6** Representative of the sea condition charts

Seakeeping directly impacts on the design of the ship. Both, main dimensions and general arrangement of the spaces are taken into account in order to consider the movements of the vessel. Thus, it fits within the design of the vessel or design stage as it is shown on figure 7. This process describes the method as a sequence of specific design disciplines, both of synthesis (geometry of the hulls, layout, etc.) and analytical (stability, seakeeping, etc.) in order to achieve an optimal design that meets the requirements, although its complexity is due to some of these disciplines incorporate hundreds of activities within it and the seakeeping is a good example (Pérez and Lamas, 2011) [19].
In fact, the seakeeping characteristics of the vessel, as the one in figure 8, depend on so many interrelated factors that it is virtually impossible to state what will happen if a specific hull forms change is made without a reasonable analysis. This is because the seakeeping depends not only of the response in movements, but also the marine environment in which it is located. The ship in the sea, submitted to the acting forces, moves according to six degrees of freedom.
All kinds of ship movement may be divided into three types of linear motion and three types of rotational motion.

As regards linear ones: motion along the longitudinal axis (surging), motion along the transverse axis (swaying) and motion along the vertical axis (heaving).

Concerning rotational motion: motion around the vertical axis (yawing), motion around the transverse axis (pitching) and motion around the longitudinal axis (rolling).

It can in general be stated that the outwardly directed centrifugal accelerations brought about by any rotational motion are not significant. This accordingly applies to yawing, pitching and rolling.

Some of the typical calculations obtained and discussed with the aim of knowing how it would perform the ship in sea conditions previously described are shown in the figure 9. The idea is to calculate the values of $m_{0θ}$, $m_{0ϕ}$, $m_{0z}$, i.e. the moments of order zero of such curves $S(w)$ or spectral density of response for each movement, representing the area beneath these curves and identify the greatest value for different directions and design speed.

These values give an idea of the energy which has the ship in the three main movements affecting the vessel determining the critical paths for each movement. This is made using polar diagrams, proposed by Comstock et al. [2], which represent a specific value depending on the course and speed. Below, figure 9, are these diagrams for each movement and sea condition. It is important to reflect that these polar diagrams are not used only for naval but also for commercial ships and small craft using different seakeeping criteria.

![Fig. 9 Movements by the action of the waves](image)

Thus, the movements are defined by the six degrees of freedom that this ship can experience. From the polar diagrams for this ship, it could be obtained the following
conclusions: in the case of rolling in normal conditions, maximum is obtained with seas with crosswise direction, reaching rolling values of 12.75°, which are not worrisome.

When it is concluded that there is a maximum sea condition, it will be also considered non-parallel sea direction, reaching rolling of 26.41°, so would be recommend to change the course for improving the rolling.

As regards pitching, under normal sea conditions, the maximum value would be on following seas about 20 knots, reaching values of 3.32°, so would be recommend to modify the speed to reduce added resistance due to waves, while for rough seas, the maximum would be also obtained with following seas at very low speeds, although for head on seas at high speed also presents a maximum, reaching pitching values of 9.32°, so would be recommended to slow down and change the course.

The same conclusions can be obtained for the movements of yawing, where in normal conditions the maximum occurs in directions of 120°, with value of 0.386 m, being low enough as to not be taken into account and with rough sea the maximum would be for head on seas, with a value of 2.245 m.

Seven is the number of factors that mainly affect the responses of the ship. Its size, its dimensions, the hull forms, the weight distribution, displacement, stability and freeboard. There are also several ways of estimating the seakeeping of a vessel and the results of some of these calculations or model tests are transfer functions called Response Amplitude Operators (RAOs).

The use of numerical simulations to predict the response of a ship is very useful, since it provides a cheap way for determining a lot of alternatives from an early stage of the design. Once the design has converged to one or two alternatives, these can then be tested in a towing tank with a greater degree of reliability. To calculate the ship response navigating in waves is a non-linear phenomenon, involving the dynamics of the ship and the hydrodynamic forces.

Although different nonlinear analysis have been presented by different authors, for many applications the order of the nonlinear is sufficiently small so a linear theory would provide adequately accurate results. Thus, numerical calculation methods for predicting the RAOs, can be divided into two large groups: in the time domain and in the domain of the frequency (Pérez, 2012) [20].

The frequency domain methods are simpler and less intensive in the calculations. Most of the Strip Theory use these methods dealing with the ship movements as low-amplitude sinusoidal movements. The two major limitations that present are that vessels must be sufficiently slender and the Froude number not too high. The hydrodynamic characteristics of its sections are calculated, assuming 2D in viscous flow, without interference by upper sections. From these values, the coefficients of the motion equations and its response to the waves are calculated. Thus, the Strip Theory is an excellent tool for the preliminary design or where the scale of the project does not allow a detailed analysis of the seakeeping.

Methods which use the time domain, could define the wave which passes around the hull forms. In small incremental time steps, instant net force integrates the water pressure and the friction forces in each part of the hull. The acceleration in the hull integrates time to calculate the new speed and position of the vessel.

The numerical tools for the prediction of seakeeping have not yet reached a stage of sufficient maturity and numerical techniques must be related to the test results in towing tanks (Pérez and Lamas, 2011) [19].

The Strip Theory could provide important information regarding the seakeeping on trimarans and be used in numerous situations. Thus, to calculate the movements in remote locations, trimaran assume rotation about the center of gravity and therefore the distance to
the remote location from that center of gravity is quite important. In the study has been calculated this distance internally, and all the positions have been measured in a coordinate system.

*Strip theory* is a robust tool for seakeeping calculations. *3-D Rankine source method* works also quite well, especially at higher speeds (Sclavounos, 1996) [26].

The main advantage of the trimaran studied for this paper is to have an smaller floating area rather than its equivalent monohull, as well as a smaller longitudinal inertia moment and a longitudinal metacentric height (Riola and Perez 2012) [23] (Tang, 2019) [28].

The reduced buoyancy is a necessary but not sufficient condition to ensure a good seakeeping. And because the two lateral hulls it is possible to reduce vertical oscillation, and increase the righting arm at high speeds.

In each relative position of the trimaran as solid-rigid, receives the pressures and impacts, plus other local effects, the acceleration forces and the speed affecting each point as the same moving parts rigid.

In this technical paper, the spectrum used has been the Bretschneider one or the *International Towing Tank Conference (ITTC)*. It is useful to define the idealized wave spectra, equation 2, which widely represent the characteristics of the real energy spectra of the wave.

\[
S_{ITTC}(\omega) = \frac{A}{\omega^3} \cdot \exp\left(\frac{-B}{\omega^4}\right)
\]  

Where the coefficients \( A \) and \( B \) are defined as it follows:

\[
A = 172.75 \cdot \frac{H_{char}^2}{T^4}
\]

\[
B = \frac{691}{T^4}
\]

Both parameters depend on the significant wave height, \( H_{char} \), and the average period \( T \). Information about the characteristics of the wave, studied in the Mediterranean Sea, is given in the reference (Riola and Perez, 2012) [23]. The different calculations of spectral moments are given by the following equations, where the significant wave height is described as:

\[
H_{char} = 4 \cdot \sqrt{m_0}
\]

Where \( m_0 \) it is defined as:

\[
m_0 = \frac{A}{4-B}
\]

The average period could be expressed as it follows:
The zero-crossing period parameter is defined as the mean time interval between upward or downward zero crossings on a wave record.

The influence of wave height and period statistics on design will be illustrated with wave run-up at sea dikes, or more general, sloped structures. For a certain range of wave steepness and slope angle $\alpha$ the wave run-up $R$ of regular waves can be characterized by the formula which is based on (Hunt, 1959) [10]:

$$R = 1.27 \cdot H^{0.5} \cdot T \cdot \tan \alpha$$  \hspace{1cm} (8)

For irregular waves usually the parameter $R_{u2\%}$ can be written in the following form:

$$R_{u2\%} = a \cdot H_{\text{char}}^{0.5} \cdot T_{\text{char}} \cdot \tan \alpha$$  \hspace{1cm} (9)

$H_{\text{char}}$ and $T_{\text{char}}$ are characteristic wave parameters from time-series or spectral analysis. $H_{\text{char}}$ is generally accepted to be the significant height $H_{1/3}$.

Thus, the Bretschneider contains all frequencies of the wave. This is the reason why the average period between peaks is zero, because there will be infinitesimally small ripple with adjacent peaks (equations 10 and 11):

$$H_{1/3} \approx 4 \cdot \sqrt{m_0}$$  \hspace{1cm} (10)

$$H_{\text{char}} \approx H_{1/3}$$  \hspace{1cm} (11)

The modal period can be found between the wave spectrum energy and finding the maximum (peak = 0) (equation 12):

$$\omega_0 = \sqrt{\frac{4-B}{5}} = \frac{4.849}{T}$$  \hspace{1cm} (12)

$$T_0 = 1.296 \cdot \bar{T} = 1.41 \cdot T_Z$$  \hspace{1cm} (13)

The added calculated resistance is due only to the movement of the trimaran in waves. For this study, the method applied is the Salvesen method [25] which is based on calculating the longitudinal strength of second-order wave affecting trimarans.

This method is valid in the most problematics navigations and can be applied with a reasonable accuracy up to approximately 20°. The square root media spectrum, $m_0$, is the area under the spectrum and gives a measure of the total response of the trimaran. It is
possible to get different results from the analysis of the trimaran based on each frequency. RMS is the *Root Mean Square*, and for this study the significant amplitude is twice the value of the RMS.

![Amplitude of the RAO of pitching for 25 knots](image)

**Fig. 10** Amplitude of the RAO of pitching for 25 knots

Spectral conditions of sea representations are key to determine the response of trimarans in the sea. The RAOs (figure 10) tend to one in low frequency, where the trimaran moves simply up and down with the tide and behaves like a cork. In high-frequency the response tends to zero, because the effect of many short waves, is contrasted with the length of the trimaran. It will have a typical peak higher than one because is close to the natural period of the trimarans, due to the resonance.

A greater RAO value than one, indicates that the response of the trimarans is greater than the amplitude of the wave (Riola et al., 2013) [24]. Due to its complexity, analysis of seakeeping usually ends in a late stage of the ship design process using computer applications.

Although these calculations can be performed successfully in monohull vessels, at the moment these calculations are not optimal in the design of innovative hull forms of multihull vessels. The recent interest in catamarans and trimarans, raises unknowns to the naval sector, while there are not accurate application tools for calculating their seakeeping.
4. Naval seakeeping applications

From naval architect experiences with naval warships, we are aware of a large number of seakeeping issues in the ship design for operational purposes, as for example, relate to crew performance, as affected by seasickness, reduced mobility or fatigue. These are governed by the vertical and transverse accelerations and to some extent by slamming induced vibrations. The issues related to vertical motions, figure 11, and accelerations and relative wave elevation in the fore ship are typically most critical in head and bow quartering seas. Rolling related issues are critical at the heading where resonant roll conditions occur. Course keeping and broaching are critical at moderate speed in steep waves yielding low encounter frequencies, i.e. in stern waves.

Other examples as damage on the fore deck, determined by the shipping of green water and its dispersion on the fore deck, spray development in the bow flare, helicopter operations as vertical replenishment, limited by the vertical velocities and accelerations and the transverse accelerations at the helideck, rolling and the performance of fin stabilizers or course keeping and related risk of broaching in waves from the stern quarter, must be controlled, validated and supervised. And all this great amount of results, data and developments are those that will give us the operational optimization of the ship.

**Fig. 11** Limits of vertical replenishment (VERTREP)
5. Conclusions

This highlights that there are too much work remaining ahead in the prediction of the seakeeping, as to compare the results of a monohull with a multihull vessel, although there has been in recent years an important advance in this area as this paper carried out for a trimaran in which we have obtained results that allow a rigorous comparison with traditional monohull platforms.

It has been proven that Computer Fluid Dynamic (CFD) applications normally used for conventional ships, can be used in this type of multihull vessel, with, in general, satisfactory approximations to the experimental results, which is a great time and money saver facing optimizations of hull forms for these types of vessels. It should also be taken into account to analyze the pressure distribution calculated under the hull when the optimization of hull forms. Seakeeping results show that the trimaran hull forms is satisfactory, with similar performance rather than a monohull with same displacement. A detailed study of the appendices should certainly introduce substantial improvements.

The response of a vessel to a certain sea condition depends on its weight, the righting arm and the characteristics of the waves. When the righting arm is smaller, in the cases of pitching and rolling, greater is the period of motion.

When studying the seakeeping of a vessel, there are four natural factors to keep in mind: topography, tides, currents and winds. Of these factors, there are two with a far superior importance: wind and waves.

Seakeeping optimization and other aspects of improvement of the comfort on board, need reliable development of numerical tools, as well as knowing an appropriate weather statistical description of the operation zone.

It is compulsory to manage test scale models in a towing tank with generation of waves and an operating criteria, aimed to ensure movements of lower amplitude, high flotation coefficient, less draft, more beam and metacentric height which provide the suitable natural period. Nowadays computers have facilitated this problem but, for the moment, they cannot replace tests in towing tanks.

Finally point out that the incorporation of multihull vessels in the shipbuilding industry has pass borders, and this type of ships is becoming a current trend of construction for different navies as opposed to monohull structures, always starting from the premise of the needs and typologies in which these units are applicable.

Acknowledgments

We offer our regards and blessings to all of those who supported us in any respect during the completion of the technical paper.

REFERENCES

FRONTIER. Multi-Objective design environment (mode) FRONTIER. User Manual. 2002. ES.TE.CO s.r.l.


THE TURBULENT BOUNDARY LAYER AND FRICTIONAL DRAG CHARACTERISTICS OF NEW GENERATION MARINE FOULING CONTROL COATINGS

UDC 532.526:532.517.4:62-408.8

Original scientific paper

Summary
The turbulent boundary layer over rough surfaces has been a widely studied research topic since most of the engineering wall-bounded turbulent flows develop under the influence of surface roughness. Accordingly, the research on rough wall turbulent boundary layer has gone a long way. However, unresolved major problems can still be found in this area such as the unsatisfying correlation of roughness and frictional drag for irregular engineering surfaces and the discrepancies about the validity of wall similarity. This study aims to contribute to further understanding of the rough-wall turbulent boundary layer flows developed over marine fouling control coatings along with the investigation of their friction drag properties. Two-dimensional Laser Doppler Velocimetry (LDV) experiments were conducted consisting of zero-pressure-gradient turbulent boundary layer measurements over surfaces coated with marine fouling control coatings together with smooth and rough references in the Emerson Cavitation Tunnel of Newcastle University by using flat plate test models. Six different surfaces were included in the experimental campaign, which consist of one hydraulically smooth reference, a sand grit surface and four surfaces coated with fouling control coatings including Self-Polishing Co-polymer (SPC) and Foul(ing) Release (FR) types, applied either by spraying or rollering. The mean velocity, local skin friction drag and roughness functions were calculated and discussed for the tested surfaces. In complementing the boundary layer tests, roughness measurements of the test surfaces were carried out by using a laser profilometry. Two new relations were proposed for the correlation of the roughness properties and roughness functions within the covered Reynolds number range. However; further work is needed in order to ensure the validity of the proposed relations at higher Reynolds number ranges.

Keywords: Marine fouling control coatings; turbulent boundary layer; roughness-friction drag correlation.
1. Introduction

The surfaces of marine structures or vehicles that are subject to seawater are exposed to fouling coverage in time. Such unwanted biological accumulation on the surface increase the surface roughness and thus the frictional resistance of the surface by altering the turbulent boundary layer flow. It is known that, with the existence of hard-shell heavy fouling in the wetted surface, the ship resistance may increase by up to 86%, which leads to remarkable added fuel consumption [1, 2]. More than 100% increase was observed in the frictional resistance coefficient of a container ship for fouled condition in the recent study of [3]. The marine antifouling paints are used in order to prevent such fouling of the wetted surfaces of marine vehicles. The marine fouling control coatings with copper and co-biocides are under environmental scrutiny and totally environment friendly non-toxic coatings are favoured. As a consequence, the Foul-Release (FR) fouling control coatings, which are the most competitive alternatives to the biocidal ones, gradually supersede the Self-Polishing-Copolymers (SPC). On the other hand, in the literature, some antifouling marine coatings were reported to even lower the frictional resistance characteristics [4, 5]. Moreover, the energy efficiency regulations of IMO for ships entered into force beginning from 2013, which include performance-based standards for ships in order to reduce the greenhouse gas emissions. Therefore, the frictional drag characteristics of the fouling control coatings in the newly applied and clean conditions gained even more importance along with their antifouling properties. Accordingly, there is a continually growing commercial interest and hence support for research and development activities for new coating systems with particular interest to their hydrodynamic characteristics.

The surfaces coated with marine fouling control coatings generally have small average amplitudes of surface roughness – although this highly depends on the application type and procedure- and turbulent rough wall boundary layers are supposed to form in the transitionally rough flow regime over them. Such surfaces are also good examples that show irregular and complex roughness structure. The research on rough wall turbulent boundary layer has gone a long way since the first surface roughness effect studies. However, there still exists unresolved major problems such as the unsatisfying correlation of roughness and frictional drag for irregular engineering surfaces such as marine fouling control coatings. Discrepancies about the validity of wall similarity are also observed. Besides, there is a lack of data on turbulent boundary layer flow over irregularly rough real engineering surfaces and the research in the literature accumulate on geometrically defined regular and mostly two-dimensional roughness types. Examples to regular rough wall turbulent boundary layer research include experimental studies for surfaces covered with spheres, bars, pyramids, meshes, sand particles and sand paper as roughness [6, 7, 8, 9, 10, 11, 12, 13, 14]. Although the studies about irregular rough wall boundary layer flow is rather limited, it continues to gain interest between some researchers such as [4, 5, 15, 16, 17, 18, 19, 20, 21, 22]. It is noteworthy that most of the mentioned studies are about surfaces coated with marine fouling control coatings or fouling.

There is also a growing interest for simulation of rough wall boundary layers growing over marine fouling control coatings or fouling by using Computational Fluid Mechanics (CFD). [23] simulates the flat plate experiments of [17] for antifouling coated surfaces by adopting the roughness scale given in the experimental work to the wall functions in order to calculate the
encountered frictional resistance difference by using CFD. Whilst, in [24] and [25] the effect of marine coatings, biofilm and barnacle fouling on frictional and wave resistance of a full-scale ship is predicted by using (Reynolds Averaged Navier Stokes (RANS) modelling. [26] presents CFD simulations in order to predict the effect of biofilm on the ship resistance by using the effective roughness length scale of [27] for biofilm along with developing a relevant roughness function to implement within wall functions.

On the other hand, looking for a correlation between the roughness parameters and the frictional drag is comparatively easier for regular rough surfaces, since the selection of the characteristic roughness parameter is obvious and the total number of roughness parameters are far limited for such geometrically defined roughness. The number of possible effective roughness parameters get larger and larger as the surface gets more complex. [28] includes the suggested frictional drag-roughness correlations in the literature which mostly accumulates on geometrically defined regular rough surfaces. [17] show an agreement with the Colebrook-White law [29] for surfaces coated with marine fouling control coatings which were being used at that time with average roughness height as the effective roughness parameter. [30] and [4] used a similar parameter as well as a combination of average roughness height and mean absolute slope for such a correlation. However, the proposed combination did not provide a satisfactory fit, as the behavior of the roughness functions they found was quite different from the Colebrook-White correlation. A similar situation can be found in [5], in which nanostructured new fouling control coatings were tested as rough walls in turbulent boundary layer experiments. It was observed that both commercially available fouling control coatings and nanostructured ones exhibited much lower roughness functions than the Colebrook-White law in the tested Reynolds number range.

The lack of studies in the literature, on turbulent boundary layer flows over transitionally rough engineering surfaces underlies the need of new research for such flows. Accordingly, in this study, it is aimed to investigate the frictional resistance properties of turbulent boundary layers growing over surfaces coated with new generation marine fouling control coatings. Within this aim, flat plate turbulent boundary layer experiments were carried out under the influence of zero pressure gradient. Velocity measurements were performed with a two-dimensional DANTEC Laser Doppler Velocimetry (LDV) system. All experiments were conducted at the Emerson Cavitation Tunnel in the School of Marine Science and Technology, Newcastle University. A total of six different plates were tested, including surfaces coated with different types of marine fouling control coatings and two reference surfaces which are hydraulically smooth and completely rough. An SPC, a hard coating (HC), and a FR type antifouling were used among the tested marine fouling control coatings. The FR type antifouling was coated using both spray (FR) and roller (FRR); so that it was also possible to see the difference effects of the application methods. In addition to boundary layer tests, roughness measurements were made on all tested surfaces by using a laser profilometry and several roughness parameters were calculated and presented. As a result of the boundary layer experiments, mean velocity profiles, frictional resistances, roughness functions and correlations between roughness and frictional resistance were investigated.
None of the tested marine fouling control coatings showed compliance with Colebrook-White law with their roughness functions creating a completely distinct trend in the range of $10 \leq k_s^+ < 100$ with much lower than expected roughness function values. Accordingly, two new correlations between roughness and frictional resistance have been proposed to represent the behavior of the new generation marine fouling control coated surfaces by using a complex roughness length scale [28].

2. Experiments

2.1 Test Set-Up and Data Collection

A rigid flat test plate of 3.924 m in length was used for the tests which allowed 600 mm long and 220 mm wide flat plate test specimens to be fitted on it. The test plate had a nose section with an aerofoil shaped leading edge that was deployed in order to slowly lead the incoming flow from the contraction part of the tunnel so that flow behind the test plate was eliminated. 36 grade sandpaper of 400 mm length was used as a turbulence stimulator following the leading edge of the plate. The boundary layer measurements were performed at the midline of the test plates and at 500 mm distances from their leading edges. Accordingly, a distance of 2724 mm was achieved for the boundary layer growth. The details of the test plate described above can be seen in Fig. 1. Steel plates with a length of 600 mm and a height of 220 mm were coated with the selected marine fouling control coatings for the boundary layer testing. The hydraulically smooth surface (ACRYLIC) was specially cut out from an acrylic plate. The rough reference surface (SAND40), which is expected to be in the fully-rough flow regime, was obtained by coating a steel plate with 40 grade sandpaper. The summary of the tested surfaces is given in Table 1 along with their abbreviations. The test plates were mounted on the large flat plate as shown in Fig. 1, respectively, and the tests were carried out at a distance of 500 mm (POS1) from the beginning of the test plate. The experiments were carried out at three different free flow rates of 2, 4, and 6 m/s. Thus, the local Reynolds number at the test location varied between $5.45 \times 10^6$ and $1.63 \times 10^7$. The streamwise turbulence intensity of the incoming flow varied between 1.4% to 2% whilst the transverse turbulence intensity stayed around 2% for the covered flow rates.

In the boundary layer, velocity-time data, were collected by a DANTEC LDV system in two dimensions. A 60 mm LDV probe was used with a 5° slope to collect data near the wall. In addition, the probe was rotated 45° so that the velocity component in the wall-normal direction, which is much lower than the one in the direction of free flow, could be better identified. A 500 mm focal length lens was used along with a 1.98 beam expander to reduce the probe volume. Spherical glass spheres with a diameter of 2 μm were used as seeding to reflect the laser beam. Boundary layer velocity data were collected at 80 points in the boundary layer for 100 to 120 seconds at each point, so that at least 10000 data points were collected near the wall where the data rate was low, and about 50,000 data were collected as the distance from the wall rapidly increased. The data rate ranged from 100 to 1200 Hz.
The turbulent boundary layer and frictional drag characteristics of new generation marine fouling control coatings

Burcu Erbaş

Fig. 1 Details of the test plate

Table 1 Tested surfaces as a summary

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Surface</th>
</tr>
</thead>
<tbody>
<tr>
<td>FR</td>
<td>Foul(ing) Release (FR) coating spray-applied</td>
</tr>
<tr>
<td>HC</td>
<td>Hard coating spray-applied</td>
</tr>
<tr>
<td>SPC</td>
<td>Self-Polishing Co-polymer (SPC) coating spray-applied</td>
</tr>
<tr>
<td>FRR</td>
<td>Foul(ing) Release (FR) coating applied by rollering</td>
</tr>
<tr>
<td>SAND40</td>
<td>40 grade sand paper</td>
</tr>
<tr>
<td>ACRYLIC</td>
<td>Acrylic as smooth reference</td>
</tr>
</tbody>
</table>

2.2 The Surface Roughness Characteristics of the Tested Surfaces

Roughness measurements were run with a laser profilometry for each test plate at intervals of 25 μm on areas of 60 mm by 90 mm. Obtained roughness profiles were analysed at different cut-off lengths using a Matlab program. Table 2 presents the results obtained for a 50 mm cut-off length.

Table 2 Surface roughness parameters, 50 mm cut-off length

<table>
<thead>
<tr>
<th>Surface</th>
<th>Rt (μm)</th>
<th>Ra (μm)</th>
<th>Rq (μm)</th>
<th>Sk</th>
<th>Ku (μm)</th>
<th>Sm (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>FR</td>
<td>46.7</td>
<td>8.0</td>
<td>9.6</td>
<td>-0.1</td>
<td>2.4</td>
<td>1167.7</td>
</tr>
<tr>
<td>HC</td>
<td>56.4</td>
<td>10.4</td>
<td>12.9</td>
<td>0.3</td>
<td>2.7</td>
<td>1570.2</td>
</tr>
<tr>
<td>SPC</td>
<td>45.3</td>
<td>7.6</td>
<td>9.4</td>
<td>-0.2</td>
<td>2.9</td>
<td>816.5</td>
</tr>
<tr>
<td>FRR</td>
<td>159.7</td>
<td>23.2</td>
<td>31.2</td>
<td>1.0</td>
<td>4.5</td>
<td>2367.6</td>
</tr>
<tr>
<td>SAND40</td>
<td>804.3</td>
<td>99.1</td>
<td>125.8</td>
<td>0.6</td>
<td>3.5</td>
<td>1248.6</td>
</tr>
</tbody>
</table>
Rt, from the roughness parameters presented in Table 2, shows the maximum difference between the roughness hump and trough point in a cut-off length. Ra is the average roughness height, and $R_q$ is the root-mean-square (RMS) of the roughness profile. Sk and Ku express the skewness and kurtosis of roughness height probability density functions, respectively. On the other hand, Sm indicates the distance between two peaks in the roughness profile. The values given in the table are the average values obtained for a total of 102 the roughness profiles. According to the results, the roughness parameters related to the mean height of the FR and SPC test plates are of the same order. On the other hand, application by rolling greatly increased the roughness heights of the surface. It has been found that the HC test plate has the highest roughness between the surfaces subjected to application by spraying. On the other hand, the sand paper surface, SAND40, has rather higher roughness amplitudes than the others. FRR and SAND40 test plates were found to have higher skewness than normal distribution; which indicates that the roughness distributions of the respective surfaces are sharper peaked and longer tailed. If so, it can be assumed that the roughness heights are usually concentrated at certain values, and very high and very small values are relatively rare. It is also possible to say that spray-coated surfaces, which show smaller Ku values than 3, exhibit flatter peaked and thick tailed distributions.

2.3 Data Analysis Methods and Uncertainty Estimates

Analysis of the velocity-time data collected in the boundary layer were carried out with the a Matlab program and a MS Excel program. The raw data were first filtered in the Matlab for noise elimination using the Chauvenet criterion [31]. Then the velocity components in the flow direction and the wall-normal direction are calculated by coordinate transformation. Subsequently, the moments of the velocities were taken with the transit time averaging technique. In order to determine the friction velocity, velocity profile fitting method and total stress method were applied as in [32] and [33], respectively. There is a maximum of 2.3% difference between the results of the two methods, which is consistent with the literature [34]. In this article, only the total stress method results will be presented.

Two methods were used for the uncertainty analysis. The method of [35] was used to account for the statistical uncertainty associated with the random sampling and limited sample population. On the other hand, repetitive tests were performed for the calculation of uncertainty according to [31] for the calculation of uncertainty due to possible inconsistencies in the experimental setup. 95% confidence bounds were used in both uncertainty analysis procedures.

The uncertainty levels in the streamwise and transverse velocities are 1.18% and 7.82% between $y/\delta = 0.05$ and $y/\delta = 0.1$ in average. The mean uncertainty in U and V for $y/\delta > 0.1$ can be given as 1.12% and 2.48%, respectively. The average uncertainty percentages for $\overline{uu}$, $\overline{vv}$ and $\overline{uv}$ are 1.30%, 1.62% and 2.93% respectively between 0.02 < $y/\delta$ < 0.15. The average uncertainty in $\overline{uu}$, $\overline{vv}$ and $\overline{uv}$ are 2.01%, 1.97% and 3.72% respectively for 0.15 < $y/\delta$. The average uncertainty for the friction velocity and local skin friction coefficient values which were calculated with the total stress method were determined as 1.37% and 3.29% whilst the maximum uncertainty in the calculated roughness functions was 7.58% for the tested surfaces.
3. Experimental Results

For all the tested surfaces, the average velocity profiles obtained at 6 m/s testing freestream velocity are presented in Fig. 2 comparatively in non-dimensional form with the inner flow scales. The viscous velocity profile and the logarithmic rule for smooth walls [36] are also present in the figure for comparison. The total stress method was used to determine the frictional velocities, which were then used for obtaining non-dimensional velocity profiles. It is seen that the average velocity profiles formed over the ACRYLIC plate, which is the smooth reference surface, are highly compatible with the logarithmic rule. When examined by the given scaling, the velocity profiles are expected to move downward and slightly to the right with the effect of roughness. Although not very obvious, it can be said that the FR and HC surfaces show a slight downward shift. On the other hand, the FRR surface and especially the fully rough reference surface SAND40 differ in severity from the smooth wall velocity profile. As expected, the velocity profiles of the surfaces show differences from the smooth wall logarithmic law depending upon the surface property, and a greater difference indicates a higher frictional resistance and roughness function expectation.

In Fig. 3, the velocity defect profiles showing the deviation of the mean velocity profiles from the free flow velocity are presented by Rotta scale, which is a mixture of inner and outer scales, for all test cases. The given graphic is an important indicator of the outer layer and wall similarity of [37]. As can be seen, all velocity defect profiles coincide exactly in the logarithmic and outer layers of the boundary layer. Accordingly, it can be said that the roughness effect remains specific to the inner layers of the boundary layer and that no significant difference is expected with the roughness effect on the wake strength parameter. This observation is consistent with, for example, [12]. In addition, while the friction velocity was calculated by the velocity profile fitting method, it was observed that the determined wake strength parameters change from 0.31 to 0.21 for all surfaces and test velocities. The different test surfaces were not necessarily effective on this change. These values are relatively low compared to the 0.52 value normally expected on the smooth wall [38]. This behavior can be attributed to the relatively high freestream turbulence intensity [33].
The calculated local frictional resistance coefficients (cf) and friction velocities ($u_\tau$) are presented in Table 3 together with the Reynolds number ($Re_{\delta_1}$) depending on the displacement thickness for all test cases. As expected, as the test velocity increases, the friction velocity increases and the local frictional resistance coefficient decreases. Smooth reference ACRYLIC exhibits the lowest friction values, while the fully-rough reference SAND40 has a very high frictional resistance compared to all other surfaces. On the other hand, it was determined that the surface coated with the spray-applied foul release paint (FR) has a lower frictional resistance of 4.0 to 4.5% than that of the other coatings applied with spray. On the contrary, when the same coating was applied with rollering, 5.2% higher average frictional resistance values were observed.

Table 4 also lists the roughness functions ($\Delta U^+$) and roughness Reynolds numbers ($k_s^+$) calculated with the total stress method, together with the Reynolds numbers ($Re_\theta$) depending on the momentum thickness. The values of $k_s^+$ presented were calculated by the formula given in Flack and Schultz (2010). According to this:

$$k_s = 4.43Rq(1 + Rsk)^{1.37}$$  \hspace{1cm} (1)

The [28] formula was found suitable for use in this work, as it is based on surface roughness data obtained with 25 μm short and 50 mm long cut-off lengths. As shown in Table 3, the roughness functions are in parallel with the frictional resistance. Very low roughness functions have been achieved for all painted surfaces. On the other hand, the SAND40 surface, which is in a fully-rough regime, exhibited very high roughness functions compared to other tested surfaces. All surfaces showed a trend with increasing roughness function values as the Reynolds number of increases.
The turbulent boundary layer and frictional drag characteristics of new generation marine fouling control coatings

**Table 3** Friction velocities and local frictional resistance coefficients

<table>
<thead>
<tr>
<th>Surface</th>
<th>$Re_{\delta_1}$</th>
<th>$u_t$</th>
<th>$c_f \times 10^3$</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACRYLIC_POS1_2</td>
<td>10049</td>
<td>0.0771</td>
<td>2.86</td>
</tr>
<tr>
<td>ACRYLIC_POS1_4</td>
<td>17600</td>
<td>0.1462</td>
<td>2.60</td>
</tr>
<tr>
<td>ACRYLIC_POS1_6</td>
<td>25950</td>
<td>0.2120</td>
<td>2.43</td>
</tr>
<tr>
<td>HC_POS1_2</td>
<td>8893</td>
<td>0.0782</td>
<td>2.99</td>
</tr>
<tr>
<td>HC_POS1_4</td>
<td>16407</td>
<td>0.1510</td>
<td>2.78</td>
</tr>
<tr>
<td>HC_POS1_6</td>
<td>23565</td>
<td>0.2220</td>
<td>2.66</td>
</tr>
<tr>
<td>FR_POS1_2</td>
<td>9701</td>
<td>0.0780</td>
<td>2.93</td>
</tr>
<tr>
<td>FR_POS1_4</td>
<td>16569</td>
<td>0.1485</td>
<td>2.69</td>
</tr>
<tr>
<td>FR_POS1_6</td>
<td>25620</td>
<td>0.2180</td>
<td>2.55</td>
</tr>
<tr>
<td>FRR_POS1_2</td>
<td>10851</td>
<td>0.0805</td>
<td>3.14</td>
</tr>
<tr>
<td>FRR_POS1_4</td>
<td>18739</td>
<td>0.1574</td>
<td>2.99</td>
</tr>
<tr>
<td>FRR_POS1_6</td>
<td>27639</td>
<td>0.2289</td>
<td>2.85</td>
</tr>
<tr>
<td>SPC_POS1_2</td>
<td>9542</td>
<td>0.0778</td>
<td>2.93</td>
</tr>
<tr>
<td>SPC_POS1_4</td>
<td>17269</td>
<td>0.1520</td>
<td>2.78</td>
</tr>
<tr>
<td>SPC_POS1_6</td>
<td>24589</td>
<td>0.2210</td>
<td>2.65</td>
</tr>
<tr>
<td>SAND40_POS1_2</td>
<td>13499</td>
<td>0.1124</td>
<td>5.97</td>
</tr>
<tr>
<td>SAND40_POS1_4</td>
<td>24775</td>
<td>0.2258</td>
<td>6.17</td>
</tr>
<tr>
<td>SAND40_POS1_6</td>
<td>36541</td>
<td>0.3352</td>
<td>6.07</td>
</tr>
</tbody>
</table>

The Reynolds number dependence of the roughness functions on the surfaces and the compliance with the Colebrook-White law can be examined in Fig. 4. The graph also includes correlations given by [39] and [40] for comparison. In addition, data from the transition regime of [34] have been added to the graph. Mentioned comparative data from [34] were obtained at different free flow velocities for a single regular rough surface and exhibit positive roughness function values starting from $k_s = 2.5$ and are more closely related to the correlation given by Ligrani and Moffat (1986). The [34] surface roughness data was also obtained with 25 μm short and 50 mm long cut-off lengths. One should note that this data is added to the graph as an example of regularly rough surface which does not fit to Colebrook-White law and should not be compared with the roughness function behavior of fouling control coated surfaces of this study.
**Table 4** Roughness functions and roughness Reynolds numbers

<table>
<thead>
<tr>
<th>Surface</th>
<th>$Re_\theta$</th>
<th>$\Delta U^+$</th>
<th>$k_\theta^+$</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACRYLIC_POS1_2</td>
<td>7826</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>ACRYLIC_POS1_4</td>
<td>14173</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>ACRYLIC_POS1_6</td>
<td>21134</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>HC_POS1_2</td>
<td>6995</td>
<td>0.30</td>
<td>5.4</td>
</tr>
<tr>
<td>HC_POS1_4</td>
<td>13204</td>
<td>0.75</td>
<td>10.6</td>
</tr>
<tr>
<td>HC_POS1_6</td>
<td>19125</td>
<td>1.01</td>
<td>15.6</td>
</tr>
<tr>
<td>FR_POS1_2</td>
<td>7652</td>
<td>0.22</td>
<td>2.4</td>
</tr>
<tr>
<td>FR_POS1_4</td>
<td>13246</td>
<td>0.37</td>
<td>4.2</td>
</tr>
<tr>
<td>FR_POS1_6</td>
<td>20776</td>
<td>0.65</td>
<td>6.7</td>
</tr>
<tr>
<td>FRR_POS1_2</td>
<td>8442</td>
<td>1.40</td>
<td>27.2</td>
</tr>
<tr>
<td>FRR_POS1_4</td>
<td>14790</td>
<td>2.04</td>
<td>50.6</td>
</tr>
<tr>
<td>FRR_POS1_6</td>
<td>22068</td>
<td>2.30</td>
<td>74.8</td>
</tr>
<tr>
<td>SPC_POS1_2</td>
<td>7445</td>
<td>0.19</td>
<td>2.1</td>
</tr>
<tr>
<td>SPC_POS1_4</td>
<td>13732</td>
<td>0.90</td>
<td>4.1</td>
</tr>
<tr>
<td>SPC_POS1_6</td>
<td>19964</td>
<td>1.04</td>
<td>5.9</td>
</tr>
<tr>
<td>SAND40_POS1_2</td>
<td>9454</td>
<td>8.83</td>
<td>104.3</td>
</tr>
<tr>
<td>SAND40_POS1_4</td>
<td>17360</td>
<td>10.55</td>
<td>205.3</td>
</tr>
<tr>
<td>SAND40_POS1_6</td>
<td>25924</td>
<td>11.31</td>
<td>308.4</td>
</tr>
</tbody>
</table>

The roughness functions of the fully-rough SAND40 from this study fit perfectly with the roughly asymptotic line of the Colebrook-White law. The roughness functions of SPC and FR surfaces are usually distributed between the correlations of [39] and [40] for $k_\theta^+ < 10$. On the other hand, the increasing roughness Reynolds numbers of HC and FRR surfaces do not show the expected increase in their roughness functions and reflects much lower values than in all other correlations given in the literature. Besides, it is clear that the surfaces coated with different type marine coatings do not comply with the Colebrook-White law, and exhibit fairly low roughness functions compared to Colebrook-White law for examined roughness Reynolds number range.

Fig. 4 also includes data from [41], who also performed experiments for the rolling application of a different type FR marine coating, for comparison. While the first three points of [41]’s data are close to the correlation of [39], the following two points form a completely separate trend similar to the data in the present study. In fact, the final point coincides with the first value of the FRR in this study. The reason for the dispersion in [41] is thought to be related to the uncertainty that can be found naturally in experiments, as well as the fact that data are collected at different locations in the flow direction.
The turbulent boundary layer and frictional drag characteristics of new generation marine fouling control coatings

On the other hand, it is clearly seen that the data obtained as a result of this study complement each other for different types of fouling control coatings, creating a completely distinct tendency in the range of $10 \leq k_s^+ < 100$. According to this observation, two different types of correlations can be proposed, namely the power function type and the Colebrook type, respectively:

$$
\Delta U^+ = 0.1302(k_s^+)^{0.7008} \tag{2}
$$

and

$$
\Delta U^+ = 2.2ln(1 + 0.03k_s^+ \tag{3}
$$

Equations (2) and (3) are also printed in Figure 4. As can be seen, the two correlations show very good agreement with the data obtained for marine fouling control coatings tested in this study except for the two higher roughness Reynolds number data gathered for the SPC surface. On the other hand, the available data are limited from the top with $k_s^+ \approx 100$, and the behavior of these surfaces in the higher Reynolds number range is still unclear.

It is seen that; surfaces coated with new generation marine fouling control coatings do not comply with the most important correlations in the literature. Moreover, at around $k_s^+ \approx 70$, in which a transition to fully rough flow regime is expected [42], roughness function values much lower than expected are also observed. Accordingly, it is clear that for such surfaces, it is necessary to extend the number of data in this study to include particularly higher Reynolds numbers, in order to determine new correlations and to confirm the given correlations in this work.

On the other hand, it can be said that the use of a complex roughness length scale which accounts for several characteristic surface roughness parameters seems to be more suitable in the derivation of a correlation between frictional drag and surface roughness for irregularly rough surfaces as in this study. It should be born in mind that the measurement procedure and filter selection in the analysis plays an important role in the calculation of surface roughness parameters and a standard procedure is needed in order to achieve a consistent correlation between surface roughness and roughness function.
4. Conclusions

In this study, flat plate turbulent boundary layer tests were performed with an LDV system on surfaces coated with new generation marine fouling control coatings in different types and the effect of roughness on the average flow parameters and on the frictional resistance was investigated in the transitionally rough flow regime. Key findings from the study can be summarized as follows:

Outer layer similarity was observed for both the surfaces covered with marine fouling control coatings and the fully rough reference surface by the comparison of velocity defect profiles in Rotta scale, which is an important indicator of the outer layer and wall similarity.

For the spray-applied new generation Foul-Release type marine antifouling coated surface, a frictional resistance reduction of up to 4.5% was captured compared to other marine fouling control coatings which were also applied with the same technique.

None of the tested marine fouling control coatings showed compliance with Colebrook-White law with their roughness functions creating a completely distinct trend in the range of $10 \leq k_s^+ < 100$ and at around $k_s^+ \approx 70$, in which a transition to fully rough flow regime is expected, much lower than expected roughness function values were observed.

Accordingly, two new correlations between roughness and frictional resistance have been proposed to represent the behavior of these surfaces by using a complex roughness length scale. However, for confirming the validity of these correlations at the higher Reynolds number range and for other types of irregular surfaces, it would be appropriate to extend this study.
The turbulent boundary layer and frictional drag characteristics of new generation marine fouling control coatings

Burcu Erbaş

Acknowledgements

I would like to thank Prof. Dr. Mehmet Atlar, who supported this study by several means such as providing access to the experiments supported by International Paint Ltd. AkzoNobel and offering the capabilities of the Emerson Cavitation Tunnel. I would also like to express my gratitude to the ITU Rectorate and the Dean of ITU Naval Architecture and Ocean Engineering Faculty for their support for my research visit to the University of Newcastle.

References


The turbulent boundary layer and frictional drag characteristics of new generation marine fouling control coatings


Submitted: 06/05/2018 Burcu ERBAŞ, erbasburc@itu.edu.tr
Accepted: 19/12/2019

Istanbul Technical University Naval Architecture and Ocean Engineering Faculty, 34469, Ayazaga, Istanbul, Turkey
Aims and scopes: 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 shipbuilding as well shipping, offshore and related shipbuilding 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, objects for wind, marine and hydrokinetic renewable energy production and sustainable transportation development at seas, oceans and inland waterways in relations to shipbuilding and naval architecture. 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, autonomous marine vehicles, power plants and equipment onboard. Brodogradnja publishes original scientific papers, review papers, preliminary communications and important professional papers. Editor Kalman Žiha.

Editor in Chief
Professor emeritus Kalman Žiha (55582) (Department of Naval Architecture and Ocean Engineering) kziha@fsb.hr

Editor of Sciences
Professor Nastia Degiuli, (197130) (Department of Naval Architecture and Ocean Engineering) nastia.degiuli@fsb.hr

Executive editor for similarity checking
PhD Student Ivana Martić (Department of Naval Architecture and Ocean Engineering) ivana.martic@fsb.hr

Executive editor for data base administration
PhD Student Andrea Farkas (Department of Naval Architecture and Ocean Engineering) andrea.farkas@fsb.hr

Editorial Board Secretary
Nikolina Zmijarević Dugum (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 Joško Parunov PhD iparunov@fsb.hr (206782)
Professor Smiljko, Rudan PhD smiljko.rudan@fsb.hr (216136)
Associate Professor Jerolim Andrić, PhD jandric@fsb.hr (219630)
Associate Professor Vedran Slapničar, PhD vslapnic@fsb.hr (203716)
Assistant Professor Ivan Čatić, PhD ivan.catic@fsb.hr (275224)
Assistant Professor Neven Hadžić, PhD neven.hadzic@fsb.hr (320461)
Assistant Professor Nikola Vladimír, 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 (retired) (Department of Naval Architecture and Ocean Engineering)
Retired professor Andrea Werner, PhD andreja.werner@fsb.hr (54173)
Retired professor Izvor Grubišić, PhD izvor.grubisic@fsb.hr (76234)
Professor emeritus Vedran Žanić, PhD vedran.zanic@fsb.hr (77913)
Retired professor Rajko Grubišić, PhD rajko.grubisic@fsb.hr (74052)
Retired professor Vačeslav Čorić, PhD veselav.coric@fsb.hr (77520)
Retired Professor Ante Šestan, PhD ante.sestan@fsb.hr (155886)

External Editors (Faculty of Mechanical Engineering and Naval Architecture)
Professor Ivan Juraga, PhD (FSB) Zagreb juraga@fsb.hr

Professor Joško Petrić, PhD (FSB) Zagreb josko.petric@fsb.hr
Professor Neven Duić, PhD (FSB) Zagreb neven.duc@fsb.hr
Professor Nedeljko Štefančić, PhD (FSB) Zagreb nedeljko.stefanic@fsb.hr
Professor Zvonimir Guzović, PhD (FSB) Zagreb zvonimir.guzovic@fsb.hr
Professor Gojko Magazinović, PhD (FSB) Split gojko.magazinovic@fsb.hr
Professor Dražen Lončar (FSB) , PhD Zagreb drazen.loncar@fsb.hr
Assistant Professor Severino Krizmanić, PhD (FSB) Zagre severino.krizmanic@fsb.hr
Assistant professor Ivan Stojanović, PhD (FSB) Zagreb ivan.stojanovic@fsb.hr
External Editors (Others)
Professor Josip Brnić, PhD (RITEH, Rijeka) josip.brnic@riteh.hr (148822)
Professor Jasna Prpić-Oršić, PhD (RITEH, Rijeka) jasnapo@riteh.hr (211853)
Professor Roko Dejhalla, PhD (RITEH, Rijeka) roko@riteh.hr (161532)
Professor Albert Zamarin, PhD (FESB, Split) roko@fesb.hr (207162)
Professor Branko Blagojević, PhD (FESB, Split) bblag@fesb.hr (218434)
Professor Zoran Vukić, PhD (FER, Zagreb) Zoran.Vukic@fer.hr (74412)
Professor Vladimir Androšec, PhD (GRAD, Zagreb) androsec@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) togunja@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) qds903@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) pssahoo@fit.edu
Professor Carlo Francesco Mario Bertorello, PhD (Department of Industrial Engineering Marine Section – Via Claudio 19, 80125 Napoli, Italy) bertorel@unina.it
Professor Yigit Kemal Demirel, PhD University of Strathclyde, Glasgow, United Kingdom yigit.demirel@strath.ac.uk
Professor Xianbo Xiang, PhD (School of Naval Architecture and Ocean Engineering, Huazhong University of science and Technology, Wuhan, China, xbxiang@hust.edu.cn
LIST OF REVIEWERS IN 2019 BY TITLES IN ALPHABETIC ORDER (107)

Mr Frank Rolland
PhD Aichun Feng
PhD Chao Wang
PhD Do Kyun Kim
PhD Ferdi Cakici
PhD Giuliano Vernengo
PhD Hanbing Sun
PhD Ivica Ančić
PhD Ivo Marinić-Kragić
PhD Jae-Hoon Lee
PhD Jason Lavroff
PhD Jichuan Kang
PhD Jinfen Zhang
PhD Josip Bašić
PhD Kristian Bertheussen Karoliussen
PhD Luca Bonfiglio
PhD Mehran Mansoori
PhD Mihael Cipek
PhD Mohamed Adel El-baz
PhD Nader R. Ammar
PhD Nobuaki Sakamoto
PhD Oleksiy Bondarenko
PhD Predrag Ćudina
PhD Riccardo Pellegrini
PhD Serkan Turkmen
PhD Silvia Carpitella
PhD Simone Mancini
PhD student Andrea Farkas
PhD student Bojan Igrec
PhD student Guillermo Chillcce
PhD Tae-Hwan Joung
PhD Tahsin Tezdogan
PhD Tomaso Gaggero
PhD Victor A. Dubrovsky
PhD Wenjun Lu
PhD Yanzhen Chen
Professor Richard Porter
Professor Abit Balin
Professor Alenka Milovanović
Professor Alexander Korobkin
Professor Andrzej Posmyk
Professor Anton Turk
Professor Antonio Carlos Fernandes
Professor Arash Eslamdoost
Professor Branko Blagojević
Professor Carlos Guedes Soares
Professor Chen-Jun Yang
Professor Claudio Pensa
Professor Damir Kolić
Professor Damir Semenski
Professor Daniel Osezua Alkhuele
Professor Davor Ljubas
Professor Denis Benasciutti
Professor Dimitris Konovessis
Professor Dunja Legović
Professor emeritus Ove Tobias Gudmestad
Professor Enrico Rizzuto
Professor Erkan Oterkus
Professor Ermina Begović
Professor Floris Goerlandt
Professor Frano Barbir
Professor Fuat Alarçin
Professor Giles A. Thomas
Professor Gin-Shuh Liang
Professor Giorgio Trincas
Professor Gojimir Radica
Professor Gregory Grigoropoulos
Professor Hironori Yasukawa
Professor Hristos Karahalios
Professor Hrvoje Juretić
Professor Ivo Senjanović
Professor Jerolim Andrić
Professor Ji-Feng Ding
Professor John Chudley
Professor Joško Parunov
Professor Kaj Riska
Professor Kalman Žiha
Professor Kwang-Jun Paik
Professor Manuel F. Silva
Professor Masaaki Sano
Professor Muhsin Aydin
Professor Neven Hadžić
Professor Nikola Vladimir
Professor Noriyuki Sasaki
Professor Pengyao Yu
Professor Pentti Kujala
Professor Pero Prebeg
Professor Peter Fröhle
Professor Petra Grošelj
Professor Prasanta Sahoo
Professor Predrag Rašković
Professor Radoslav Nabergoj
Professor Raju Datla
Professor Roko Dejhalla
Professor Ryszard Wawruch
Professor Saleem Abdullah
Professor Šime Malenica
Professor Stefano Gaggero
Professor Tomislav Mrakovčić
Professor Vedran Mrzljak
Professor Vinko Tomas
Professor Xianbo Xiang
Professor Yakup Kaya
Professor Zao-jian Zou
Professor Zdravko Schauperl
Professor Zhu Zhifeng