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Editorial

POLISH MARITIME RESEARCH is a scientific journal of worldwide circulation. The journal appears as a quarterly four times a year. The first issue of it was published in September 1994. Its main aim is to present original, innovative scientific ideas and Research & Development achievements in the field of:

- Engineering, Computing & Technology,
- Mechanical Engineering,

which could find applications in the broad domain of maritime economy. Hence there are published papers which concern methods of the designing, manufacturing and operating processes of such technical objects and devices as: ships, port equipment, ocean engineering units, underwater vehicles and equipment as well as harbour facilities, with accounting for marine environment protection.

The Editors of POLISH MARITIME RESEARCH make also efforts to present problems dealing with education of engineers and scientific and teaching personnel. As a rule, the basic papers are supplemented by information on conferences, important scientific events as well as cooperation in carrying out international scientific research projects.

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TECHNOLOGY OF SPATIAL DATA GEOMETRICAL SIMPLIFICATION IN MARITIME MOBILE INFORMATION SYSTEM FOR COASTAL WATERS

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ABSTRACT

The paper undertakes the subject of spatial data pre-processing for marine mobile information systems. Short review of maritime information systems is given and the focus is laid on mobile systems. The need of spatial data generalization is underlined and the concept of technology for such generalization in mobile system is presented. The research part of the paper presents the results of analyzes on selected parameters of simplification in the process of creating mobile navigation system for inland waters. In the study authors focused on selected layers of system. Models of simplification for layers with line features and with polygons were tested. The parameters of tested models were modified for the purposes of study. The article contains tabular results with statistics and spatial visualization of selected layers for individual scales.

Keywords: simplification; maritime information systems; mobile systems; generalization

INTRODUCTION

Navigational systems are a specialized kind of Geographic Information Systems (GIS), focused on presenting real-time positional data on map background. From this point of view it can be noticed that also maritime information systems are in fact an examples of GIS technology. Although it is not referred directly by IMO (International Maritime Organization), IHO (International Hydrographic Organization) or IALA (International Association of Lighthouse Authorities) in their performance standards, the basics of systems like ECDIS or VTS rely on GIS technology. The main problem in these systems is proper handling of spatial data to provide desired cartographic products and spatial analysis results.

The scope of using GIS methodology in maritime environment is even wider nowadays, when an e-navigation concept is being widely examined and developed in IMO forum and outside of it. At the moment the concept of bringing more ICT into navigational technology is resulting in many test applications delivered in various R&D projects also for mobile devices.

One of the main problems in any maritime information systems as well as in other GIS systems is delivering of up-to-date model of spatial data for these systems. The most popular approach used mainly in conventional systems is to implement the model proposed by IHO in ENC and ECDIS standards. They include both, data and cartographical model, used in conventional navigation systems. Other applications however may use other data or cartographic models, dedicated for
particular user needs. The spatial data, used in the system, usually coming from various databases has to be integrated and prepared for cartographic visualization. One of the key issues in preparing of cartographical model from existing data sets is called generalization.

In general, map generalization may be defined as adapting the content and the level of detail of a map to suit its scale and purpose. Generalization is one of the most important and yet one of the most difficult subjects in cartography. One of the main problems is that it cannot really be fully automated – it rely much on the decisions made by operator and visual check of cartographic presentation. Nevertheless, one of the assumptions in project was that all processing, including generalization, shall be as much automatized as possible. Thus suitable models and algorithms have been provided to perform this task [1].

The paper shows the entire path of working on generalization in an example mobile navigational system for coastal waters. The description is started with theoretical model of generalization, through the proposed approach to simplification, up to experimental research and it results, showing the efficiency of the method used. The idea was to present the possibilities of spatial data handling in any maritime systems, however the research themselves are focused on system for mobile devices as the kind of device is also an important factor in generalization process and the mobile devices are the future of navigation. Although the paper describes generalization in general, the research focuses on geometrical simplification as one of the methods of generalization.

**MARITIME INFORMATION SYSTEMS**

Maritime information systems are understood in this paper as the systems in which maritime spatial data are managed and visualized. This means that any GIS covering maritime area and processing spatial data affecting sea environment can be treated as maritime information system. The most popular examples here can be Electronic Chart Display and Information System (ECDIS) on board of the ship used for navigation or Vessel Traffic Management System (VTS or VTMS) ashore used for traffic management.

In general, maritime information systems may be divided into conventional and non-conventional systems. Conventional are these, which follow any particular standard given by international body, like ECDIS standard by IHO [2] or VTS standard by IALA [3]. There is however more and more applications used by so called non-conventional users that use other systems for navigational purposes. These also can be understood as maritime information systems, and as non-standardized, they require more attention in the process of spatial data management. It should be added anyway that in both kinds of systems a quality of data is a key issue for usefulness of the system. Traditional sources of data, like ashore and off-shore surveying are nowadays exchanged or supplemented with modern techniques, among which remote sensing performs the leading role. The devices like laser scanner, spectral cameras, high-speed cameras etc. are a crucial source of spatial data used in official ENCs, topographical databases, but also in non-conventional databases. The advantages of these methods can be found in [4] with an application also for maritime areas.

An important issue that has to be undertaken in deliberations about maritime information systems is the mobility of them. It can be noticed, that in fact any ship system can be treated as mobile, as it is not used stationary, but on the moving platform. However, when talking about spatial data handling, it is more reasonable to connect mobility to the cartographical model and to the device on which the system runs as it is in the concept of mobile cartography [5, 6].

**A. Shore systems**

The most popular maritime information shore system is VTS. Standardized by IALA, VTS aims at providing real-time information about traffic in the particular sea-area to the VTS operator in shore center. The main information function, may be enhanced by traffic management or other features (like navigational assistance), depending on legal issues and needs in particular VTS area [7]. The system is stationary and the cartographic model is static and usually based on ENC.

In the coastal areas however also other systems, not only for supporting navigation, may be found. In general they can be treated as a group of so called coastal GIS systems and mostly focus on environmental modelling [8]. They are usually specialized in a particular aspect of maritime environment and in general aims at providing comprehensive knowledge about it. Data model here is usually more complicated and includes also properly formatted spatial data from other databases than ENC. The model is sometimes supplemented with raster data [9, 10].

Interesting examples of wider concept of information system are also River Information Services. RIS may be defined as a collection of ICT systems for providing various information to wide scope of users in river navigation and transportation. These systems require also special handling of spatial data, not only for Inland Electronic Navigational Charts, but also for spatial analysis [11, 12].

**B. On-board systems**

The most numerous group of information systems at sea are on-board systems, used for obtaining spatial information around own ship. Most of them aim at navigational purposes, but there are also examples of other interesting applications. Once again the division between conventional and non-conventional systems can be made.

**1) Conventional on-board information systems**

The development of conventional navigational information systems started when traditional electronic devices, like radar begun to provide more and more information. Nowadays tracking radar used on board is in fact an ICT system in which radar antenna is one of input sources, while others like AIS or ENC database are also used. The differences between radars
and ECDIS still exists, as their origin is different, however it can be noticed that the scope of information they provide are more and more similar [13]. Thus, many systems integrating this information appeared. Among them the conventional example is Integrated Navigational System (INS). According to Resolution MSC.252(83), INS is a navigational system, which provides multifunctional displays integrating at least the functions of route planning and collision avoidance [14].

From spatial data handling point of view, all conventional systems have to follow data and cartographic model provided in ENC standard, which makes it inflexible for other purposes. The issue common for all conventional and non-conventional systems is the requirement of real-time presentation of dynamical data, which requires separate algorithms for data quality control and uncertainty management [15, 16].

2) Non-conventional navigational systems

Apart from traditional conventional systems there is a variety of additional features, which have not been adopted by IMO yet, but which are being developed in R&D projects. Important enhancements of standardized systems for navigational purposes are for example Navigational Decision Support Systems. They provide recommendation on vessel movement, taking into account vessel safety, rules of the road and route efficiency [17, 18]. These system usually use traditional data model, however they can be more flexible and provide non-standardized cartographic model. The crucial information here is dynamic data about own ship and target ships. This requires advanced fusion algorithms which provide reliable data suitable for cartographic presentation [19, 20].

3) Survey systems

Talking about non-conventional systems, it should be also mentioned about complete software toolbox for offshore hydrographic survey on board of research vessels. These systems collect data from connected instruments and sensors, e.g. navigation systems, weather sensors, sound propagation sensors and hydrographic instruments. They store all data in a relational database. Often the products have the possibility of integration with WMS/GIS and other software solutions through import and export of standard data formats. The basic for all these systems is very accurate bathymetric chart providing data about seabed as a basis for any survey job [21, 22, 23]

C. E-navigation systems

E-navigation is a wide concept undertaken by IMO to provide legal and technological framework for introducing more ICT technologies into maritime solutions and applications. Any particular standards and solutions have not been adopted yet, but, as a concept has been developed for several years already, some applications have been already proposed and tested. The works are carried in working groups with the leading role of IMO, as well as in R&D projects. IMO has adopted in strategy implementation plan [24], five basic solution towards which the works are carried out. On of this solution is integration and presentation of integrated information in graphical displays received via communication equipment, which means also working on cartographic model on navigational displays.

Recent works in the field of e-navigation provides interesting results in the field of so called Maritime Cloud, which is a service for exchanging maritime spatial data. The concept is however more technical-centered that cartographic. Some interesting issues about mobility have been anyway undertaken in so called Maritime Android Approach inside Maritime Cloud.

D. Maritime mobile information systems

The popularity of mobile devices like smartphones or tablet, enforced maritime community to employ these devices also at sea. The survey undertaken for the purpose of this paper has shown that mobile information systems at sea are mostly commercially sold support systems for non-conventional users, mainly tourists, sailors, and leisure boats owners. The systems are dedicated for mobile devices, however it must be said that most of them do not follow the achievements of mobile cartography, and in such case does not fully use the possibilities given by mobile devices. Most of them are also provided with own-made charts and some use official S-57 charts. In some cases vector information is also supplemented with raster charts, while some use only RNC’s. Typical chart information is enhanced with route data, weather and hydro meteorological information.

The research has shown that no advanced cartographic model is used in these applications and the focus is laid mostly on functionality and not on presentation itself. This leads to the conclusion that improvement of cartographic model is possible. Based on these and on questionnaire research, the assumption has been made that improvement of cartographical model would lead towards better usability of the systems.

Regardless of what cartographic presentation model is used, the pre-processing of spatial data imported from any vector database should be made. Taking into account future presentation of data this shall include also generalization algorithms to prepare the cartographic model. These issues are even more important in mobile systems, where the possibility of real time data modification is restricted. Thus the generalization process shall be performed in the stage of data pre-processing. Large number of features and its complexity in wider areas may be significant computational challenge for average mobile device. Especially, if one takes into account, that one of the major requirements for the system is working fluently online in real time. In such a situation the number of features and vertices shall be significantly reduced with the scale, which leads to simplification as a part of generalization.

RESEARCH CONCEPT

The research presented in the paper aimed at showing possible generalization concept for mobile navigational system and at analyzing of geometrical generalization methods for this
purpose. The research has been made with mobile navigational system MOBINAV, which was selected as an example of systems dedicated for mobile devices and touristic purposes. An area of Szczecin harbor was chosen as a testbed. The research focused on restricted areas of coastal waters as there are much more complicated in geometry and as such, the spatial data from these areas are much more difficult to process in mobile system. By choosing it the authors were expecting receiving of more valuable results. The assumptions of an experiment itself are given in the further part of the paper.

A. MOBINAV as an example of mobile maritime information system

Mobile navigation MOBINAV is an example of GIS (Geographic Information System) designed for mobile devices. It was originally created for inland waters, however it performs well also in maritime coastal waters. Development of the new system called MOBINAV was a fundamental objective of R&D project. The main assumptions of the system are based on its destiny – system is dedicated for recreational users with so called pleasure crafts. The project aims at developing such technology to implement modern mobile cartographic presentation model into navigational system on mobile devices. From final user’s point of view, the system will be yet another application for navigation, but this time it will be tailored to him/her and the cartographic presentation will be significantly better than in other applications.

During the work on building of the system it was decided to elaborate an own model of spatial data MODEF (MO-binav Data Exchange Format). The model of mobile cartographic presentation described in [25] was used as a base and modified for the purpose of model implementation in MOBINAV. Large dynamic of mobile presentation enforces frequent change of scale of observed presentation. This leads to a need of generalization of input data, including spatial simplification. The research presented in the paper focus on this aspect of generalization as the most computational-demanding issue.

B. Data simplification issue

In mobile navigation system MOBINAV, the authors focus on ensuring efficiency of cartographic communication, which has relationship with the effectiveness of contents, utility of chart and its usefulness to the user. During the development of the system model the needs of future users were taken into account, including the purpose of using and technical capabilities of mobile devices. Process of generalization is needed in order to ensure a flexible presentation mode for a user in a situation, when geospatial information has to be visually communicated on a small-display device.

Spatial simplification is one of the generalization techniques. During this transformation process number of vertex in a line that is approximated by a series of points is reduced. Unnecessary coordinate information from line features is removed, whilst maintaining the perceptual line’s character. The main advantage of using the simplification is contributed to increase of readability map. Additionally, some practical considerations of simplification include reduced storage space or faster plotting time.

PROPOSAL OF DATA SIMPLIFICATION

Generalization of map content depends on its purpose, scale, level of details and data. All of these factors have been included in Fig. 1, presenting general schema of generalization in MOBINAV. The purpose is directly related with selection of source data and term of partial geocompositions, scale refers to definition of geocompositions and SCAMIN/SCAMAX definition, while level of details and data may be combined with each element of created schema. Therefore, the generalization process of spatial data in the system can be considered at several levels (Fig. 1).

The first step of generalization was determination of the values of SCAMIN and SCAMAX attributes. These attributes are responsible for the appearance of the object or its disappearance from the geovisualization window of mobile device. Analyzing these, authors drew a number of important conclusions. One of them is the fact that determination of partial geocompositions, which are responsible for displaying a specific set of the features, is insufficient and does not ensure the perspicuity of map content. Defining attributes SCAMIN and SCAMAX proved to be satisfactory in the case of point features. For the purposes of correct interpretation of the map and to avoid the effect of “littering” the small screen with too many details in case of the line features and polygons, geometric simplification has to be performed. The research presented in the paper focus on this aspect of generalization. Classical algorithms of simplification were used in the research with properly set parameters on each of the scale levels.

Given the requirements of the future user of MOBINAV system, a separate simplification model for each of the layers
in system was created. These models are combinations of the methods listed below. During the simplification of line features, Douglas-Peucker algorithm was mainly implemented. During the generalization of polygon features, simplification method was applied, maintaining the basic shapes and sizes of the features. The parameter of simplification tolerance and parameter determining the minimum area of the feature was used. Additionally, features within an established distance were merged. Smoothing tool for shape and size of buildings and PEAK method (Polynomial Approximation with Exponential Kernel) were used as well. In addition, selection tool was employed and features with secondary importance to the user of the system during navigation mode, were eliminated. The overriding factor that has been taken into account during the research of simplification methods was the limitation of sharpness of human eyes. It has been taken based on literature that it is about 0.2 millimeter at a distance of 30 centimeters from the human eye [1].

The research covered all 29 vector layers of MODEF data model used in MOBINAV, however the results for only two selected layers are given in the paper, due to limited number of pages. The results for the other layers are similar and the conclusions are suitable for all analyzed data.

**EXPERIMENT**

For the purposes of the experiment in this paper, two simplification models were prepared for two layers – PIER_L (wharfs) and VEGETA (vegetation). The parameters of the tested models were modified for different scales. Simplification models have been tested on original data from Szczecin Port. In the next step, statistics were calculated and some of them were represented as relative values for better comparison of simplification effectiveness. The outcomes are rounded to the second decimal place. The results were shown in tabular form and as and spatial visualization. The last stage was their analysis. All research was carried out in ArcGIS software, using ArcCatalog and ArcMap applications and the simplification models were prepared as separate tools with the use of ModelBuilder.

**A. Test area**

Part of area of Szczecin Harbour was selected to be a test area. The original data was extracted from ENC (Electronic Nautical Chart) with compilation scale 1: 2 000 and Polish Topographic Database with compilation scale 1: 10 000 and clipped to the area. Two layers have been selected for the research – VEGETA, presenting vegetation, as an example of polygon features and PIER_L, presenting wharfs, as an example of line features. Thus a set of original features to be simplified was created. Fig. 2 shows the selected test area.

Layer with acronym PIER_L represents wharfs. Apart of general attributes, PIER_L also accepts attribute CATPIR (category of wharf): retaining wall, shore strengthened, wharf and other. The quantitative characteristics of PIER_L in test area are as follows:

- number of features is 277
- number of vertices is 5 621
- total length of all features is 79 463,51 meters
- and total volume is 198 KB.

Layer with acronym VEGETA represents vegetation. The additional attribute of this layer is CATVEG (category of vegetation), which can take the value of: forest, bush, cultivation, park, agricultural area, grass, rushes and other. The quantitative characteristics in the test area covers:

- number of VEGETA features is 2 476
- number of vertices is 179 160
- total area of all features is 166,58 square kilometers
- total volume is 3,66 MB.

Both layers were processed with dedicated simplification models prepared.

**B. Simplification models**

The idea of the research was to build a simplification model for each dataset with the use of simplification tools and selection of proper parameters. The output of each model was a set of simplified features for each output scale. Tools parameters were chosen for each scale separately.

While simplifying layer PIER_L, authors focused only on four scales: 1:7 500, 1:10 000, 1:17 500 and 1:25 000. In smaller scales wharfs will not be displayed. For the generalization of PIER_L networks, simplify line tool was used. It simplifies a line by removing small extraneous bends from it, while maintaining its basic shape. Its operation is based on Douglas-Peucker algorithm, which is the most commonly used global
simplification algorithm in GIS (Geographical Information System). In this algorithm, at the beginning all the vertices between the start and end original vertices are marked to be kept. Next, for every intermediate vertex, its perpendicular distance to the initial simplified line (connecting the start and end original vertices) is calculated. Vertex with maximum distance larger than the founded tolerance is used to divide the initial simplified line into two segments. This step is repeated until the perpendicular distance for each original vertex is smaller than the founded tolerance. All other vertices are deleted.

The basic flow in simplification model for PIER_L, as a model in Model Builder with one output scale is presented in Fig. 3.

The layer VEGETA was simplified for each of the scales used in the system and simplify building tool was used. It simplifies the boundary of polygons while retaining their fundamental shape and size. Very important is that simplification process preserves and enhances orthogonality. The tool simplifies separate polygons and group of polygons (connected with straight lines). It is adapted to a whole feature, not only for an feature segment [26].

The basic flow in simplification model for VEGETA with one output scale is presented in Fig. 4.

C. Selected parameters and assessment criteria

The limitation of sharpness of human eyes has been taken into account during simplification methods research. As previously mentioned, it is about 0.2 millimeter at a distance of 30 centimeters from the human eye. This value was used as a reference for setting thresholds in the research. The authors decided to implement the following parameters for each of the scales:

- ½ limitation of sharpness of human eyes (further referred to as A);
- 1 ½ limitation of sharpness of human eyes (further referred to as B);
- 2 limitation of sharpness of human eyes (further referred to as C);
- 3 limitation of sharpness of human eyes (further referred to as D);
- 3 limit of sharpness of human eyes (further referred to as E).

Simplify line algorithm keeps the so-called critical points, which depict the essential shape of a line and remove all other points. The main parameter of this algorithm is maximum allowable offset – the tolerance that determines the degree of simplification. All parameters used in tests are presented in Table 1.

Tab. 1. Parameters (maximum allowable offset) for PIER_L

<table>
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<th>Scale</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
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<td>1.50m</td>
<td>2.25m</td>
<td>3.00m</td>
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<td>2.00m</td>
<td>3.00m</td>
<td>4.00m</td>
<td>6.00m</td>
</tr>
<tr>
<td>1:17500</td>
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<td>3.50m</td>
<td>5.25m</td>
<td>7.00m</td>
<td>10.50m</td>
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<tr>
<td>1:25000</td>
<td>2.50m</td>
<td>5.00m</td>
<td>7.50m</td>
<td>10.00m</td>
<td>15.00m</td>
</tr>
</tbody>
</table>

During simplifying of vegetation the following parameters have been changed:

- simplify tolerance, that sets the tolerance for simplification (presented in meters);
- minimum area, that sets the minimum area for a simplified polygon to be preserved (presented in square meters). The parameters used during experiments are presented in Table 2.

Tab. 2. Parameters (simplification tolerance and minimum area) for VEGETA

<table>
<thead>
<tr>
<th>Scale</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
</tr>
</thead>
<tbody>
<tr>
<td>1:7500</td>
<td>0.75m</td>
<td>0.56m²</td>
<td>1.50m</td>
<td>2.25m²</td>
<td>2.25m</td>
</tr>
<tr>
<td>1:10000</td>
<td>1.00m</td>
<td>1.00m²</td>
<td>2.00m</td>
<td>4.00m²</td>
<td>3.00m</td>
</tr>
<tr>
<td>1:17500</td>
<td>1.75m</td>
<td>3.06m²</td>
<td>3.50m</td>
<td>12.25m²</td>
<td>5.25m</td>
</tr>
<tr>
<td>1:25000</td>
<td>2.50m</td>
<td>6.25m²</td>
<td>5.00m</td>
<td>25.00m²</td>
<td>7.50m</td>
</tr>
<tr>
<td>1:50000</td>
<td>5.00m</td>
<td>25.00m²</td>
<td>10.00m</td>
<td>100.00m²</td>
<td>15.00m</td>
</tr>
<tr>
<td>1:100000</td>
<td>10.00m</td>
<td>100.00m²</td>
<td>20.00m</td>
<td>400.00m²</td>
<td>30.00m</td>
</tr>
<tr>
<td>1:250000</td>
<td>25.00m</td>
<td>625.00m²</td>
<td>50.00m</td>
<td>2500.00m²</td>
<td>75.00m</td>
</tr>
</tbody>
</table>

To assess the performance of the algorithms for various parameters, various statistics have been calculated. First of all the number of deleted vertices in each of the generalized
layers was used and corresponding to this number of remained vertices in correspondence to original data. This showed the efficiency of an algorithm. Additionally for line feature total length of features and for polygonal features total area of features is given in relation to original set.

To judge computational overload of data in future application, the volume in KB and MB is shown. Finally one of the parameters of evaluation is visual assessment of the layers after simplification in a particular scale (one star to five stars). It should be noted that more stars in each field represent a better performance in terms of the assessment results in the selected scale.

D. Results

This section compares received results for different scales and various parameters set out above in point C.

A summary of statistics for layer PIER_L is given in Table 3, which contains number of vertices, number of deleted vertices, total length, data volume and visual assessment. While simplifying the layer, the model used does not remove any features. The results of simplification for selected scales are compared with source objects. Number of features was decreased along with the increase of the parameter maximum allowable offset. Using A parameter for scale 1:7 500 this number was reduced by more than a half i.e. until 44%. While for scale 1:25 000 it was reduced almost four times. During the use of the E parameter for scale 1:7 500 it was decreased nearly five times and for scale 1:25 000 almost six times. The results for total length of all features were slightly changed – till 3%. Data volume varied from 157 KB to 133 KB. For larger scales visual assessment is better when using lower value of maximum allowable offset (A and B parameters). Whereas for smaller scales visual assessment is better using bigger values of this parameter.

Fig. 5 represents received number of vertices for layer PIER_L. It can be seen that number of vertices is reduced with scale, but the relationship is not quite linear and for smaller scales the differences for different parameters are smaller.

Fig. 6 represents total length for all features in layer PIER_L. Total length of all features was varied very slightly. It is in the range from 79 304.16 meters to 77 261.9 meters. The biggest decline in the value can be seen using E parameter. Anyway this time the differences for various parameters are bigger, while the scale gets smaller. The decrease in total length is partly caused by reduced number of features, but also by simplification of shapes.

Tab. 3. Comparison of statistics for PIER_L

<table>
<thead>
<tr>
<th>Scale</th>
<th>N of V [%]</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
</tr>
</thead>
<tbody>
<tr>
<td>1:7500</td>
<td>44,07%</td>
<td>32,65%</td>
<td>28,13%</td>
<td>25,21%</td>
<td>22,01%</td>
<td></td>
</tr>
<tr>
<td>N of DV</td>
<td>3144</td>
<td>3786</td>
<td>4940</td>
<td>4204</td>
<td>4384</td>
<td></td>
</tr>
<tr>
<td>TL [%]</td>
<td>99,80%</td>
<td>99,54%</td>
<td>99,36%</td>
<td>99,14%</td>
<td>98,82%</td>
<td></td>
</tr>
<tr>
<td>DV [KB]</td>
<td>157</td>
<td>154</td>
<td>146</td>
<td>142</td>
<td>138</td>
<td></td>
</tr>
<tr>
<td>VA</td>
<td>*****</td>
<td>*****</td>
<td>*****</td>
<td>*****</td>
<td>*****</td>
<td></td>
</tr>
<tr>
<td>1:10000</td>
<td>38,69%</td>
<td>29,11%</td>
<td>25,21%</td>
<td>22,68%</td>
<td>20,53%</td>
<td></td>
</tr>
<tr>
<td>N of DV</td>
<td>3446</td>
<td>3985</td>
<td>4204</td>
<td>4346</td>
<td>4467</td>
<td></td>
</tr>
<tr>
<td>TL [%]</td>
<td>99,71%</td>
<td>99,40%</td>
<td>99,14%</td>
<td>98,91%</td>
<td>98,54%</td>
<td></td>
</tr>
<tr>
<td>DV [KB]</td>
<td>147</td>
<td>144</td>
<td>140</td>
<td>138</td>
<td>137</td>
<td></td>
</tr>
<tr>
<td>VA</td>
<td>*****</td>
<td>*****</td>
<td>*****</td>
<td>*****</td>
<td>*****</td>
<td></td>
</tr>
<tr>
<td>1:17500</td>
<td>30,78%</td>
<td>23,63%</td>
<td>21,17%</td>
<td>19,53%</td>
<td>18,09%</td>
<td></td>
</tr>
<tr>
<td>N of DV</td>
<td>3891</td>
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<td>4431</td>
<td>4523</td>
<td>4604</td>
<td></td>
</tr>
<tr>
<td>TL [%]</td>
<td>99,48%</td>
<td>99,00%</td>
<td>98,67%</td>
<td>98,32%</td>
<td>97,92%</td>
<td></td>
</tr>
<tr>
<td>DV [KB]</td>
<td>143</td>
<td>141</td>
<td>137</td>
<td>136</td>
<td>135</td>
<td></td>
</tr>
<tr>
<td>VA</td>
<td>****</td>
<td>****</td>
<td>****</td>
<td>****</td>
<td>****</td>
<td></td>
</tr>
<tr>
<td>1:25000</td>
<td>26,85%</td>
<td>21,53%</td>
<td>19,23%</td>
<td>18,24%</td>
<td>16,60%</td>
<td></td>
</tr>
<tr>
<td>N of DV</td>
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<td>4411</td>
<td>4540</td>
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<td>4688</td>
<td></td>
</tr>
<tr>
<td>TL [%]</td>
<td>99,26%</td>
<td>98,75%</td>
<td>98,24%</td>
<td>97,97%</td>
<td>97,23%</td>
<td></td>
</tr>
<tr>
<td>DV [KB]</td>
<td>141</td>
<td>139</td>
<td>136</td>
<td>135</td>
<td>133</td>
<td></td>
</tr>
<tr>
<td>VA</td>
<td>***</td>
<td>***</td>
<td>***</td>
<td>***</td>
<td>***</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 5. Number of vertices for PIER_L.

The largest decrease in the number of vertices can be seen using ½ limitation of sharpness of human eyes. It is set in the range from 2 477 vertices to 1 509 vertices. However, the smallest decline can be seen using E and D parameters. For the latter it is in the range from 1 237 to 933.

Fig. 6. Total length for PIER_L. Total length of all features was varied very slightly. It is in the range from 79 304.16 meters to 77 261.9 meters. The biggest decline in the value can be seen using E parameter. Anyway this time the differences for various parameters are bigger, while the scale gets smaller. The decrease in total length is partly caused by reduced number of features, but also by simplification of shapes.
A comparison of statistics for layer VEGETA is given in Table 4. This time number of features, number of vertices, number of deleted vertices, total area, data volume and visual assessment are given.

Looking at the values in the table, following observations can be made. Increasing of the minimum area parameter was very important for the number of features after simplification. Number of deleted features is in the range from 0.04% to 74.92%. This means that for the smallest scale only every fourth feature remained. This for sure is major reduction of computational load for application. It can be noticed thus, that joined impact of reduced scale and increased parameter value has an important impact on data. On the other hand, the decrease of number of vertices for particular scales was very similar for each of the parameters (from A to E parameters) – approximately 40%. Values related to total area were changed irregularly. In general they remain on the same level, however slight increase of area may be noticed in most of the cases. This means that the smallest areas are reduced and that simplification of polygons geometry does not mean reduction of the area. Changes in data volume are significant – it was decreased by more than three times, and this is mostly due to reduction of number of features. However it should be noticed that also the number of vertices is reduced significantly in smaller scales, which must have an influence on number of data.

For larger scales visual assessment is better when using lower value of maximum allowable offset and for smaller scales it is better using greater value of this parameter.

Fig. 7 represents received number of features for layer VEGETA. For the smallest value of minimum area (for A parameter and scale 1:7 500 it was equal 0,56m²) only one feature has been deleted. Whereas for its biggest value (for E parameter and scale 1:25 000 it was equal 22 500m²) 1 855 features have been eliminated – only 621 objects have been remained. It can be noticed that in very small scales (less than 1 : 50000) the number of features drops rapidly, while for the larger scales it is insignificant in fact. This leads to a conclusion that most of the features remain in these scales, but their shape is being modified. In smaller scales, the impact of method’s parameters are much bigger.

Fig. 8 shows the number of deleted vertices for layer VEGETA.

Tab. 4. Comparison of statistics for VEGETA

<table>
<thead>
<tr>
<th>Scale</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>NoF [%]</td>
<td>NoV [%]</td>
<td>NoDV</td>
<td>TA [%]</td>
<td>DV [MB]</td>
</tr>
<tr>
<td>1:7500</td>
<td>99,96%</td>
<td>81,60%</td>
<td>32964</td>
<td>99,98%</td>
<td>3,21</td>
</tr>
<tr>
<td></td>
<td>99,72%</td>
<td>76,66%</td>
<td>41824</td>
<td>99,98%</td>
<td>3,07</td>
</tr>
<tr>
<td></td>
<td>99,64%</td>
<td>72,14%</td>
<td>49909</td>
<td>100,00%</td>
<td>2,95</td>
</tr>
<tr>
<td></td>
<td>99,43%</td>
<td>68,93%</td>
<td>55666</td>
<td>99,99%</td>
<td>2,86</td>
</tr>
<tr>
<td></td>
<td>99,19%</td>
<td>64,46%</td>
<td>63682</td>
<td>100,02%</td>
<td>2,73</td>
</tr>
<tr>
<td>1:10000</td>
<td>99,88%</td>
<td>80,05%</td>
<td>35736</td>
<td>99,98%</td>
<td>3,17</td>
</tr>
<tr>
<td></td>
<td>99,64%</td>
<td>73,52%</td>
<td>47444</td>
<td>99,99%</td>
<td>2,99</td>
</tr>
<tr>
<td></td>
<td>99,43%</td>
<td>68,93%</td>
<td>55666</td>
<td>99,99%</td>
<td>2,86</td>
</tr>
<tr>
<td></td>
<td>99,23%</td>
<td>65,68%</td>
<td>61486</td>
<td>99,99%</td>
<td>2,77</td>
</tr>
<tr>
<td></td>
<td>99,15%</td>
<td>61,32%</td>
<td>69295</td>
<td>100,02%</td>
<td>2,65</td>
</tr>
<tr>
<td>1:17500</td>
<td>99,64%</td>
<td>75,03%</td>
<td>44744</td>
<td>99,98%</td>
<td>3,03</td>
</tr>
<tr>
<td></td>
<td>99,31%</td>
<td>67,27%</td>
<td>58642</td>
<td>99,98%</td>
<td>2,81</td>
</tr>
<tr>
<td></td>
<td>99,19%</td>
<td>63,15%</td>
<td>66023</td>
<td>100,02%</td>
<td>2,7</td>
</tr>
<tr>
<td></td>
<td>99,03%</td>
<td>59,47%</td>
<td>72606</td>
<td>100,04%</td>
<td>2,6</td>
</tr>
<tr>
<td></td>
<td>98,71%</td>
<td>54,44%</td>
<td>81622</td>
<td>100,04%</td>
<td>2,46</td>
</tr>
</tbody>
</table>

a. Number of features.  
b. Number of vertices.  
c. Number of deleted vertices   
d. Total area   
e. Data volume  
f. Visual assessment
In this case the situation is different. The relationship is almost linear and the reduced number of vertices significantly increases for all scales. The grade of diagrams of number of deleted vertices for particular simplification tolerance are close to each other – their growth is very similar. Minimum number of deleted vertices is 32,964 (for A parameter and scale 1:7500) and maximum is 13,7531 (for E parameter and scale 1:250,000). For each of simplification tolerance parameters this value increases in a similar way and it is in the range from 68,602 (for A parameter) to 75,439 (for B parameter). The conclusion might be that although in bigger scales the features remain, their shape is simplified as the vertices are deleted.

CONCLUSIONS

The paper presented research on geometric generalization methods for maritime mobile navigation system dedicated for touristic purposes in coastal waters. At first, short survey of maritime information system was given in the aspect of mobility and spatial data management. The focus of the research was laid on mobile system, which is intended for mobile devices like smartphones or tablets.

For this purpose suitable generalization model has been proposed and the experiment on selected layers was performed. The aim of the research was to determine relation between methods parameters and method efficiency. The base for parameters calculation was the value limitation of sharpness of human eyes, which was multiplied by specified factor. The efficiency of methods was determined by a number of vertices, number of features, data volume and additionally visual assessment.

The main conclusion is that, taking into account visual assessment, for bigger scales smaller parameter shall be used and for smaller scale bigger values of parameter performs better. On the other hand analysis of other criteria does not follow directly these findings. Thus, in general it is proposed to use value of 1½ limitation of sharpness of human eyes (C in the research) or to use various parameters for different scales.

The most important factor for real-time mobile application is in fact number of objects, which is why additionally smaller objects shall be completely deleted. Achieved results allow stating, that in case of mobile device and real-time applications the presentation model should include more simplified features than in case of stationary model. However dynamical mobile model should consists of many features generalized separately for different scale ranges.

ACKNOWLEDGMENT

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Marine Technology Ltd.
Szczecin
POLAND
MAPSERVER – INFORMATION FLOW MANAGEMENT SOFTWARE FOR THE BORDER GUARD DISTRIBUTED DATA EXCHANGE SYSTEM

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Sylwester Kaczmarek, D. Sc.
Magdalena Młynarczuk, M. Sc.
Marcin Narloch, Ph. D.
Gdańsk University of Technology, Poland

ABSTRACT

In this paper the architecture of the software designed for management of position and identification data of floating and flying objects in Maritime areas controlled by Polish Border Guard is presented. The software was designed for managing information stored in a distributed system with two variants of the software, one for a mobile device installed on a vessel, an airplane or a car and second for a central server. The details of implementation of all functionalities of the MapServer in both, mobile and central, versions are briefly presented on the basis of information flow diagrams.

Keywords: distributed system, information flow control, maritime areas monitoring

INTRODUCTION

A vital element of the Polish Republic maritime border protection is the Automatic National System of Radar Control for Maritime Areas of Poland (Zautomatyzowany System Radarowego Nadzoru Polskich Obszarów Morskich - ZSRN) which is an integrated security system. The main task of this system is to ensure the continuous monitoring of Polish maritime areas, at least territorial and internal sea waters [5]. The purpose of this system is to improve the work of Border Guard (BG) units so they could more efficiently minimize threats which might occur in border areas. The most important of these threats are ([7], [9]):

- organized international drug trafficking;
- illegal migration through the territory of the Republic of Poland to Western Europe;
- smuggling goods through Polish seaports and harbors and on ships or other vessels;
- smuggling psychotropic substances and radioactive weapons, stolen yachts and motor boats, cars, board electronic equipment, goods which are subject to excise taxes;
- illegal fishing and trade of caught fish by Polish and foreign fishing vessels.

A hierarchical structure of ZSRN consist of the following essential components ([4], [9]):

- Main Control Center (MCC) and Reserve Main Control Center (RMCC);
- Division Control Centers (DCCs);
- Local Control Centers (LCCs) and Observation Posts (OPs) which are distributed along the coast of Poland;
- Mobile observation posts (MOPs).

OPs and MOPs constitute main sources of information in ZSRN. OPs are basic sources while MOPs are complementary sources of information. Since ZSRN is an open system it can be readily expanded or integrated with other systems [9].

The MapServer presented in this paper is a part of developed proposition for ZSRN system expansion presented in Figure 1. This expansion comprises consoles and Universal Radio Controllers (URCs) installed in MOPs (mobile BG units) which can move by sea, land or air. These mobile units are connected to designated OP by means of ad hoc radio network [3] and URC in OP relays MOPs communication to core BG's IP network.

Fig. 1. The functional diagram of the developed system
The URC which provides services to the consoles controls radio voice, text and data communications. Additionally, the consoles can present current or archival data about objects of interests on maps or in tables. Implementation of this additional functionality is the requirement of the MapServer software. Although from point of view of the console operator the MapServer simply provides on request data about detected objects in form that can be readily presented on map or in tables, it also automatically collects data from ARPA radar, AIS and GPS receivers connected to the URC. Since the data sources are connected to the URC and consoles, which are MapServer clients, are connected to URC, each mobile unit’s URC runs the mobile version of the MapServer (the mobile MapServer). Additionally, data collected by the MapServer from local sources, apart from being collected in URC and presented on consoles, have to be transferred to central MapServer and integrated there with data from other MOPs and external sources (web service in Figure 1) ([4], [9]) so they can be presented on stationary consoles in CON/ZCON and mobile consoles in MOPs on request. Communication between mobile MapServers and the central MapServer is managed by URCs in MOP and designated OP and involves communication through a digital radio channel and a fixed IP network. Moreover, since mobile consoles, URCs and the central MapServer use different hardware, the communication module needs to be platform independent. The distribution of data over many BG units spread over large area required solution that could manage a specific distributed system of information collection, storage and retrieval with many local and single central database [8]. Moreover the nodes in this system are interconnected via links containing unreliable radio channels [3].

The MapServer discussed in this paper have been implemented in C++ using open source tools [6] and tested in laboratories on the equipment designed and manufactured under a grant on real and simulated data [12]. In the remaining part of the paper functionalities and the structure of the MapServer is discussed.

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THE GENERAL MAPSERVER ARCHITECTURE

Since the MapServer is used in two different environments, URC and central server (CENTER), which differ not only in hardware but, what is more important, require different sets of functionalities, there are two distinct MapServer architectures. Dissimilarities in mobile (URC) and central MapServer architectures reflect the differences in provided functionalities.

The fundamental task of the mobile MapServer is to store data about objects collected from local sources and provide these data (current or archival) on request to consoles attached to the URC so they can be presented in the console on maps (Figure 2) or in tables (Figure 3) by the Map Service and the Data Service respectively (Figure 4).

To be able to provide necessary data to consoles’ services the mobile MapServer needs to maintain a local database with data collected by the mobile unit’s ARPA radar, AIS or GPS receiver (units’s own position). Additionally, consoles’ operators can manually add notes associated with selected objects. Beside managing the local database the mobile MapServer transmits position data and notes collected by the mobile unit to the central MapServer (CENTER). Moreover, since mobile unit might have an incomplete image of surface situation, position data gathered in central MapServer from other mobile units and external sources, like web service, can be requested by consoles’ operators. In order to limit digital radio channel load, the data received from central MapServer, apart from being presented to the operator, are integrated into local database for future reference.

The mobile MapServer (Figure 4) supports two mobile consoles connected to the URC using TCP/IP protocol and the connection to the central MapServer is achieved through the radio channel using Dual Transfer Protocol (DTM) ([1], [2], [10]). From perspective of mobile MapServer each mobile console consists of separate map and data services, each using...
separate communication module (communication module 1 – map #1 and data #1). Both services are served separately using data obtained from the local database.

Fig. 4. A block diagram illustrating the major functional modules of mobile MapServer and consoles

In Figure 4 following modules of mobile MapServer are distinguished: communication module 1 and 2, preprocessing and deduplication, database client and the mobile MapServer kernel. The communication module 1, provides reliable communication between MapServer and local consoles with simplified addressing. Using the send method offered by this module the mobile MapServer kernel can send messages to a specific service in an arbitrary selected console attached to the URC. Messages sent by consoles are accessed by the mobile MapServer kernel from a queue common to all the consoles and their services. The second communication module, the communication module 2, provides communication between the mobile MapServer and the central MapServer through an unreliable digital channel involving radio channel. This module implements a queue collecting send message requests with priorities. Received frames and reports of transmission errors and acknowledgments are passed to the mobile MapServer kernel’s queue using a callback function. Since frame losses and lower data rates are expected ([3], [11]) this module implements automatic retransmissions with possibility of withdrawal of transmission requests. Next module, the preprocessing and deduplication module, initially processes input data from devices providing position data about vessels and airplanes (AIS, ARPA and GPS) (preprocessing). After initial analysis and filtration of data received from external devices deduplication eliminates redundant objects’ data and integrates data from different sources when they represent the same object at the same position. Additional important task of the deduplication is the assignment of local identifiers (with guaranteed uniqueness in a mobile MapServer) to all detected objects and when it is possible global identifiers (with guaranteed uniqueness in all MapServers – mobile and central). Subsequent module, the database client, offers a common programing interface to all MapServer modules that require access to the local database to read or write data. The mobile MapServer kernel module implements a framework of MapServer’s information flow control. This module controls transfers of requests and data between other MapServer modules. Fulfillment of this task involves redirection of collected data to the database client, processing of consoles’ queries, redirection of queries addressed to the central MapServer and management of synchronization of local (URC) and central database (CENTER). This module is described in more detail in the subsection.

The second variant of the MapServer, the central MapServer (Figure 5), handles several consoles connected directly through IP network (stationary consoles) and additionally provides data on request to all mobile consoles connected indirectly through digital radio channel and IP network. As we can see in Figure 5, the central MapServer has a similar set of main modules to the mobile MapServer. Since, for the MapServer the stationary consoles are functionally identical with mobile consoles, the communication modules are identical in both variants of the MapServer. The preprocessing and deduplication functionality is similar, but the different external sources (web services) require different approach especially, at the preprocessing stage. Additionally, all identifiers assigned by this module in central MapServer are global. The database client has the same interface with some of the required functions different. Since the central MapServer doesn’t need to transfer console queries to the center, the central MapServer kernel doesn’t need such functionality. Instead, this module needs to process requests from mobile consoles. Of course, data received from local databases (MapServers in URCs of mobile units) within synchronization needs to be integrated into the central database. Additionally global identifiers need to be assigned to objects when the mobile MapServer requests them. This variant of MapServer kernel module is also described in more detail in the subsection.

Fig. 5. A block diagram illustrating the major functional modules of the central MapServer

ARCHITECTURE OF THE MOBILE MAPSERVER KERNEL

In Figure 6 a block diagram of mobile MapServer with details of kernel implementation with all internal modules of the kernel distinguished. In this figure all mutual relations can be seen. The MapServer designated for URC is composed of following functional modules: the new data processing with internal cache, the queries queue, the local queries processing, the center queries processing, the deduplication 2 and the database synchronization 1. Functions and the internal information flow in the mobile MapServer is described in more detail in the next section.
ARCHITECTURE OF THE CENTRAL MAPSERVER KERNEL

The Figure 7 presents a block diagram of central MapServer with details of kernel implementation with all internal modules of the kernel distinguished. As in the case of the central MapServer kernel described in the previous section, in this figure all mutual relations can be seen. The central MapServer designated for CENTER is composed of following functional modules: new data processing with internal cache, queries queue, local queries processing, deduplication 3, database synchronization 2. Functions and the internal information flow in the central MapServer is described in more detail in the next section.

INFORMATION FLOW IN THE MOBILE MAPSERVER

Since, as it was mentioned before, mobile and stationary consoles are functionally identical, the functionalities of the central MapServer kernel related to the local consoles support are similar. Nevertheless, there are important changes in the MapServer kernel architecture. First of all, the central MapServer doesn’t need the center queries processing module since we already are in the center and have direct access to the central database. Another difference is that the queries queue module additionally manages the requests from the mobile MapServers received via the communication 2 module. Also database synchronization works differently since data from multiple mobile units needs to be integrated into the central database with unique global identifies assigned. Moreover, the global identifies have to be distributed to all units which have detected the same object using their own equipment.

THE INFORMATION FLOW IN THE MAPSERVER

Since the MapServer must provide diverse functionalities, the internal information flow is complex and a need for the information flow management arises. There are five main functionalities of the MapServer:

1. collecting objects data from external sources;
2. providing current or archival data from local sources for maps and tables;
3. retrieving current or archival data from central database for maps and tables;
4. assigning local and global identifiers to all objects;
5. databases synchronization.

The above mentioned functionalities determine internal structure of the MapServer which is responsible for controlling the flow of information in the mobile and central MapServer. The details of implementation of all functionalities of the MapServer in both, mobile and central, versions are briefly discussed in the following subsections on the basis of information flow diagrams.

INFORMATION FLOW IN THE MOBILE MAPSERVER

In this subsection the information flow in the mobile MapServer is discussed on the basis of the functional modules of the mobile MapServer kernel (Figure 6).

The processing of new data module processes new data received from the preprocessing and deduplication module and manages the cache for the basic set of current objects data needed for the most frequent request for the data of the current situation on requested part of the map (Figure 9). For each supported console there is a separate data cache. Apart from updating data in the caches the module passes the data to the database client so all the data obtained from local sources (Figure 8) and the central database (Figure 10) is stored in the local database. Another important function of the processing of new data module is management of the databases synchronization process (Figure 11). This requires that local information about new objects’ data obtained from the local sources is transferred to the database synchronization 1 module. This module integrates the new data with data stored in the local database based on the global identifier assigned by the center or the preprocessing and reduplication module. The new data which are obtained from the central database is transferred to the reduplication 2 module and via the database client stored in the local database.
The data requests from mobile consoles connected locally to the URC are processed by three modules: the center queries processing module (Figure 10) and the queries queue module, the local queries processing module (Figure 9). Firstly, the queries queue module analyzes messages received from the communication 1 module and separates them between the local queries processing and the center queries processing modules on the basis of the FromCenter flag. Next, the actual processing is performed in one of queries processing modules, local or center. These modules manage two separate processing queues so the longer processing time of the data requests sent to the center would not throttle the processing of local queries. This guarantees that only these console services that issued the queries with FromCenter flag would have to wait for the response form the center.

The center queries processing module attempts to retrieve requested data from the center via the communication 2 module. Additionally, if data are not already available in the local database the retrieved data are stored into the local database (Figure 10). Next, the request is the data retrieved from the center is formed into a message which is transferred to the appropriate console via the communication 1 module. Afterwards, if data from the central database falls into regions of local caches then the new data processing module is notified and if needed selected caches are reinitialized based upon the updated local database. Finally, either the timeout message or the message with data received from the central database is send to the appropriate console via the communication 1 module.

The local queries processing module retrieves data from the local database or in case of the request of current data for presentation on the map from the cache of the new data processing module (Figure 9a). If the cached region changes then the cache is reinitialized from the local database (Figure 9a). Additionally, if the console operator adds the note, apart from adding the note into the local database the new data processing module passes it to the database synchronization 1 module (Figure 9c). Finally, the prepared data is formed into a message which is transferred to the appropriate console via the communication 1 module.
The last functional module of the mobile MapServer kernel is the databases synchronization 1 (Figure 11). This module collects information received by the MapServer from local sources and the consoles operators notes that has to be send to the center during cyclic synchronization sessions (Figure 8 and 9c). Apart, from sending data to the central database, the database synchronization 1 module writes into the local database global identifiers assigned by the center.

**INFORMATION FLOW IN THE CENTRAL MAPSERVER**

Like in the case of the mobile Mapserver, the information flow in the central MapServer is discussed on the basis of the functional modules of its kernel (Figure 7).

The processing of new data module in the center processes new data received from the preprocessing and deduplication module and manages the cache (Figure 12) for the basic set of current objects data needed for the most frequent request for the data of the current situation on requested part of the map (Figure 13). For each supported stationary console there is a separate data cache. Apart from updating data in the caches the module passes the data to the database client so all the data obtained from local sources is stored locally in the central database. This module also takes part in the databases synchronization (Figure 15) but in the center it simply integrates the new data with data stored in its cache.

In the center the queries queue module collects queries from multiple stationary (Figure 13) and mobile consoles (Figure 14) from an appropriate communication module with all queries processed by the local queries processing module which retrieves data from the central database via the database client module (Figure 13a and Figure 14) or from the cache of the new data processing module which might be updated based on current data stored in the central database (Figure 13b). It is worth noting that the cache in the new data processing module do not support the mobile consoles since the data access from such console is intermittent. There is also no need for special processing of notes added by the stationary consoles’ operators since all such notes are stored directly in the central database from where mobile consoles’ operators can retrieve them on demand. Just like in the mobile MapServer kernel the prepared data is formed into a message which is transferred to the appropriate console via a communication module. The difference is that in the central MapServer prepared response message can be send either via the communication 1 or communication 2 module depending on whether the destination console is mobile or stationary.

The databases synchronization 2 (Figure 15) passes new data from mobile MapServers the deduplication 3 module which integrates them into the central database assigning global identifiers to all registered objects. Since area covered by mobile units’ equipment can overlap the deduplication module has to identify and discard duplicates. If new data stored in the database replace the most recent one, then the deduplication 3 module notifies the new data processing module so it can ensure consistency of data in the cache with the data in the central database. Additionally, if the message received via the communication 2 module contains data related to the object that is encountered first time, the database synchronization 2 module can request additional data from the mobile MapServer and store save them to the central database via the database client module.
CONCLUSIONS

The paper presents architecture and information flow of the MapServer software which manages data collected by Polish Border Guard units. With developed system the mobile units automatically transfer all data about detected objects and operator’s notes to the central database. Moreover, apart from providing locally collected data to the operator’s console, the developed solution allows for data retrieval from the central database. The system has been tested in laboratories of the Faculty of Electronics, Telecommunications and Informatics of Gdańsk University of Technology on hardware manufactured by the DGT which partici-pates in the research project.

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A METHOD FOR THE ASSESSING OF RELIABILITY CHARACTERISTICS RELEVANT TO AN ASSUMED POSITION-FIXING ACCURACY IN NAVIGATIONAL POSITIONING SYSTEMS

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ABSTRACT

This paper presents a method which makes it possible to determine reliability characteristics of navigational positioning systems, relevant to an assumed value of permissible error in position fixing. The method allows to calculate: availability, reliability as well as operation continuity of position fixing system for an assumed, determined on the basis of formal requirements - both worldwide and national, position-fixing accuracy. The proposed mathematical model allows to satisfy, by any navigational positioning system, not only requirements as to position-fixing accuracy of a given navigational application (for air, sea or land traffic) but also the remaining characteristics associated with technical serviceability of a system.

Essence of the method in question consists in the working-out of recorded empirical position-fixing data as well as the making use of multi-state Markov processes (appropriate to a maximum error value permissible for various navigational applications) as a result of which reliability characteristics based on real data would be determined. About usefulness of a given navigational positioning system for its possible application would decide a vector of variables (both dealing with position and reliability) which satisfies / or does not satisfy formal navigational requirements for a given application.

Keywords: reliability, accuracy, availability, navigational positioning systems

FORMAL REQUIREMENTS FOR NAVIGATIONAL POSITIONING SYSTEMS

The working-out radio-navigational plans, resolutions and recommendations, by international navigation-orientated institutions, is conductive for forming legal regulations aimed at ensuring a balanced and harmonized development of national, regional and worldwide radio-navigational systems. Their range of application deals with position-fixing requirements for all kinds of navigation [1, 2, 3, 5, 6] or is limited to one of its kinds: maritime [4], airborne [9] or even to a single application, e.g. hydrographical one [8]. Their scope covers:

- description of current functioning state of radio-navigational systems as well as time services available in a given country and neighbouring regions, together with a short technical description and services,
- plans concerning extending, development, closing or modernization of radio-navigational systems with taking into account research on their development carried out domestically and abroad,
- determination of national requirements for all types, phases and kinds of navigation as well as definition of institutional responsibility for radio-navigational and positioning service. The requirements constitute a set of operational
characteristics of the systems together with minimum numerical values assigned to them.

It should be stressed that only a few countries have undertaken to describe the matters in question in the form of uniform national official regulations: USA [6], Sweden [5], UK [3], covering all the forms of navigation. However the remaining ones introduce such requirements in practice in the form of national documents of a lower rank. Out of all legal solutions, the approach of USA should be especially distinguished, where Federal Radio-navigational Plan, very important legal document concerning the requirements for process of navigation, and which describes also the process of development and evolution of systems, is regularly published.

In the above mentioned documents the requirements for navigation process concern all kinds of navigation (maritime, aircraft and land) with taking into account their phases. An example of the requirements for maritime navigation process during harbour entrance and approach phase is presented in Tab. 1.

As results from the above mentioned normalizing documents, in order to safely perform process of navigation (maritime, aircraft or land) one should have at his disposal a positioning system which meets all the above specified navigational characteristics. Hence a question of a fundamental character arises: which way to assess whether a given positioning system is capable of meeting the above specified requirements. An additional difficulty in the assessing is continuous improving the positioning systems, which results in changes (elevating) navigational characteristics of majority of contemporary positioning systems. Let’s consider for instance GPS system which, for a dozen or so of last years, has permanently increased position-fixing accuracy [14]. The accuracy with the use of GPS system (in horizontal plane) was equal to: in 1993 - 100 m (2drms) [10, 11], in 2001 - 13 m (2drms) [12], and in 2008 – as much only as 9 m (2drms) [13]. As a result of this fact, number of applications of the system increases and increases. For instance, as early as in 1993 the system in question did not meet position-fixing requirements for car navigation due to its too low accuracy (100m), but already in 2001 it could be used satisfactorily in the above mentioned kind of navigational application (10m). The below attached figure presents a synthesis of the requirements for transport applications (maritime, aircraft and land) in function of position-fixing accuracy (x-axis) as well as availability (y-axis), based on an analysis of the data contained in the publications [2, 3, 5, 6]. The transport applications are presented in the form of icons. The grey-marked applications are those in which GPS system met their requirements in the years 1994÷2001. Blue colour shows the possibility of GPS in the years 2001÷2008, and green colour relates to the years from 2008 till now.

The analysis shown in Fig. 1 proves that number of applications of GPS system has grown year in year out and that it results from values, currently attained by the system, of particular characteristics (accuracy, availability, reliability, continuity and credibility) which are published by U.S. Defence Department in the official form only every a few years. The last standard was published in 2008 [13].

In this paper is proposed a method which makes it possible to assess possibility of using a positioning system in an arbitrary navigational application from the point of view of requirements dealing with its accuracy, availability, reliability and continuity. It consists in working out empirical position-fixing data achieved from an operating system as well as using multi-state Markov processes (relevant to maximum value of permissible position-fixing error in different navigational applications); as a result of this, reliability characteristics based on real data would be determined. About usefulness of a given navigational positioning system for its possible application would decide a vector of variables (both dealing with position and reliability) which satisfies (or does not satisfy) formal navigational requirements for a given application.

Fig. 1. Transport applications of GPS system during the years: 1993–2008, based on the requirements contained in [1, 2, 3, 5, 6].

MODEL OF POSITION-FIXING PROCESS EXECUTED BY NAVIGATIONAL SYSTEM AS A MARKOV PROCESS

Let’s consider the position-fixing process executed by positioning system, whose error in determination of coordinates in function of time is presented in Fig. 2. It may be assumed that it is the random process of reliability state changing \( \{W(t): t \geq 0\} \) constituted by the consecutive states \( s_i \) – corresponding to accuracy requirements of the three navigational applications: Highway Vehicle Identification, Inland Waterway for all ships and Resource Exploration close to Harbour Approaches. The states belong to the set of the distinguished subsets of classes – states, \( S = \{s_i; i = 1, 2, \ldots, j\} \).

Fig. 2. Value of position-fixing error in function of time – as a multi-state random process (of 3 states)

Acceptance of the following hypothesis to be right: „process of reliability state changing of positioning system is that whose state considered in an arbitrary instant \( t_n \) \((n = 0, 1, \ldots, m; t_0 < t_1 < \ldots < t_m) \) depends on the state directly preceding it and does not stochastically depend on the states which occurred earlier and their occurrence times” makes it possible to develop an adequate reliability model by using the theory of Markov (or semi-Markov) processes [16]. Such model may be hence a Markov process of a discrete set of distinguished states and continuous time of their duration. In the light of the above discussed issues, in the set of possible reliability states of positioning system, \( S \), the following subsets (classes) of states, called further the states, may be distinguished – from the point of view of its usability for realization of objective function (i.e. capability of securing a given navigational application) – depending on position-fixing accuracy:

- Subset of states \( S_1 \) – state which meets requirements of the application No. 1 for which value of the position-fixing accuracy is: \( \varepsilon < 1 \) m
- Subset of states \( S_2 \) – state which meets requirements of the application No. 2 and does not meet requirements of the application No.1, and for which value of the position-fixing accuracy is in the range of \( 1 \) m < \( \varepsilon < 2 \) m
- Subset of states \( S_3 \) – state which meets requirements of the application No. 3 and does not meet requirements of the applications No.1 and 2, and for which value of the position-fixing accuracy is in the range of \( 2 \) m < \( \varepsilon < 3 \) m.

Taking into account the above introduced division of the set \( S \) one can present a graph of states – transitions of the process \( \{W(t): t \geq 0\} \), as follows [Fig. 3]:

Fig. 3. Graph of states – transitions of the process \( \{W(t): t \geq 0\} \) between the states: \( s_1, s_2, s_3 \); \( \lambda_{ik} \) – intensity of transition from the state \( s_i \) under condition of transition of the process to the state \( s_k \) \((i, k = 1, 2, 3; i \neq k)\); \( T_i \) – random variable which describes time of staying in the state \( s_i \) irrespective of which state will be the next.

Initial distribution of the process \( \{W(t): t \geq 0\} \), under condition that in the initial instant \( t = 0 \) the system is in the state \( s_1 \), can be described as follows:

\[
p_1 = P(W(0) = s_1) = 1, \quad p_i = P(W(0) = s_i) = 0 \quad \text{for} \ i = 2, 3.
\] [1]

To determine a distribution of a considered process it is first necessary to estimate values of the transition intensities \( \lambda_{ik} \) \((i, k = 1, 2, 3; i \neq k)\) of the following interpretation:

\[
\lambda_{ik}(t) = \lim_{\tau \to 0} \frac{P(W(t+\tau) = s_k | W(t) = s_i)}{\tau}.
\] [2]

In practice, a convenient and credible estimation of the
magnitude may be the following which is determined on the basis of investigation of realizations of the random variables $T_{ik}[17]$: 

$$
\hat{\lambda}_{ik} = \frac{1}{E(T_{ik})} \geq \frac{1}{T_{ik}}, \quad (i, k = 1, 2, 3; i \neq k) \tag{3}
$$

where:

$E(T_{ik})$ – expected value of random variable which describes duration time of the state $s_i$ under condition that the state $s_k$ will be the next;

$T_{ik}$ – mean time of staying the system (engine) in the state $s_i$ under condition that the state $s_k$ will be the next.

Determination of probabilities of staying the system in particular states is possible by solving the set of Kolmogorov equations which can be generally presented in the following form, [18]:

$$
\frac{dP_i(t)}{dt} = \left( - \sum_{k \neq i} \hat{\lambda}_{ik} \right) P_i(t) + \sum_{k \neq i} \left( \hat{\lambda}_{ik} \cdot P_k(t) \right) \quad i \neq k. \tag{4}
$$

With taking into account the graph of changes of states of the process $\{W(t) : t \geq 0\}$ and its initial distribution for the considered case, the set may be expressed as follows:

$$
\frac{dP_i(t)}{dt} = - (\lambda_{1i} + \lambda_{2i}) \cdot P_i(t) + \lambda_{21} \cdot P_1(t) + \lambda_{31} \cdot P_1(t),
$$

$$
\frac{dP_i(t)}{dt} = - (\lambda_{2i} + \lambda_{3i}) \cdot P_i(t) + \lambda_{22} \cdot P_2(t) + \lambda_{32} \cdot P_2(t),
$$

$$
\frac{dP_i(t)}{dt} = - (\lambda_{3i} + \lambda_{2i}) \cdot P_i(t) + \lambda_{33} \cdot P_3(t) + \lambda_{23} \cdot P_2(t),
$$

Among other important reliability characteristics possible to be determined with the use of the theory of Markov (or semi-Markov) processes also the following may be numbered [16]:

- Limiting distribution of the process in question:

$$
p_i = \lim_{t \to \infty} P(W(t) = s_i); \tag{6}
$$

- Distribution of time of the first transition from the state $s_i$ to the distinguished subset of states $A : \Phi_{\alpha}(t)$;

- Approximate distribution of total staying time of the process in the state $s_i$ under condition that the state $s_i$ will be initial one;

- Expected value of the duration time $T_i$ of the process state $s_i$, regardless of to which state its transition takes place in the instant $\tau_{mi} : E(T_i)$;

- Variance of the time $T_i : D^2(T_i)$;

- Expected value of the duration time $T_i$ of the process state $s_i$ under condition that the state $s_i$ will be the next: $E(T_{ik})$;

- Mean number of “come-ins” (per unit of time) of the process to the state $s_i$ under condition that the state $s_i$ will be initial one: $\hat{\lambda}_{ik}(t)$.

### MODEL OF THE DETERMINING OF AVAILABILITY, RELIABILITY AND CONTINUITY OF MARKOV PROCESS STATES

Let’s define process of staying-in and failure of the state $S_i(t)$ in function of time as follows:

$$
S_i(t) = \begin{cases} 
1, & Z_{i_n} \leq t < Z_{i_{n+1}}; \\
0, & Z_{i_{n+1}} \leq t < Z_{i_{n+1}} 
\end{cases} \quad n, i = 0, 1, \ldots \tag{7}
$$

where: the variables $Z_{i_n}$ are instants of failures (changes from a given state to another), and the instants $Z_{i_{n+1}}$ are those of renewal of a considered state. Let $X_1, X_2, \ldots$ determine working time durations for a given state, and $Y_1, Y_2, \ldots$ correspond to instants of occurrence of failures (staying in other states). Let’s assume that the random variables $X_i, Y_i$ for $i = 1, 2, \ldots$ are independent on each other and that duration times of work and failure of a structure have the same distribution. Let’s assume that the cumulative distribution functions are continuous on the right, then:

$$
P(X_i \leq x) = F(x), \quad P(Y_i \leq y) = G(y) \quad i = 1, 2, \ldots \tag{8}
$$

where:

$F(x)$ - cumulative distribution function of work duration times of a navigational structure,

$G(y)$ - cumulative distribution function of failure duration times of a navigational structure.

Moreover, let’s assume that the variables have finite expected values and variances:

$$
E(X_i) = E(Y_i), \quad V(X_i) = \sigma_x^2, \quad V(Y_i) = \sigma_y^2.
$$

Let’s define availability of a given interval of position-fixing accuracy which corresponds to the state $S_i$, as a probability that a given position-fixing error is in the serviceability (working) state in an arbitrarily selected instant $t$, and that it may be expressed as follows:

$$
A_{s_i}(t) = P[S_i(t) = 1] \tag{9}
$$

Then the formula for availability of a given position-fixing interval takes finally the form [15]:

$$
A_{s_i}(t) = 1 - F(t) + \int_0^t \left[ 1 - F(t - x) \right] dH_{s_i}(x), \tag{10}
$$

where:

$$
H_{s_i}(x) = \sum_{n=1}^{x} \Phi_i(x), \tag{11}
$$

is a renewal function of the stream formed by renewal instants of the state $S_i$. The detailed derivation of the formula can be found in [15].
A functional form of $A_N$ is rather not very useful from the point of view of the estimating of availability of a given position-fixing accuracy interval, therefore in maritime navigation its limiting value is usually applied, that may be described as follows:

$$A_N = \lim_{t \to \infty} P[N(t) = 1] = \lim_{t \to \infty} A_N(t) = \lim_{t \to \infty} \left[1 - F(t) + \int_0^t [1 - F(t - x)]dH(x) \right]. \quad [12]$$

After its transformations, one can achieve the following final form for the availability:

$$A_N = \frac{E(X)}{E(X) + E(Y)}. \quad [13]$$

The notion of reliability of a given position-fixing accuracy interval may be defined in an analogue way, namely, $R_N$ is probability of staying the position-fixing error in a given time interval (staying in the state $[t, t + \tau]$), which has its final form as follows:

$$R_N(t, \tau) = 1 - F(t + \tau) + \int_0^t [1 - F(t + \tau - x)]dH(x), \quad [14]$$

and, its limiting value of the form:

$$\lim_{t \to \infty} R_N(t, \tau) = \frac{1}{E(X) + E(Y)} \int_0^\infty [1 - F(u)]du, \quad [15]$$

where: $x + \tau = u$.

The continuity of a given position-fixing accuracy interval and its limiting value take their respective forms as follows:

$$C_N(t, \tau) = \frac{1 - F(t + \tau) + \int_0^t [1 - F(t + \tau - x)]dH(x)}{1 - F(t) + \int_0^t [1 - F(t - x)]dH(x)} \quad [16]$$

and

$$\lim_{t \to \infty} C_N(t, \tau) = \frac{1}{E(X)} \int_0^\infty [1 - F(u)]du. \quad [17]$$

RESULTS OF A NUMERICAL EXPERIMENT

The developed model was applied to real position measurements taken by means of a typical maritime receiver of the differential system GPS (DGPS) operating in the frequency band of 283.5÷325 KHz. The analysis covered 25-day measurement session carried out with 1 Hz frequency. The total number of measurements was equal to 2210266 events of position fixing. The so large number of measurements made it possible to obtain a similar large number of states sufficient for conducting representative statistical reasoning. The real coordinates of the measurement point: $B = 54.317552$ N, $L = 18.335741$ E, $H = 68.07$ m were determined in WGS-84 system with employing the geodesic measurement techniques GNSS, with the accuracy of 1 cm (rms). Next, the geodesic coordinates measured by the measurement receiver DGPS were recalculated into the plane (Cartesian) coordinates by using Gauss-Kruger transformation. On the basis of the real coordinates as well as measurement results, errors of particular measurements (in meters) were determined for 2D coordinates – geographical latitude and longitude. Distribution of the coordinates versus real values is presented in Fig. 4.

[Figure 4: Distribution of the positions measured by means of DGPS receiver versus real coordinates]

The analysis was started from calculation of statistics of position-fixing errors with the use of typical navigational methods for estimating position accuracy, and consequently the following results were obtained: Latitude (rms) - 0.81 m, Longitude (rms) - 0.63 m, Altitude (rms) - 1.43 m, drms (2D) - 1.02 m, 2drms (2D) - 2.04 m, R35(2D) - 1.79 m, CEP (3D) - 0.83 m, R95 (3D) - 3.04 m. For further analyses was used a set of 2D position-fixing errors, in which three example classes of process states were determined depending on $H$ - value:

- $s_1$ – the position-fixing error $e \leq e_{\text{wp}} = 0.5117$,
- $s_2$ – the position-fixing error $0.5117 < e \leq e_{\text{wp}} = 1.0234$,
- $s_3$ – the position-fixing error $e > e_{\text{wp}} = 1.0234$

As a result of the processing of data obtained on the basis of real measurements of position, were obtained realizations of the random variables $T_{ik}$, $T_{ik}$ (i, k = 1, 2, 3; i ≠ k) as well as the following numbers of process transitions from the state $s_i$ to the state $s_j$:

- $n_{12} = 4870$;
- $n_{13} = 0$;
- $n_{21} = 4868$;
- $n_{23} = 4433$;
- $n_{31} = 2$;
- $n_{32} = 4431$;
The achieved statistics made it possible to modify the graph of states and transitions (Fig. 3) as well as the set of equations (5) into the following form:

\[
\begin{align*}
\frac{dP_1(t)}{dt} &= -(\lambda_1 + \lambda_2) \cdot P_1(t) + \lambda_1 \cdot P_3(t) + \lambda_2 \cdot P_2(t) \\
\frac{dP_2(t)}{dt} &= -(\lambda_2 + \lambda_3) \cdot P_2(t) + \lambda_2 \cdot P_1(t) + \lambda_3 \cdot P_3(t) \\
\frac{dP_3(t)}{dt} &= -(\lambda_1 + \lambda_3) \cdot P_3(t) + \lambda_1 \cdot P_1(t) + \lambda_3 \cdot P_2(t)
\end{align*}
\]

The empirical distributions of the random variables \(T_i\) and \(T_{ik}\) were used to verify statistical hypotheses as to their compliance with exponential distribution, and it was assumed that the values \(\lambda_{\hat{}}\) determined according to the relations (3) may serve as the estimators of the parameters of the distributions, \(\lambda_{\theta}\). Their values are presented in Tab. 2.

Tab. 2. Values of estimators of parameters of exponential distributions of the random variables \(T_i\) and \(T_{ik}\)

<table>
<thead>
<tr>
<th>Random variable</th>
<th>(\lambda_{\hat{}}) [s(^{-1})]</th>
<th>Random variable</th>
<th>(\lambda_{\hat{}}) [s(^{-1})]</th>
</tr>
</thead>
<tbody>
<tr>
<td>(T_1)</td>
<td>0,0095</td>
<td>(T_{11})</td>
<td>0,100</td>
</tr>
<tr>
<td>(T_2)</td>
<td>0,100</td>
<td>(T_{12})</td>
<td>0,100</td>
</tr>
<tr>
<td>(T_3)</td>
<td>0,0058</td>
<td>(T_{21})</td>
<td>0,0058</td>
</tr>
<tr>
<td>(T_{12})</td>
<td>0,0095</td>
<td>(T_{22})</td>
<td>0,0086</td>
</tr>
</tbody>
</table>

The verification was performed with the use of the goodness-of-fit test \(\chi^2\), and in order to select a true hypothesis \(H_0\), i.e. that for which is no basis to be rejected, the hypothesis selecting principle was assumed as follows:

- if \(g_0 \geq g_\alpha\) for \(\alpha = 0,05\) then \(H_0\) should be rejected;
- if \(g_0 < g_\alpha\) for \(\alpha = 0,10\) then \(H_0\) should be accepted;
- if \(g_0 < g_\alpha\) for \(\alpha = 0,05\) and for \(\alpha = 0,10\) \(g_0 \geq g_\alpha\), then goodness of \(H_0\) is dubious and the testing should be continued;

where:

- \(g_0\) – characteristic value of the test, determined on the basis of investigation results;
- \(g_\alpha\) – limiting value for the significance level \(\alpha\), i.e. value of \(\alpha\) - order quantile of a selected statistics.

The applied test made it possible to state that there is no basis to reject successive hypotheses \(H_0\) on the conformity of the considered variables \(T_i, T_{ik}\), with exponential distribution, therefore they were taken to be true.

By making use of Laplace transform, initial distribution of the process \([W(t) : t \geq 0]\) and values \(\lambda_{\hat{}}\) for solving the set of equations (18), the following set of linear equations was obtained in the domain of transformations:

\[
\begin{align*}
S \cdot P_1'(s) - 1 &= -0,0095 \cdot P_1'(s) + 0,1 \cdot P_2'(s) + 0,0086 \cdot P_3'(s) \\
S \cdot P_2'(s) &= 0,0095 \cdot P_1'(s) - 0,2 \cdot P_2'(s) + 0,0058 \cdot P_3'(s) \\
S \cdot P_3'(s) &= 0,1 \cdot P_2'(s) - 0,0144 \cdot P_3'(s)
\end{align*}
\]

and after putting in order:

\[
\begin{align*}
\{s \cdot P_1'(s) - s - 0,0095 \cdot P_1'(s) - s - 0,0086 \cdot P_3'(s) = 1\} \\
\{s \cdot P_1'(s) + (s + 0,2) \cdot P_2'(s) - 0,0058 \cdot P_3'(s) = 0\} \\
\{-0,0095 \cdot P_1'(s) + (s + 0,144) \cdot P_3'(s) = 0\}
\end{align*}
\]

The solving of the presented set of equations and the executing of Laplace inverse transformation made it possible to find the distribution of the temporary process \(p(t) = P(W(t) = s_j)\). A graphical representation of the solution is given in Fig. 6.

Assuming the exponential distribution of operation and failure times, where:

\[
\begin{align*}
A_{\exp}(t) &= \frac{\mu}{\lambda + \mu} + \frac{\lambda}{\lambda + \mu} e^{-(\lambda + \mu)t}.
\end{align*}
\]

with limiting value:
Positioning GNSS networking service reliability in the interval of time \([t, t + \tau]\) can be defined as the survival probability of a system where the position-fixing error reliability function can be presented as follows:

\[
R_{\text{exp}}(t, \tau) = \left[ \frac{\mu}{\lambda + \mu} + \frac{\lambda}{\lambda + \mu} e^{-(\lambda + \mu)\tau} \right] e^{-\lambda t},
\]  

with limiting value:

\[
\lim_{\tau \to \infty} R_{\text{exp}}(t, \tau) = \frac{\mu}{\lambda + \mu} e^{-\lambda t}
\]

And positioning service continuity could be represented by the formula:

\[
C_{\text{exp}}(t, \tau) = e^{-\lambda t},
\]  

with

\[
\lim_{\tau \to \infty} C_{\text{exp}}(t, \tau) = e^{-\lambda t},
\]

Results of calculation are presented in the form of availability and reliability functions (Fig. 7).

Complete results of limiting values of availability, reliability and continuity are given in Tab. 4.

**Tab. 4. Limiting values of availability, reliability and continuity for three selected states.**

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Level &lt;= 0.517</th>
<th>Level &gt; 0.517 and &lt;= 1.0234</th>
<th>Level &gt; 1.0234</th>
</tr>
</thead>
<tbody>
<tr>
<td>MTBF</td>
<td>105.62 s</td>
<td>100.10 s</td>
<td>172.76 s</td>
</tr>
<tr>
<td>Availability limiting value</td>
<td>23.25 %</td>
<td>42.10 %</td>
<td>34.64 %</td>
</tr>
<tr>
<td>Reliability limiting value</td>
<td>22.59 %</td>
<td>40.86 %</td>
<td>34.05 %</td>
</tr>
<tr>
<td>Continuity limiting value</td>
<td>97.19 %</td>
<td>97.04 %</td>
<td>98.27 %</td>
</tr>
</tbody>
</table>

**CONCLUSIONS**

In this paper has been presented a method which makes it possible to determine reliability characteristics of navigational positioning systems relevant to a given value of permissible error in position fixing. Essence of the method consists in working-out the recorded empirical positioning data as well as using multi-state Markov processes (adequate to maximum value of position-fixing error permissible for various navigational applications) - and in that the reliability characteristics based on real data can be consequently determined. The proposed mathematical model allows to assess if it is possible to satisfy, by a given navigational positioning system, not only requirements for position-fixing accuracy of a given navigational application. Additionally a numerical
example of application of the proposed method, based on recorded data of 2 million positions from EGNOS system, has been presented in this paper.

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POLAND
MAXIMUM LOAD CARRYING CAPACITY ESTIMATION OF THE SHIP AND OFFSHORE STRUCTURES BY PROGRESSIVE COLLAPSE APPROACH

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ABSTRACT

The aim of the paper is to represent step by step progressive collapse analysis for maximum load carrying capacity estimation of a hull girder by using variant of Smith Method, named HULT by authors, with different element separation including single plates, stiffeners, hard corners and stiffened panels.

The structural elements that form the ships and offshore structures are exposed to large vertical bending moments and especially compression or tension forces in the longitudinal axis in case of hogging and sagging under bad sea conditions. In recent years, it becomes very important and valuable to practically, fast and nearly correct estimation of the maximum vertical bending moment just before breaks in two (collapse) under the worst conditions. The optimum (accuracy, time, practicality) estimation of these values depend on how accurate the stress-strain relation of the structural elements are established. In this first part of study, the ultimate strength behaviour of the stiffened panels in decks, bottoms and sides is estimated by developed semi-analytical method with updated orthotropic panel calculation approach under uniaxial (only longitudinal axis) compression loads. The second part of calculation is focused on the progressive collapse analysis of hull girders under longitudinal uniaxial compression with Smith Method but with different element discretization in contrast to the conventional beam-column elements. Also some benchmark studies of such methods on ultimate limit state assessment of stiffened panels and nine benchmark hull girders of ships are conducted, using some candidate methods such as IACS Common Structural Rules (CSR), FEA with Ansys v13 and HULT prepared by authors. The results from the tests, FEM analysis and different computational approaches are compared to determine performance of the method.

Keywords: Ship and offshore structure, stiffened panel, orthotropic panel, ultimate strength, progressive collapse analysis, Smith method.

INTRODUCTION

The ship and offshore structures' hulls consist of stiffened panels such as bottom construction, side shell construction, upper deck construction, watertight bulkheads. Unlike most land based structures, ships and offshore structures operate in a dynamic and unstable sea environment. The large number of loads caused by sea conditions and own cargos are much less than the structural capacity of the ship's hull girder. However, these structures must not only be designed according to be capable of withstanding normal loads, but also with extreme sea condition scenarios.

Two types of structural design methods are relevant for ship and offshore structures, namely "Allowable Stress Design" and "Limit State Design". Methodology used in this paper focused on Ultimate Limit State Design. This design is a philosophy in which the "capacity" (ultimate strength) of a structure is evaluated directly and compared to the "demand" (extreme load) applied to the structure [1]. Limit state design methods can be applied to a hull structure by applying longitudinal bending to the hull girder. Loads produce a distribution of longitudinal bending moments and compression/tension loads. If the bending moment value exceeds the ultimate strength value of the hull girder, the ship can fail due to buckling and progressive collapse of the compressed part [2]. At this point, the ultimate strength subject matter that shows us maximum load carrying capacity of hull girder under bending moment has been considered very important in the academic area and classification societies. Since the loads due to rough sea conditions, unusual loading and unloading of cargoes during operations acting on the ship hull are uncertain, the hull girder may collapse like breaking in two or something more catastrophic losses under these uncertain loads.
Published researches about progressive collapse analysis of hull girder can be classified three categories such as: (1) derivation of theoretical methods to estimate progressive collapse or ultimate strength [2,4-12]; (2) results from theoretical modelling of sections using FEM approaches [13,14]; (3) reporting of physical experiments on box girders or ship structures [15].

One of the important methods to estimate the ultimate hull girder strength is idealized structural unit method (ISUM) developed by Ueda and Rashed [4]. For an ISUM type analysis the hull girder is usually divided into several different types of structural members such as support members (single stiffeners), beam-columns, rectangular plates and stiffened panels.

Another successful progressive collapse method is Smith Method [2]. This method is one of the simplified and most well recognized methods in the marine field to predict the global strength of a hull girder. In this paper, Smith Method principles are used for progressive collapse calculations of benchmark ship hull sections. This most useful hull girder ultimate strength evaluation method utilizes load-end shortening (stress-strain) curves describing the strength behaviour of structural elements including plates, stiffeners, stiffened plates and stiffened panels. For a progressive collapse type analysis the hull girder is usually discretized into plate-stiffener combination beam-column elements; each element comprises a single longitudinal stiffener with attached plating located between adjacent transverse frames. Failure of the hull girder in overall bending occurs by inter-frame failure of these elements.

According to the limit states based approach, the buckling collapse failure modes of a stiffened panel under compressive loads are: overall collapse after overall buckling of the plating and stiffeners as a unit, plate- induced failure by yielding at the corners of plating between stiffeners, plate-induced failure by yielding of stiffeners with attached plating at mid-span, stiffener induced failure by local buckling of stiffener web and stiffener-induced failure by lateral-torsional buckling or tripping of stiffeners [1]. It is known that the behaviour and collapse strength of stiffened panels under compressive loads depend on several factors as geometric and material properties, loading characteristics, boundary conditions and welding-induced initial imperfections (initial distortions in the plate and residual stresses). For local elements, earlier works by many researchers have studied about tripping of stiffeners, stiffeners local buckling and ultimate strength of stiffened plates/panels under in-plane and lateral loads theoretically, experimentally and numerically [16-20]. Smith [19] presents a series of tests and FEM analysis on full scale welded steel grillages subjected to a combination of axial compression and lateral pressure. Paik et al. [20] extended the early works and provide an extensive contribution to the ultimate strength evaluations for stiffened panels by developing practical methods and empirical formulas. Benson et al. [8] investigated aluminium stiffened panels behaviours under uniaxial compressive loads and developed a semi-analytical method by using FEA and orthotropic plate theory. For more realistic analysis and results, all failure modes have to be considered [1]. In this study, according to our solution methodology all five modes are considered separately and collapse of stiffened panels occurs at the lowest average among the various ultimate loads. The semi-analytical method described in this paper combines established empirical formulas of IACS-CSR [22] and large deflection orthotropic panel approach [20-21], for estimating the failure (buckling) modes of local single plate and stiffener elements, respectively with extended large deflection orthotropic method for estimating the global failure mode of stiffened panel [8].

METHODOLOGY

In this study, the steps presented below are followed respectively to obtain, firstly, the load-end shortening curve of discretized hull section elements and then the maximum load carrying capacity of hull girders.

At the beginning, the stiffener strength is evaluated using stress-strain curves derived from FEM analysis and IACS-CSR equations describing the load-end shortening curve flexural–torsional (tripping) buckling and web local buckling of stiffeners (Step 1). Besides, the unstiffened plate strength is evaluated using standard stress-strain curves derived by using Marguerre [17] governing large deflection nonlinear equations of initially deflected single plate theory are extended from von Karman’s [16] original equilibrium and compatibility equations (Step 2). Then, the plate-stiffener combination beam-column elements’ strength is evaluated combining the plate and stiffener strength obtained from Step 1 and Step 2, respectively and a comparison is conducted with plate-stiffener combination buckling strength evaluation using standard stress-strain curves derived from IACS-CSR equations (Step 3). Then, the overall panel strength between two adjacent frames is evaluated by large deflection orthotropic panel approach but using renewed instantaneous longitudinal geometric properties like $E_p$, $D_p$ and etc. determined by instantaneous tangent modulus $E_{tp}$ and $F_{tp}$ from the plate components’ and stiffener components’ load shortening curves, respectively (Step 4). Peak load values obtained from load-end shortening curves of each step mentioned above are compared to determine the lesser one and the collapse strength of the stiffened panel (Step 5). Finally, progressive collapse calculations have been carried out for estimation of hull girder ultimate strength using load-end shortening curves obtained from previous steps (Step 6).

DETERMINATION OF SINGLE STIFFENER COMPONENTS’ STRESS-STRAIN RELATION BEHIND AND BEYOND THE ULTIMATE STRENGTH (STEP 1)

Stiffeners’ failure strength values can be determined by empirical formulas. In this paper, three different stress-strain curves such as single stiffener web buckling, single stiffener
lateral torsional buckling and plate-stiffener combination beam-column buckling type failure behaviours are obtained by using IACS-CSR (2012), Appendix-A, Chapter 2.3. The results have been validated by non-linear FEM analysis with Ansys v13.

The peak load value of each curve is assumed as the ultimate strength for single stiffener. The curve that gives the minimum ultimate strength among three curves is used in extended orthotropic panel calculation process for Step 3 and Step 5 as $P_{stf}$.

**DETERMINATION OF SINGLE PLATE ELEMENT'S STRESS-STRAIN RELATION BEHIND AND BEYOND THE ULTIMATE STRENGTH (STEP 2)**

Large deflection plate theory used in this study allows examining the behaviour of initially deflected single rectangular plates under in-plane loads and lateral pressure. Foundations of the theory have been taken by von Karman to examine the behaviour of the initially flat plates. Marguerre, based on this equation, consider the plate with $z = a (x,y)$ form initial out of plane deflection. Summation of the out of plane deflection in mid-plane under load ($w$) and the initial out of plane deflection ($z$) make the total plate out of plane deflection ($w+z$) at failure.

The governing nonlinear equations of large deflection plate theory extended by Marguerre from von Karman's original equilibrium and compatibility equations, respectively, obtained after algebraic manipulations for initially deflected single plates are as follows:

\[
\nabla^2 F + \left[ (1 + \frac{1}{t^2}) \nabla^2 w_x \nabla^2 w_y + 2 \nabla^2 w_x \nabla^2 w_y - \nabla^2 w_x \nabla^2 w_y - \nabla^2 w_x \nabla^2 w_y \right] = 0
\]

\[
(1)
\]

\[
(2)
\]

Here,
- $F$: Airy stress function
- $t$: Plate thickness
- $D$: Plate flexural rigidity
- $E$: Young's modulus
- $Z$: Lateral load
- $m$: Number of half-sine wave at buckling mode shape

According to the membrane stress distribution towards both the length and width of a single plate, there are three possible regions for beginning of plastic yielding: midpoint of unloaded edges, midpoint of loaded edges and each of four corner points [21]. In this approach, the three possible plasticities ($\sigma_{vm-1}$, $\sigma_{vm-2}$, $\sigma_{vm-3}$) is checked by von-Mises equivalent yield criterion of plate material. The load is increased step by step and it is assumed that the plate will collapse if the von-Mises equivalent yield criterion values reach the material yielding stress. The ultimate load caused this yielding stress is assumed as the ultimate collapse strength of plate.

\[
\sigma_{e=vm-2} = \sqrt{\sigma_{e=vm-1}^2 + \sigma_{e=vm-2}^2 + \sigma_{e=vm-3}^2} = \sigma_0 \cdot \text{loaded edge mid-width} [21]
\]

(5)

The membrane stresses, than, can be calculated by equations (3) and (4) obtained using the Airy stress function, load functions and determined deflection value ($w_0$) with assuming the unloaded edges kept straight.

\[
\sigma_x = \frac{1}{t} \cdot \frac{\partial^2 F}{\partial y^2} = \left( \sigma_{s_{av}} + \sigma_t \right) + \left[ -\frac{E \cdot W_0 \cdot (x^2 + y^2)}{2} \cdot \frac{\lambda^2 \cdot \cos 2\beta y}{\lambda^2 \cdot \cos 2\lambda x} \right]
\]

\[
(3)
\]

\[
\sigma_y = \frac{1}{t} \cdot \frac{\partial^2 F}{\partial x^2} = \left( \sigma_{s_{av}} + \sigma_t \right) + \left[ -\frac{E \cdot W_0 \cdot (x^2 + y^2)}{2} \cdot \frac{\lambda^2 \cdot \cos 2\beta y}{\lambda^2 \cdot \cos 2\lambda x} \right]
\]

\[
(4)
\]

Here;
- $\sigma_{s_{av}}$: Averaged applied stress
- $\beta$: $\pi/b$
- $b$: Plate width
- $\sigma_t$: Residual stress
- $\lambda$: $\pi/L$
- $L$: Plate length
- $m$: Number of half-sine wave at buckling mode shape

**DETERMINATION OF INTER-FRAME LOCAL PANEL STRENGTH (STEP 3)**

Plate and stiffener elements' resistances are estimated using the properties of the individual load shortening curves defined in the previous two steps. Two separate calculations are used to determine inter-frame local panel resistance at a given end shortening.

1. The resistance of the plate and stiffener as defined by the individual load shortening curves are merged to determine a combination resistance, $P_{p-stf}$.

2. The combination resistance is then compared to the plate-stiffener combination buckling strength, $P_{b-c}$ obtained using standard stress-strain curves derived from IACS-CSR equations.

The instantaneous beam-column element resistance (load value), $P_c$ is the lesser of these two calculated values. These steps are repeated in an incremental analysis to produce a complete plate-stiffer load shortening curve. Beam-column collapse is indicated if $P_c$ becomes less than $P_{plt}$ or $P_{stf}$. If $P_c$ remains greater than $P_{plt}$ or $P_{stf}$, failure is assumed to be by stiffener tripping or local plate failure depending on which component load shortening curve peaks first.
COMBINATION STRENGTH OF CORRESPONDING LOCAL PLATE AND STIFFENER ELEMENTS STRENGTH

Paik and Thayamballi (2003) gives a simple equation to estimate peak strength of a combined element ($P_{p-s}$) by combining the plate and stiffener strength using the proportional area of each component:

$$P_{p-s} = \frac{[P_{Pl}(b\times t)+P_{Stf}(bW\times t_{Pl})+bY\times t_{Pl}]}{(b\times t)+(bW\times t_{Pl})+bY\times t_{Pl}}$$

(6)

The combination strength includes plate buckling, stiffener web buckling and stiffener lateral torsional buckling, which are essentially local individual failures, but does not include beam column buckling where the plate and stiffener fail as one unit. Besides, this one unit buckling, $P_{p-s}$, is evaluated by standard stress-strain curves derived from IACS-CSR equations [22].

CALCULATION OF STIFFENED PANEL’S INSTANTANEOUS ULTIMATE STRENGTH WITH UPDATED LARGE DEFLECTION ORTHOTROPIC PANEL APPROACH BY USING STRESS-STRAIN CURVES OBTAINED IN STEP 1 AND STEP 2 (STEP 4).

As mentioned and emphasized earlier, the main resistant parts of ship and offshore structure hull girders are stiffened panels. The elastic overall panel buckling value can be determined using one formulae in classical orthotropic panel theory. Therefore, it is not possible to evaluate post-buckling and collapse behaviour. In this paper, a more realistic stress-strain relation of stiffened panel behind and beyond the buckling is obtained using renewed large deflection orthotropic panel theory. This approach depends on element discretization such as single stiffeners and single plates. To derive the stiffened panel load-end shortening curve, solution process is conducted by using these components’ stress-strain curves with iterative semi-analytical approach. In this method, the axial load is applied and controlled by increasing the axial end-shortening iteratively. In each iterative step, all single element stresses are compared with overall panel stress derived by updated orthotropic panel calculation procedure but using stress-strain curves derived from IACS-CSR equations. The minimum stress among them determines the stress-strain relation for that step. The important point here is, the instantaneous tangent modulus ($E_{p-s}$, $E_{n-s}$), updated at each step, of each single elements’ stress-strain curves are used to re-calculate the elastic constants $E_x$ and $E_y$ at that step for conducting orthotropic panel solution [8]. In post-collapse region, the stress-strain relation is evaluated similar to the single plates but using overall elastic buckling ($\sigma_{x, x, 0}$) of stiffened panel instead of elastic buckling of single plate.

APPLICATION OF INCREMENTAL ITERATIVE APPROACH TO THE STIFFENED PANEL FOR DETERMINATION OF THE STIFFENED PANEL’S STRESS-STRAIN RELATION BEHIND AND BEYOND THE ULTIMATE STRENGTH (STEP 5)

The results of four steps mentioned above are used for calculations in this step. Description of the procedure to easy understanding is presented below and calculation flowchart is given in Fig. 2.

1 - The single stiffener strength ($P_{stf}$) and single plate strength ($P_{plt}$) are evaluated from curve obtained in Step 1 and Step 2, respectively.
2 - The corresponding local combined plate-stiffener element strength, $P_{p-s}$, is calculated by combining $P_{stf}$ and $P_{plt}$.
3 - The plate-stiffener combination beam-column element strength, $P_{p-s}$, is evaluated from stress-strain curves derived from IACS-CSR equations.
4 - The inter-frame local panel strength, $P_c$ is the minimum of $P_{p-s}$ and $P_{n-c}$.
5 - The instantaneous overall orthotropic panel strength, $P_{n-p}$, is obtained using updated orthotropic panel calculations for stiffened panels.
6 - The instantaneous panel strength value, $P_{pnl}$ is chosen to be the minimum of $P_c$ and $P_{n-p}$. At each axial displacement increment, $P_{pnl}$ provides an additional point for the panel load-end shortening curve.
7 - Similar calculations 1 to 6 is repeated until ultimate strength is reached, then a simpler post collapse curve can be postulated by taking the initial gradient from the post collapse point and assuming a linear post collapse curve [8].

PROGRESSIVE COLLAPSE CALCULATIONS OF HULL GIRDER ULTIMATE STRENGTH (STEP 6)

The main considered methodology employed in this part is the Smith method [4]. This method is the best established, simplified and the most appropriate method for progressive collapse analysis. The moment-curvature relationship is calculated by forcing an incremental curvature around the neutral axis of the hull section. Details of method can be seen in published papers and books [1, 2, 8, 9, 10, 21]. The considered method presented in this study is composed from ISUM based element discretization and Smith based iterative collapse analysis including orthotropic panel calculations for stiffened panels updated in every load step. Within the method, stiffened panel elements’ load-end shortening (stress-strain) relations behind and beyond the ultimate strength are obtained by updated orthotropic panel calculation procedure but using stress-strain curves of several type single stiffeners and single plates. The sample of calculation table and calculation flowchart can be seen in Table 1 and Figure 3, respectively. Details of this procedure and obtaining the load-end shortening curves of other single elements can be find in studies of Benson [8] and Olmez [10].

Table 1. Progressive collapse analysis calculation table

<table>
<thead>
<tr>
<th>Step No</th>
<th>$t_1$</th>
<th>$t_2$</th>
<th>$t_3$</th>
<th>$t_4$</th>
<th>$t_5$</th>
<th>$t_6$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1,2</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>$\Delta t$</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>$t_2$</td>
<td>$t_3$</td>
<td>$t_4$</td>
<td>$t_5$</td>
<td>$t_6$</td>
<td>$t_7$</td>
</tr>
</tbody>
</table>

1 - Iteration number; 2 - element number

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Fig. 2. Stiffened panel load-end shortening curve derivation flow chart

Fig. 3. Progressive collapse calculation flowchart used in presented study
3. CONSIDERED SHIPS OF VALIDATION
BENCHMARK STUDY

The accuracy of strength predictions of ship hull girders is now examined in order to validating HULT’s ultimate strength procedures for ship and offshore structures. This is accomplished through evaluations of nine benchmark case studies for which detailed structural information and associated numerical or measured results have been reported in references [11, 23, 24].

CROSS SECTIONS AND STRUCTURAL
CHARACTERISTICS

In this part, the characteristics of progressive collapse behaviour of merchant ships under vertical sagging or hogging are investigated using the HULT code. The ten typical ship type designs represented with cross sections and main hull section properties have been studied. The cross sections of ten ships and main hull section properties are shown in Fig.4 and Table 2, respectively.

Fig. 4. Mid-ship cross sections of nine benchmark case ships

Table 2. Principal dimensions of the nine typical ship hull sections.

<table>
<thead>
<tr>
<th>Hull Sec. Prop.</th>
<th>FRG</th>
<th>SHOT</th>
<th>CNT35</th>
<th>DHOT1</th>
<th>DHOT2</th>
<th>SSBC</th>
<th>DSBC</th>
<th>CNT90</th>
<th>FPSO</th>
<th>DHOT3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length (m)</td>
<td>18</td>
<td>313</td>
<td>230</td>
<td>233</td>
<td>315</td>
<td>282</td>
<td>273</td>
<td>305</td>
<td>230.6</td>
<td>182.5</td>
</tr>
<tr>
<td>Breadth (m)</td>
<td>4.2</td>
<td>48.2</td>
<td>32.2</td>
<td>42</td>
<td>58</td>
<td>50</td>
<td>44.5</td>
<td>45.3</td>
<td>41.8</td>
<td>32.2</td>
</tr>
<tr>
<td>Depth (m)</td>
<td>2.8</td>
<td>25.2</td>
<td>21.5</td>
<td>21.3</td>
<td>30.3</td>
<td>26.7</td>
<td>23</td>
<td>27</td>
<td>22.9</td>
<td>18.1</td>
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<tr>
<td>Draft (m)</td>
<td>-</td>
<td>19</td>
<td>12.5</td>
<td>12.2</td>
<td>22</td>
<td>19.3</td>
<td>15</td>
<td>13.5</td>
<td>14.15</td>
<td>12.6</td>
</tr>
<tr>
<td>Cb</td>
<td>-</td>
<td>0.83</td>
<td>0.68</td>
<td>0.83</td>
<td>0.82</td>
<td>0.83</td>
<td>0.84</td>
<td>0.65</td>
<td>0.83</td>
<td>0.77</td>
</tr>
<tr>
<td>Sec. Area (m²)</td>
<td>-</td>
<td>7.858</td>
<td>3.84</td>
<td>5.31</td>
<td>9.64</td>
<td>5.65</td>
<td>5.79</td>
<td>6.19</td>
<td>4.88</td>
<td>3.03</td>
</tr>
<tr>
<td>N. Axis (m)</td>
<td>1.42</td>
<td>12.10</td>
<td>8.50</td>
<td>9.19</td>
<td>12.97</td>
<td>11.19</td>
<td>10.06</td>
<td>11.61</td>
<td>10.22</td>
<td>7.68</td>
</tr>
<tr>
<td>I (Vertical) (m⁴)</td>
<td>0.06</td>
<td>863.7</td>
<td>237.5</td>
<td>359.5</td>
<td>1346.1</td>
<td>694.3</td>
<td>508.3</td>
<td>682.8</td>
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<tr>
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<tr>
<td>σt (MPa)</td>
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<td>315</td>
<td>355</td>
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<td>315</td>
<td>315</td>
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</table>

Neutral Axis: Ship hull cross section neutral axis identifies a plane along which there is neither tension nor compression.
I (vertical): Moment of inertia of cross-sectional area about neutral axis.
Σ: Ship hull girder section modulus.
PROGRESSIVE HULL COLLAPSE ANALYSIS AND CALCULATED RESULTS

The ultimate vertical bending moment of the ten hull structures are studied using HULT and IACS-CSR (KTU) [10, 22], then the results are compared with published results [5, 7, 8, 9, 10, 11, 12, 23, 24]. It is noted that the hull structural dimensions applied for the all analysis were defined by including 50% corrosion margin (0.5 × t corr) values of individual structural components as specified by IACS-2006a causes obtained results as incomparable with all other results. Fig. 5 shows IACS-CSR and HULT models employed for the progressive hull collapse analysis under vertical bending. Hull cross-section model between two adjacent transverse frames at mid-ship is adopted as the extent of the analysis. For HULT code modelling, structural elements between support members are idealized such as single stiffeners, single plates, single stiffeners with attached plating, hard corners and identically stiffened panels.

Fig. 5. Element discretization of CNT35 and SHOT with HULT and IACS-CSR

The main assumptions used for ten hull girder progressive collapse calculations are:
1 - Analysis are carried out between two transverse frames.
2 - Plane sections remain plane after bending (Euler-Bernoulli Bending Theory).
3 - The neutral axis of the hull cross section changed as the collapse of individual structural components progressively occurs. Decrease or increase of the neutral axis position according to hogging or sagging is taken into account as the vertical bending moment is increasingly applied.
4 - Average level welding residual stresses (0.15 × σ base) is considered [21].
The result comparison of hogging and sagging condition ultimate strength calculations with HULT, IACS-CSR and published other results obtained by different authors using various methods are comparatively represented in Table 3 and Fig. 6, respectively.

<table>
<thead>
<tr>
<th>Remarks</th>
<th>FRG</th>
<th>SHOT</th>
<th>CNT3S</th>
<th>DHOT1</th>
<th>DHOT2</th>
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<td>1. Chen*[46]</td>
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<td>20.01</td>
<td>19.00</td>
<td>8.07</td>
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<tr>
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<td>19.03</td>
<td>16.84</td>
<td>6.72</td>
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<tr>
<td>7. ALPS/HULT*[19]</td>
<td>10.45</td>
<td>9.84</td>
<td>16.77</td>
<td>15.83</td>
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<tr>
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<td>9.53</td>
<td>17.66</td>
<td>16.95</td>
<td>7.34</td>
</tr>
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</table>

Mean / All Methods: 11.86 ± 9.784 ± 19.04 ± 17.91 ± 7.21 ± 6.74 ± 8.62 ± 6.98 ± 28.01 ± 22.62

Table 3 Cont. Summary of benchmark ship’s ultimate bending moments for all methods

<table>
<thead>
<tr>
<th>Remarks</th>
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<th>DSBC</th>
<th>CNT90</th>
<th>FPSO</th>
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<td>2. Cho*[46]</td>
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<td>13.69</td>
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<td>3. Masaoaka*[46]</td>
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<td>16.02</td>
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<td>4. Rigo[1]*[46]</td>
<td>18.71</td>
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<td>-</td>
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<tr>
<td>5. Rigo[2]*[46]</td>
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<td>14.34</td>
<td>-</td>
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<tr>
<td>6. Yao*[46]</td>
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<tr>
<td>8. Park-FEM[47]</td>
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<tr>
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<td>14.8</td>
<td>-</td>
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<td>10. Benson*[25]</td>
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<td>14.8</td>
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<td>13. Ogne*[23]</td>
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<td>14. Tayyar*[24]</td>
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<td>17. CSR-CR*[47]</td>
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<td>17.45</td>
<td>15.12</td>
<td>11.76</td>
<td>12.69</td>
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</table>

Mean / All Methods: 17.89 ± 15.01 ± 12.02 ± 12.25 ± 13.77 ± 16.51 ± 9.202 ± 7.491

Table 3 Cont. Summary of benchmark ship’s ultimate bending moments for all methods
Results obtained by methods 1 to 6, 10, 14, 15 and 18 to 22 were considered for calculation of Mean, St. Dv. and COV values of Smith Based (different formulations for obtaining the load-end shortening curves) calculations. Coefficient of variation (COV) calculated for all methods and Smith based methods are given in Table 3. COV for all methods varies from 0,013 to 0,107 and COV for Smith based methods varies from 0,013 to 0,115. Average COV calculated for considered groups are 0,053 and 0,049, respectively. This decreasing on average COV shows that similarity of methods increases as expected. It can also be noted that closed results are obtained by various Smith based methods (1 to 6, 10, 14, 15, 20 and 22) and the IACS-CSR method implementations use same formulations for load-end shortening curves (18, 19, 21 and 22). Although standard formulations for standard element idealizations are used, there are small differences among CSR results due to researcher factors (code algorithm, structural discretization, assumptions, etc.).

Fig. 6. Moment-curvature curves of benchmark models
If we zoom in one another moment-curvature curve of benchmark case ships, for example DHOT3 given in Table 2, we can clearly see and discuss which element(s) in which part of the hull section will collapse when. This is the main advantage of progressive collapse calculation approach. After progressive collapse analysis of DHOT3, it can be seen also in Fig. 7, the main deck longitudinals and the center line bulkhead upper longitudinals are collapsed firstly. After these elements, inner bulkhead upper longitudinals and side upper longitudinals are collapse. Finally, the side longitudinal elements around half the height of cross section are collapsed. Collapsed structural elements on the moment-curvature curve are presented in Fig. 7.

Fig. 7. Collapse sequence on moment-curvature curves of DHOT3 obtained by HULT

CONCLUSION

In this paper, first part (element stress-strain curve obtaining module) and second part (hull progressive collapse calculations) of the systematic calculation for hull girder ultimate strength analysis developed by authors, namely HULT is represented. The main target of the present study has been to bring out the reliability and applicability of stress-strain relations of stiffened panel elements to the hull girder collapse analysis and also bring out the reliability and applicability of progressive hull girder collapse calculations by HULT. Its reliability and applicability is tested by benchmark analysis for ten ships.

According to the results of verification case studies, as a main consequence, developed calculation flow including stress-strain curves for single plate, stiffener and stiffened panel can be reliably merged to progressive hull girder collapse analysis in terms of the resulting approximation. One of the other consequences should be highlighted is determination of the active role of orthogonal components’ sizing and material stress-strain relation on the global failure of a stiffened panel. Considering these active effects, taking into consideration all possible types of failure provide a more realistic and more understandable calculation of stiffened panel ultimate strength.

For all benchmark ships detailed analysis of collapse sequence for both hogging and sagging conditions are performed. As expected, obtained ultimate strength (maximum load carrying capacity) values are higher for hogging than sagging for all benchmark ships except DSBC and CNT90. The collapse of the compression flange of the tanker hulls takes place prior to the yielding of the tension flange as expected from usual ship hull girders. Thus, the ultimate hogging moment of the tanker hull is higher than the ultimate sagging moment. For all that, because bulk carriers have large deck openings, the section modulus at bottom is much larger than that at deck.

Similarly, because the deck panels of DSBC are typically much sturdier than bottom panels, in contrast to the tanker hulls, the bottom plates of the DSBC hull yields prior to buckling collapse of the deck plates under sagging condition. Under hogging condition, however, buckling collapse of the bottom plates takes place prior to yielding of the deck plates due to same reason. This result is contrary to what is normally expected ultimate strength characteristics of usual ship hull girders since the ultimate sagging moment of DSBC is higher than the ultimate hogging moment. For all that, because bulk carriers have large deck openings, the section modulus at bottom is much larger than that at deck.

Last of all, according to the results of verification case studies, it can be observed that calculations with HULT and implementation of IACS-CSR method used by authors are closely compatible with overall mean values for all benchmark ship hull girder models. Also, developed calculation flow including stress-strain curves for single plate, stiffener and stiffened panel can be reliably merged to progressive hull girder collapse analysis in terms of the resulting approximation. It should be also underlined that there will be less elements in the calculation table due to the selected structural element idealization and discretization. In this way, fewer load-axial end shortening curves will be needed and this will decrease the computation time. Hereby, HULT has adequate reliability to estimate hull girder ultimate bending moment and determining the collapse sequence of structural elements.

BIBLIOGRAPHY


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CFD MODELING OF SYNGAS COMBUSTION AND EMISSIONS FOR MARINE GAS TURBINE APPLICATIONS

Nader R. Ammar
Ahmed I. Farag
Department of Naval Architecture & Marine Engineering,
Alexandria University, Egypt

ABSTRACT

Strong restrictions on emissions from marine power plants will probably be adopted in the near future. One of the measures which can be considered to reduce exhaust gases emissions is the use of alternative fuels. Synthesis gases are considered competitive renewable gaseous fuels which can be used in marine gas turbines for both propulsion and electric power generation on ships. The paper analyses combustion and emission characteristics of syngas fuel in marine gas turbines. Syngas fuel is burned in a gas turbine can combustor. The gas turbine can combustor with swirl is designed to burn the fuel efficiently and reduce the emissions. The analysis is performed numerically using the computational fluid dynamics code ANSYS FLUENT. Different operating conditions are considered within the numerical runs. The obtained numerical results are compared with experimental data and satisfactory agreement is obtained. The effect of syngas fuel composition and the swirl number values on temperature contours, and exhaust gas species concentrations are presented in this paper. The results show an increase of peak flame temperature for the syngas compared to natural gas fuel combustion at the same operating conditions while the NO emission becomes lower. In addition, lower CO2 emissions and increased CO emissions at the combustor exit are obtained for the syngas, compared to the natural gas fuel.

Keywords: CFD, Syngas fuel, Combustion characteristics, Exhaust gas emissions.

INTRODUCTION

Emissions from the marine transport sector contribute significantly to air pollution globally [1, 2, and 3]. Around 15% of global nitrogen oxides (NOx), 2.7% of global CO2 emissions, and 5–8% of global sulfur oxides (SOx) emissions are produced by oceangoing ships [3, 4, 5, and 6, 7, 8]. As a result of maritime transport emissions, the International Maritime Organization (IMO) has implemented mandatory regulations to reduce these exhaust gas emissions. On the one hand, NOx emissions fall under IMO Regulation 13 and a number of methods can be used to meet the emission limits [9]. On the other hand, IMO Regulation 14 dictates the emission limits for SOX and carbon particles from ships. In addition, IMO marine environmental protection committee requires increasing the energy efficiency of new ships to reduce CO2 emissions from ships [10]. These environmental legislations on emissions from ships can be met by either refining the fuel with the aid of one or more systems of exhaust gas cleaning to reduce sulfur content, or by using an alternative clean fuel [10, 11]. Synthesis gas fuel is one of challenging renewable fuels for ships to satisfy increasingly strict emissions regulations. It can be considered a greener alternative for internal combustion engines [12, 13, and 14]. In addition, it is the transition fuel from the carbon based economy to hydrogen based economy in the transportation sector [15, 16]. Moreover, syngas fuel can be produced from coal using Integrated Gasification Combined Cycle (IGCC) [17]. Therefore, syngas is a competitive option for power generation, compared to direct fired pulverized coal plants and natural gas fired combined cycle plants [18, 19].

Great majority of prime movers and auxiliary plants of ocean-going ships are diesel engines [4]. These ships are powered by low-speed, two-stroke diesel engines. The primary reasons for this dominance are their high efficiency and the use of heavy fuel oil, which is a cheap fuel. Unfortunately, it contains large quantities of sulfur and other impurities. As the international marine regulations are expected to prohibit the use of fuels with high sulfur content in the near future, gas turbine-based systems will become competitive in terms of specific fuel consumption cost [20]. They can burn different types of fuels and produce lower emissions. In addition, using combustors with swirl for gas turbines helps in burning the fuel efficiently and reducing the exhaust gas emissions [21, 22, 23, and 24].
The aim of the paper is numerical analysis of different environmental impacts of firing syngas fuel in gas turbine combustor and predicting changes in the firing temperature and emissions, with the reference to natural gas combustion. For this purpose a numerical model for syngas fuel combustion in a can combustor has been developed. The can combustor with swirl is used to burn the syngas fuel. The obtained numerical results are compared with experimental data to validate the CFD model. The effect of syngas compositions and swirl number on both temperature distributions and pollutant emissions is discussed. Finally, NO, CO, CO₂, and SO₂ pollutant emissions are compared with those produced by natural gas and with IMO regulations.

NUMERICAL INVESTIGATION

The paper takes into account the effect of swirl number on the numerically obtained combustion and emissions characteristics of syngas fuel. Measurements were performed using a test rig with a swirl combustor shown in Fig. 1. Fuel and combustion air lines are used for verifying the numerical analysis. The present model was used in [25] to study the effect of secondary air configurations in a gas turbine combustor firing natural gas. The air entering the gas turbine combustor is divided into two streams, the primary air (combustion air) and the secondary air, the latter used for cooling purposes as shown in Fig. 1. The present study focuses only on the combustion air, while the secondary air is not considered since the combustor is water-cooled. Consequently, the secondary air inlets were closed during the reported runs.

The primary air (combustion air) represents a small portion of the total air entering the gas turbine combustor while the big portion is used for combustor cooling. The effect of air to fuel ratio on flame size was studied by [25] using the air to fuel ratios from 30 up to 70. It was found that the air to fuel ratio of 40 is appropriate for obtaining a suitable flame size. Therefore, the air to fuel ratio of 40 was used in the present study. The combustion air was introduced into the combustor through an air swirler mounted coaxially upstream of the combustor, while the fuel was introduced through a centrally located simple nozzle. The swirler used in this study is shown in Fig. 2.

Different air swirlers with the same basic geometry were used. Their inner swirl vane radius (r₁) and the outer swirl vane radius (r₂) were equal to 0.036 m and 0.05 m, respectively. The swirlers had different vane angles (α) of 30°, 39°, 46°, 52°, 58°, and 64° with swirl numbers of 0.5, 0.7, 0.9, 1.1, 1.5 and 2, respectively. The swirl number (S) is defined as follows [26]:

\[
S = \frac{2 \left(1 - \left(\frac{r_1}{r_2}\right)^3\right) \tan \alpha}{3 \left(1 - \left(\frac{r_1}{r_2}\right)^2\right)}
\]

GEOMETRY, MESH, AND BOUNDARY CONDITIONS

The basic geometry of the combustor used in this study is shown in Fig.3. The model geometry and the mesh were generated using the program Gambit [27]. The combustor model has a restricted end, to help in stabilizing the reverse flow zone and consequently the flame within the combustor for all swirl numbers. The dimensions of the combustor are 100 cm in length (z direction) and 20 cm in diameter (x or y direction). The inlet combustion air is guided by the swirler vanes to give the air swirling velocity component. The fuel enters through a simple pipe of 1 cm in diameter, located coaxially at the center of the air swirler. The can combustor outlet shown in Fig. 3 has a conically shaped restriction end with the exit diameter of 5 cm.

A computational mesh was generated for the combustor and the swirler. The model and mesh shown in Figs.3 and 4 represent a tubular combustor with an air swirler of vane angle of 45 degrees. The mesh consists of about 300,000 tetrahedral cells. The three-dimensional models available in the CFD Ansys-Fluent package were used to simulate the turbulent reacting flow. Mesh cells sizes, densities and distributions are essential for numerical solutions. Numerical accuracy can be improved by careful distribution of grid nodes, being a compromise between computational time and numerical accuracy to avoid the need for more complicated meshes. The combustor mesh is divided into a number of regions to adjust cell size distributions, taking into account their importance to numerical solutions. The mesh has high cell density for certain
zones of interest, such as the swirler and the area upstream of the combustor, and low density in other zones of less interest, see Fig.4. The grid quality was checked by applying the quality criteria to tetrahedral mesh cells, as a result of which the maximum cell squish of 0.72 and the maximum cell skew less of 0.75 were obtained.

![Computational mesh for the can combustor and air swirler](image)

CFD MODELING

The finite volume and the first-order upwind methods were used to solve the governing equations. The mathematical equations describing the combustion process are based on the equations of conservation of mass, momentum, and energy, complemented by the equations for turbulence and combustion. The convergence criteria were set to $10^{-6}$ for the continuity, momentum, turbulent kinetic energy and its dissipation rate, energy and radiation equations, and mixture fraction. The shear-stress transport (SST) $k$-$\omega$ model was used to model the turbulence. Its features are given in Table 1.

The SST $k$-$\omega$ model has an advantage in performance over both standard $k$-$\varepsilon$ and $k$-$\omega$ models. The modifications introduced to this model include the cross-diffusion term in the $\omega$ equation and a blending function to ensure that the model equations behave appropriately both in the near-wall zone and in the far-field zone [26, 28]. Then, the equations for the turbulence kinetic energy, $k$, and the specific dissipation rate, $\omega$, are solved.

**Tab.1. Features of turbulence models**

<table>
<thead>
<tr>
<th>Model</th>
<th>Features</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard $k$-$\omega$ model</td>
<td>Solves for $K$-$\omega$, $\omega = \text{Specific dissipation rate (}\varepsilon/k)$</td>
</tr>
<tr>
<td></td>
<td>Recommended for low $Re$ flow, wall bounded boundary layer, and for transitional flows.</td>
</tr>
<tr>
<td>SST $k$-$\omega$ model</td>
<td>Variant of standard $k$-$\omega$, behaves like $K$-$\omega$ in the near wall region and like $k$-$\varepsilon$ in the free stream</td>
</tr>
<tr>
<td></td>
<td>More accurate and reliable for a wider class of flows, like adverse pressure gradient areas in airfoils, transonic shock waves, etc.</td>
</tr>
</tbody>
</table>

The non-premixed combustion with inlet diffusion model was used to model combustion in the present study. In the model, the equation for the conserved scalar is solved, and concentrations of individual component are derived from the predicted mixture fraction distribution. In the non-premixed combustion, the fuel and the oxidizer enter the reaction zone in distinct streams. The analyzed non-premixed model (NPM) has the flowing input chemistry properties: (1) the equilibrium, non-adiabatic, operating pressure equal to 101.325 Pascal, and (2) the fuel stream rich flammability limit ranging from 0.046 to 0.092 according to different fuel compositions. The boundary fuel species used in different runs are shown in Table 2. The P-1 radiation model was used to model radiation from the flame. The model is based on the expansion of the radiation intensity into an orthogonal series of spherical harmonics. The P-1 radiation model is the simplest case of radiation models [29, 30]. The formation of prompt nitric oxide (NO) was included in the calculations and the formation of thermal NO was modelled assuming that all nitrogen in the fuel is released as hydrogen cyanide (HCN), then further reacts to form NO or molecular nitrogen ($N_2$) depending on combustion conditions. The soot distribution within the combustor is calculated using the two-step soot model with soot-radiation interaction.

**Tab.2. Different fuel species for non-premixed model**

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<thead>
<tr>
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</tbody>
</table>

NUMERICAL SOLUTION

After modelling the syngas combustion process in the can combustor, numerical calculations were performed and the results were displayed using the code Ansys-Fluent. Different algorithms were used to solve the governing equations. The Semi Implicit Method for Pressure Linked Equations (SIMPLE) algorithm was used for pressure/velocity coupling. This algorithm satisfies the mass conservation equation by using relations between velocity and pressure corrections. Moreover, in the areas where pressure variations between consequent mesh cells were expected to be significant, the pressure staggering scheme was used which computes the staggered pressure values at the face of each cell to capture pressure non-uniformity. The running steps of the solution procedures are as follows: (1) completing the Probability density function (PDF) look-up table calculation, (2) starting the reacting flow simulation to determine the flow files and to predict the spatial distribution of the mixture fraction, (3) continuing the reacting flow simulation until a convergence solution is achieved, and (4) determining the corresponding values of the temperature and individual chemical species mass fractions from the look-up tables. Multiple species, including radicals and intermediate species, can be included in the problem definition and their concentrations can be derived from the predicted mixture fraction using the assumption of equilibrium chemistry. The property data for the species were accessed through chemical database, and the turbulence-chemistry interaction was modeled using a beta-PDF.
GRID INDEPENDENCE STUDY

Grid independence is an important parameter to be studied during numerical investigation. In order to obtain a grid-independent solution, the generated computational grid should be refined until the solution stops varying when increasing the number of grid cells. The effect of grid refinement on the solution was examined for the combustion air mass flow of 100 g/s. The cell numbers of the examined meshes were equal to 100,000; 200,000; 300,000; 350,000 and 400,000 cells, respectively. The axial velocity distribution is a parameter which can be successfully used in grid independence study, due to its importance in evaluating the flow field in the swirl gas turbine combustor. In particular, the locus of zero axial velocity represents the boundaries of the reversed flow zone which is responsible for flame stabilization [31, 32]. That is why the axial velocity was used here as parameter for studying the grid independence. The axial velocities along the combustor centerline were calculated for the above assumed grid cell numbers. The obtained results show good agreement of axial velocities for grid cell numbers of 200,000 and 300,000, while the results for other cell numbers differ more, as shown in Fig. 5. Consequently, the mesh size of 250,000 cells can provide a sufficiently grid independent and accurate solution.

MODEL VALIDATION

The validation of the CFD model was achieved by comparing the CFD results of temperature distributions with the experimental data recorded for natural gas fuel, published by [25] where the same combustor and the corresponding mesh were used. This was performed by changing the methane fuel concentrations during runs until 100% methane was reached. Once validated, the CFD model was considered useful in analyzing different compositions of syngas fuels and consequently, the remaining results were obtained using this model. The centerline temperature distributions for S = 0.9 and S = 1.5, as obtained from computations and experiments, are shown in Figs. 6-a and 6-b. Comparing these results, a satisfactory and acceptable agreement is observed, with the maximum error of about 8%. The computed temperatures are higher than those coming from measurements. This is due to the fact that the combustor is cooled with water, which absorbs a lot of heat, and further the heat is lost by convection and radiation. In addition, the use of a bare wire thermocouple where its junction does not sense the same temperature of the combustion gases is also a reason why the measured temperatures are relatively far from the calculated ones.
RESULTS AND DISCUSSION

EFFECT OF SYNGAS AND NATURAL GAS MIXTURES ON TEMPERATURE DISTRIBUTIONS

In this section, the effect of using different mixtures of natural gas and syngas fuels on combustion characteristics is analyzed. The natural gas flame is very long at AF=20 and 30. Firstly, the effect of increasing CH₄ percent in syngas fuel was investigated. The CH₄ content was increased until it reached 100%. In the reported case, AF=40 was selected to produce a suitable flame length relative to the combustor configuration. Also, the swirl number of 0.9 was selected for the same reason, and to obtain reasonable comparisons among different cases of CH₄ content, especially when the hydrogen content was high (longer flame).

Methane (CH₄) is an essential component in syngas and it is the main component of natural gas. Consequently, studying the effect of CH₄ percent in syngas is of high importance. Two different syngas compositions were examined. The first composition was: CH₄=0.07, H₂=0.62, N₂=0.018, CO=0.26, CO₂=0.028, H₂O=0.004 [33], and the second one was: CH₄=0.07, H₂=0.26, N₂=0.018, CO=0.62, CO₂=0.028, H₂O=0.004. As can be seen, they differ by H₂ and CO values, while other constituents remain constant. The effect of CH₄ content variation in syngas fuel on temperature maps is shown in Fig. 7. As the value of CH₄ increases, the flame length and the size of high temperature regions decrease. This is due to the decrease of hydrogen content in the fuel mixture. Also, as the methane percent increases, the high temperature region of the flame is shifted to the downstream end of the combustor.

Fig. 7. Temperature maps for different mixtures of syngas and natural gas fuels @ (S=0.9, AF=40)

Axial flame temperature distribution is an important parameter in syngas combustion. Axial temperatures for different percentages of CH₄ up to 100% are shown in Fig. 8. It can be noticed that, as the CH₄ content increases, the axial temperature levels decrease in the front part of the combustor (0≤x/D≤1.5), while in the rear part the effect of adding CH₄ is reversed, i.e. the temperature levels increase with increasing the CH₄ percent (1.5≤x/D≤5).

Fig. 8. Effect of using different mixtures of natural gas and syngas fuels on axial temperature distributions @ (S=0.9, AF=40)

EFFECT OF SYNGAS COMPOSITION ON TEMPERATURE DISTRIBUTIONS

Typical syngas fuels are primarily composed of H₂ and CO, and may also contain smaller amounts of CH₄, N₂, CO₂, H₂O, and other higher hydrocarbons. Their individual composition depends upon the fuel source and processing technology. The effect of variations in syngas compositions on the combustion and emissions in the gas turbine combustor is analyzed in this section. Different compositions of syngas fuels with different volume fractions are given in Table 3.

Tab. 3. Different compositions of syngas fuels [33]

<table>
<thead>
<tr>
<th>Constituents</th>
<th>Syngas No (1) [Schwarzpumpe]</th>
<th>Syngas No (2) [Exxon Singapore]</th>
<th>Syngas No (3) [Tarun]</th>
<th>Syngas No (4) [PST]</th>
<th>Syngas No (5) [Sar axis]</th>
<th>CH₄</th>
</tr>
</thead>
<tbody>
<tr>
<td>H₂</td>
<td>61.9</td>
<td>44.5</td>
<td>37.2</td>
<td>24.8</td>
<td>22.7</td>
<td>0</td>
</tr>
<tr>
<td>CO</td>
<td>26.2</td>
<td>35.4</td>
<td>46.6</td>
<td>39.5</td>
<td>28.6</td>
<td>0</td>
</tr>
<tr>
<td>CH₄</td>
<td>4.9</td>
<td>9.5</td>
<td>9.1</td>
<td>1.5</td>
<td>0.2</td>
<td>100</td>
</tr>
<tr>
<td>CO₂</td>
<td>2.8</td>
<td>17.9</td>
<td>13.3</td>
<td>9.3</td>
<td>5.6</td>
<td>0</td>
</tr>
<tr>
<td>N₂</td>
<td>1.8</td>
<td>1.4</td>
<td>2.5</td>
<td>2.3</td>
<td>1.1</td>
<td>0</td>
</tr>
<tr>
<td>H₂O</td>
<td>0.4</td>
<td>9.3</td>
<td>0.3</td>
<td>22.6</td>
<td>39.8</td>
<td>0</td>
</tr>
<tr>
<td>HHV (kJ/kg)</td>
<td>12492</td>
<td>9477</td>
<td>9962</td>
<td>8224</td>
<td>—</td>
<td>33570</td>
</tr>
<tr>
<td>LHV (MJ/kg)</td>
<td>27.8</td>
<td>12.8</td>
<td>12.7</td>
<td>10.6</td>
<td>—</td>
<td>50.1</td>
</tr>
<tr>
<td>H₂/O</td>
<td>2.36</td>
<td>1.25</td>
<td>0.8</td>
<td>0.63</td>
<td>0.74</td>
<td>—</td>
</tr>
</tbody>
</table>

The syngas composition bearing the name of Schwarzpumpe syngas was selected as one of syngas types for the present analysis. The combustion of the syngas fuel in air converts all carbon and hydrogen in the syngas fuel into carbon dioxide (CO₂) and water vapor (H₂O) in the products, respectively. Thus, simple chemical reactions of the syngas fuel combustion process can be expressed as follows:

\[
CH₄ + 2O₂ + 7.52N₂ \rightarrow CO₂ + 2H₂O + 7.52N₂ \quad (2)
\]

\[
1.5CO + 0.75O₂ + 2.82N₂ \rightarrow 1.5CO₂ + 2.82N₂ \quad (3)
\]

\[
H₂ + 0.5O₂ + 1.88N₂ \rightarrow H₂O + 1.88N₂ \quad (4)
\]
The effect of hydrogen percent in syngas composition on temperature maps is shown in Fig. 9-a. The examined syngas compositions differed by hydrogen content, which amounted to 0.26, 0.44, 0.50 and 0.62. It is shown that, as hydrogen content increases in the syngas fuel, the flame length increases and the size of high temperature regions in the flame becomes larger. This is due to high burning velocity of hydrogen and its high energy content.

Fig. 9-a. Effect of hydrogen percent in syngas composition on temperature maps

Hydrogen content in the syngas fuel has a remarkable effect on temperature distributions. Fig. 9-b shows the effect of syngas fuel composition on axial temperature distributions along the combustor centerline. It is shown that the temperature levels increase with increasing the hydrogen content at the front part of the combustor (0 ≤ x/D ≤ 1.5). In contrast, the flame temperature levels increase with the decreasing hydrogen content in the rear part of the combustor (1.5 ≤ x/D ≤ 5.5). This is because the flame is shorter for lower hydrogen content and the two high temperature regions around the combustor center line come closer to each other. This leads to higher temperature levels for higher hydrogen content near the fuel injector.

Fig. 9-b. Effect of hydrogen percent in syngas fuel composition on axial temperature distributions @ (S=0.9, AF=40)

EFFECT OF SYNGAS COMPOSITION ON POLLUTANT EMISSIONS

Hydrogen percent in the syngas fuel has a remarkable effect on CO₂ species concentrations, NO pollutant concentrations, and soot volume fraction along the combustor centerline. Figure 10-a shows the effect of syngas fuel composition on axial distribution of CO₂ species concentrations along the combustor centerline. The effect is very clear for different hydrogen contents in the syngas fuel. For 26% H₂, the volume of CO₂ increases rapidly to a high mole fraction of 0.045. Then, it decreases to a lower value of 0.02 at the combustor end. For 44% and 50% H₂, the CO₂ volume also increases, but to values lower than that for 26% H₂, and then it decreased again after x/D ≥ 3.2 and x/D ≥ 3.9 for H₂ = 44% and 50%, respectively. For 62% H₂ (Schwarzepumpe syngas fuel type) the CO₂ emission is the lowest of the four examined hydrogen percentages in the syngas fuel. Comparing the results of CO₂ emission with that for natural gas in [25], the use of the Schwarzepumpe syngas fuel results in lower CO₂ emissions than that of natural gas fuel. The maximum values of CO₂ emissions decreased from 0.088 to 0.013 mole fractions at the same operating conditions. In addition, the combustor exit values of CO₂ emissions decreased from 0.048 to 0.013 mole fractions at the same operating conditions.

Fig. 10-a. Effect of syngas fuel composition on axial distribution of CO₂ species concentrations @ (S=0.9, AF=40)

Nitrogen oxide (NO) emissions from syngas combustion come primarily from two sources, which are fuel (prompt) NO formation, and thermal NO formation. Prompt NO formation results from oxidation of nitrogen-bearing species, such as HCN and NH₃, which evolved from the fuel during gasification. The thermal NO arises from oxidation of molecular nitrogen (N₂) in the combustion air, and is formed in appreciable quantities at elevated combustion temperatures, ranging above 1370°C [34]. Figure 10-b shows the effect of syngas fuel composition on axial distribution of NO pollutant concentrations along the
combustor centerline. Both thermal and prompt mechanisms were selected for modeling NO pollutants. It is shown that NO mole fractions decrease with increasing the hydrogen content in the front part of the combustor. Then, the NO pollutant concentrations increase in the rear part of the combustor with increasing the hydrogen content. The maximum NO level remarkably increases with increasing the hydrogen content and is shifted to the rear end of the combustor. The NO values decrease sharply at the end of the combustor due to the reaction with oxygen in air to produce NO₂ and NO₃. Comparing the results for NO emissions with those published for natural gas in [25], the use of the Schwarzepeumpe syngas fuel results in lower NO emission than that produced by natural gas fuel. The maximum values of NO emissions decreased from 0.00045 to 5e-5 mole fractions at the same operating conditions. The NO and CO₂ results are satisfactory and agree with those reported in Ref. [35]. Consequently, the syngas fuel satisfies the new international NOₓ emission regulations, as the natural gas satisfies new IMO regulations on NOₓ emission, according to Lloyd’s Register options for the compliance with new IMO regulations[36, 37].

Finally, Fig. 10-c shows the effect of syngas fuel composition on axial distribution of soot volume fraction along the combustor centerline. The maximum volume fraction of soot exists at the middle section of the combustor, for hydrogen contents of 0.44 and 0.50. The concentration of the soot volume fraction increases for hydrogen content of 0.62 more than that for 0.26 but less than for 0.44 and 0.50, with the maximum value more shifted towards the rear section of the combustor.

EFFECT OF SWIRL NUMBER ON TEMPERATURE DISTRIBUTIONS

The zero axial velocity point divides the flow within the combustor into two zones with respect to the flow direction, which are the forward and reversed flow zones (RFZ). The RFZ is found within the recirculation zone (RZ) and has slightly smaller volume than RZ. So, the two zones can be considered the same. For high swirl numbers, the adverse axial pressure gradient is sufficiently high to result in reverse flow along the combustor axis and setting up an internal RZ. This RZ plays an essential role in flame stability. It constitutes a mixed zone of combustion products and acts as a source of heat. It chemically activates species near the swirler exit.

The air swirl number was changed by using six swirlers with the same dimensions but different vane angle inclinations, ranging from 30° up to 64°. For swirl numbers above 0.5, the RFZ took the shape of a closed loop and ended inside the combustor. This helps in shortening the flame. The filled contours of flame temperatures or temperature maps indicate the flame shape and size. Figures 11-a and 11-b show the effect of swirl number on syngas and natural gas temperature maps at different swirl numbers, from 0.5 to 2.0. It is shown that, as the swirl number increases, the flame size decreases and the high temperature regions become attached to the upstream direction. Consequently, the flame length is decreased. The temperature maps are nearly the same for moderate swirl numbers S=0.7 and S=0.9. But for S >1(strong swirl), the swirl number has more effect on the flame size.

As the swirl number increases, the high temperature regions are shifted upstream, since the RFZ moves relative to the swirler exit. Also, for lower swirl numbers, a higher flame temperature is obtained since the residence time for the mixture is reduced. Thus, the fuel is efficiently burned and the emissions of pollutants are reduced. This is due to good mixing of the fuel and air. The combustion process is enhanced and this leads to lower emissions.
Swirl number has an effect on the temperature distribution, since increasing the swirl number leads to the decrease of the flame size and, consequently, to the reduction of the flame temperature inside the combustor, partially at the combustor end. This is a benefit when selecting the turbine blade material. Fig. 12 shows the effect of swirl number on axial temperature distributions along the combustor centerline. It is shown that, as the swirl number increases, the temperature level increases in the front part of the combustor and decreases in the rear part. The temperature increase in the front combustor part is due to good mixing between fuel and air, as a result of high turbulence intensity caused by higher swirling effect. Far from the swirller exit, the swirl effect and the turbulence decrease, and this leads to the decrease of the temperature levels.

EFFECT OF SWIRL NUMBER ON POLLUTANT EMISSIONS

Air swirl number has an obvious effect on the flow field and combustion inside the gas turbine combustor. As the swirl number increases, the reverse flow zone also increases and flame stabilization is improved. Combustion characteristics are enhanced by improving flame stabilization, as shown in details in Figs. 13-a and 13-b. Figure 13-a shows the effect of swirl number on unburned fuel Carbon monoxide species. CO in syngas combustion products comes from two primary sources: (1) unburned syngas CO, resulting from inefficient fuel/air mixing which creates regions with equivalence ratios outside the ignition range, and (2) incomplete combustion of hydrocarbon species in the syngas. CO emissions from gas turbine-based IGCC systems are lower than those produced by conventional combustion-based systems [38]. The hydrogen content in synthesis gaseous fuels extends the operational limit and facilitates the oxidation of other organic fuel fractions by providing elevated combustion temperatures. For turbine and boiler applications, CO emissions from syngas combustion are lower than those observed in diesel or spark ignition engines [34]. Comparing the results for CO emissions with those for natural gas in [25], the use of the Schwarzepumpe syngas fuel results in higher CO emission than that of natural gas fuel. The maximum values of CO emissions increased from 0.028 to 0.14 mole fractions at the same operating conditions. However, the maximum CO value for syngas decreases at the end of the combustor, but is still slightly higher than that of natural gas. This result agrees with those reported in Ref. [35].

Figure 13-b shows the effect of swirl number on unburned fuel species H2. The H2 concentrations decrease along the combustor length and approach zero at about x/D≤ 2, due to burning out. Also, the H2 concentrations decrease along the combustor centerline with the increase of the swirl number and approach zero for all swirl number (S), in the combustor part between 2.0 ≤x/D≤5. In addition, as the swirl number increases, the N2 concentrations decrease along the combustor centerline in the combustor section (x/D≤1.5), remaining nearly constant in subsequent sections, as shown in Fig. 13-c. Figs. 13-d and 13-e show that as the swirl number increases, the concentrations of HCN and CO2 decrease in the rear section of the combustor, at x/D≥1.5. In the other part of the combustor this trend is inversed, due to the turbulence effect produced by the swirling flow which improves air/fuel mixing. As a consequence, the exhaust gas emissions decrease in the end part of the combustor size.
Finally, a sulfur component in the syngas fuel is hydrogen sulfide, the concentrations of which can exceed 2% for sulfur-rich fuels. Lower concentrations of other reduced sulfur compounds, such as carbonyl sulfide, mercaptans, dimethyl sulfide, dimethyl disulfide, and carbon disulfide, may also be observed. Prior to combustion in the gas turbine, nearly all sulfur compounds are to be removed from the synthesis gas [39]. The syngas fuel used in the present paper contains no sulfur, and there are no emissions of sulfur oxides from the can combustor. This satisfies the international IMO SOx emission regulations.

CONCLUSIONS

Numerical modeling of syngas fuel combustion in a can combustor with swirl is presented as a solution proposed to satisfy the requirements of international marine regulations. Four examined syngas fuel compositions differed by H2/CO ratios, ranging from 0.63 up to 2.63. Also, mixtures with different natural gas-syngas ratios were investigated. The SST k-ω model was used for turbulence modeling. The non-premixed combustion model was used to model combustion, and P-1 for modeling radiation. The effect of syngas fuel compositions on the flame size, axial temperature, as well as on emissions of carbon dioxide (CO2) and nitrogen oxides (NO) was examined. Based on the reported study the following conclusions have been formulated:

1. Comparing the measured and calculated axial temperatures of natural gas combustion for swirl numbers of 0.9 and AF ratio of 40, provided satisfactory agreement.
2. At lower swirl numbers, the temperature levels are high. Also, it has been noticed that in those cases the high temperature regions have large sizes for both natural gas and syngas fuels.
3. With increasing the swirl number, the temperature levels and the high temperature region decrease. They also shift towards the front of the combustor, thus leading to the decrease of the flame length.
4. Increasing the swirl number leads to the decrease in axial temperature levels, and CO, H2, HCN and CO2 concentrations. At the same time the N2 concentrations increase along the combustor centerline. The syngas composition has a remarkable effect on flame shape and size as indicated in the temperature maps. The flame length and the size of high temperature regions increase with the increased hydrogen content in the syngas fuel.

5. Syngas combustion temperature levels depend on the combustible constituents (hydrogen, carbon monoxide, and methane), and on their heating values and the presence of non-combustibles (inert) constituents in the syngas fuel.

6. Axial temperatures of syngas fuel increase with the increase of hydrogen content, especially at the end of the combustor.

7. The concentrations of pollutant species depend on syngas constituents (hydrogen, carbon monoxide, and methane). The volume of CO2 decreases with the increase of hydrogen content, while NO emissions increase as the hydrogen content increases.

8. Temperature levels for syngas flames are lower than those of natural gas. Increasing the methane content leads to the increase of axial temperatures, in particular near the combustor end.

9. Syngas fuel has lower emissions of NO and CO2 than those from natural gas fuel at the same operating conditions. It has higher peak flame temperature and CO emissions, as compared to natural gas fuel at the same operating conditions. Also, the used syngas fuel has no sulfur oxides emissions.

10. Finally, using syngas fuel in the can gas turbine combustor with swirl can meet the required new IMO emission regulations.

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A MODEL OF FUEL COMBUSTION PROCESS IN THE MARINE RECIPROCATING ENGINE WORK SPACE TAKING INTO ACCOUNT LOAD AND WEAR OF CRANKSHAFT-PISTON ASSEMBLY AND THE THEORY OF SEMI-MARKOV PROCESSES

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Gdańsk University of Technology, Poland

ABSTRACT

The article analyses the operation of reciprocal internal combustion engines, with marine engines used as an example. The analysis takes into account types of energy conversion in the work spaces (cylinders) of these engines, loads of their crankshaft-piston assemblies, and types of fuel combustion which can take place in these spaces during engine operation. It is highlighted that the analysed time-dependent loads of marine internal combustion engine crankshaft-piston assemblies are random processes. It is also indicated that the wear of elements of those assemblies resulting from their load should also be considered a random process. A hypothesis is formulated which explains random nature of load and the absence of the theoretically expected detonation combustion in engines supplied with such fuels as Diesel Oil, Marine Diesel Oil, and Heavy Fuel Oil. A model is proposed for fuel combustion in an arbitrary work space of a marine Diesel engine, which has the form of a stochastic four-state process, discrete in states and continuous in time. The model is based on the theory of semi-Markov processes.

Keywords: engine load, marine internal combustion engine, probability, semi-Markov process, crankshaft-piston assembly

INTRODUCTION

The safety of motion of a sea going vessel significantly depends on, among other factors, correct operation of the main engine and auxiliary internal combustion engines on this vessel [7, 13, 17]. The operation of marine engines of this type can be interpreted as the conversion of energy $E(t)$ in their work spaces to produce heat and work, which are then passed to receivers at a given time $t$ [4, 13, 15, 16]. What is interesting, in this interpretation the operation of engines of this type (in an evaluating approach) can be expresses by a physical quantity with the attributed numerical value and the unit of measurement bearing the name of joule-second [4].

In Diesel engines, energy conversion $E(t)$ initially consists in chemical conversion $E_{C, \text{ch}}$ of the energy contained in the fuel-air mixture formed in the combustion chamber into the internal energy of the exhaust gas generated during the combustion. The product of this conversion is characterised by thermal energy $E_C$. In author's opinion, this form of energy conversion can be referred to as heat. The next form of energy conversion consists in converting the thermal energy $E_C$ of the exhaust gas into mechanical energy $E_M$ of the crankshaft-piston assembly (Fig. 1). In the same convention, this form of energy conversion can be referred to as work, as it makes it possible for the engine to perform useful work $L$. This work is an effect of action of the torque ($M_t$) generated by the crankshaft-piston assembly at given engine revolutions ($n$) [1, 2, 4, 5, 6, 9]. Hence, the operation of an arbitrary engine can be interpreted as conversion of energy $E(t)$ which leads to
useful work $L_s = f(t)$ done in time $t$. In this interpretation the entire action can be described in the form of the relation [4]:

$$D_{k_s} = \int_0^T L_s(t) \, dt = 2\pi \int_0^\pi n(\tau) M_s(\tau) \, d\tau = 2\pi \cdot c \cdot \int_0^\pi n(\tau) p_s(\tau) \, d\tau$$

(1)

The above interpretation of engine operation has not only the cognitive but also practical value, as it enables to make distinction between possible engine operation $D_{k_s}$ and the operation $D_{r_s}$ required by the user of this engine. The possible operation $D_{k_s}$ depends on work cycles executed in this engine, which are in turn affected by the technical state of the engine resulting from its wear and by chemical properties of the fuel-air mixture formed in the combustion chambers. On the other hand, the required operation $D_{r_s}$ reflects requirements of the engine user, in terms of energy characteristics which are to be met for the ship to perform successfully its transport task.

Two facts result from the relation (1). The first fact is that the operation of the Diesel engine depends on correct execution of consecutive work cycles, as the average useful pressure $(p_s)$, and, consequently, the average torque $(M_s)$ of this engine depend on how much the real work cycles differ from their theoretical course. The other fact is that the engine is capable of performing its task if the following inequality is fulfilled:

$$D_M \geq D_W$$

(2)

Real work cycle execution depends mainly on the course of fuel combustion process in the work spaces (cylinders) of the reciprocal internal combustion engine. The correct course of this process mainly depends on time histories of: combustion pressure $- p_s = f(\alpha)$, combustion temperature $- T_s = f(\alpha)$, heat generation rate $- dq_s/d\alpha = f(\alpha)$ and fuel doses delivered to engine cylinders $- G_s = f(\alpha)$ [13, 15]. The combustion in the engine work space can have a quasi-detonation form or even, rather rarely, a detonation form, or a non-detonation (normal) form. The normal combustion can be full (complete and perfect) or partial, i.e. complete, but imperfect, or even incomplete and imperfect [13, 16, 17]. Each type of fuel combustion in cylinders generates different combustion pressures and temperatures. As a consequence, different loads of the crankshaft-piston assembly are observed in engines of this type (Fig. 1). The quasi-detonation combustion is the cause of significant bearing loads, while the imperfect combustion leads to the formation of carbon deposits on combustion chamber elements, in particular on the piston crown. In this context, it would be of high importance to assess the probability of appearance of particular types of combustion. This piece of information can be obtained after modelling the real combustion process in the engine as the stochastic process $[X(t); \quad t \geq 0]$, with the values having the form of such states as: $s_1$ – full (complete and perfect) combustion, $s_2$ – imperfect combustion, $s_3$ – incomplete and imperfect combustion, $s_3$ – quasi-detonation combustion. This model would enable to assess the probabilities of appearance of particular states: $p_1 = P(s_1)$, $p_2 = P(s_2)$, $p_3 = P(s_3)$ and $p_4 = P(s_3)$ of the process $[X(t); \quad t \geq 0]$ as the model results from specific nature of loads of crankshaft-piston assemblies in engines of this type. These loads and the resultant wear of their elements, in both linear and volumetric form, mainly depend on the course of the combustion process in work spaces of these engines [1, 7, 9, 12, 13, 17].

CRANKSHAFT-PISTON ASSEMBLY LOADS IN INTERNAL COMBUSTION ENGINES

During the operation of marine trunk piston Diesel engines, chemical energy of fuel delivered to the engine work space is initially converted into the internal energy of the exhaust gas generated during the fuel combustion process, and then to the mechanical energy of piston motion. The mechanical energy is passed by the crankshaft-piston assembly to the receiver, for instance to the ship propeller, electric generator, piston compressor, or displacement pump. A sample scheme of energy conversion in the work space of a trunk piston engine and the formation of loads of its crankshaft-piston assembly is shown in Fig. 1.

**Fig. 1. Sample scheme of energy conversion in a trunk piston Diesel engine with pin connection between piston and connecting rod head: Pl – fuel, Pw – air, ECh – chemical energy, EC – thermal energy, EM – mechanical energy, ET – thermal and mechanical loss energy, UTK SoZS – Diesel engine crankshaft-piston assembly, SoZS – Diesel engine, OEM – mechanical energy receiver (ship propeller, electric generator, compressor, pump), Q(t) – thermal load at time t, Qm(t) – mechanical load at time t, GMP – top dead centre of piston motion, DMP – bottom dead centre of piston motion, 1 – connecting rod length, r – crank radius, 1 – piston pin, 2 – connecting rod, 3 – crankpin, 4 – crankshaft bearing bush, 5 – main bearing bush, 6 – main journal.**
Part of the energy converted in the engine work spaces is used to perform useful work \( (L) \), while the remaining part \( (E_p) \) is lost (dissipated), according to the second law of thermodynamics [16]. The process is accompanied with time dependent thermal loads \( (Q_t) \) and mechanical loads \( (Q_m) \) of crankshaft-piston assemblies of those engines.

It is well known from empirical examinations that the loads of engine crankshaft-piston assemblies at time \( t \) cannot be precisely predicted [5, 7, 8, 9]. Therefore the following hypothesis \( H \) can be formulated: „at an arbitrary time \( t \) the load of crankshaft-piston assembly of the internal combustion engine is a random variable, as its values recorded in consecutive measurements can only be predicted with certain probability”.

The following consequences result from this hypothesis:

- \( K_1 \) – in each work cycle of the engine taking place in a given cylinder we cannot predict precisely the dose \( (G_i) \) of fuel injected to the combustion chamber. It can only be predicted with a certain probability, which can be assessed using its point or interval estimation,
- \( K_2 \) – in each work cycle of the engine taking place in a given cylinder we cannot predict precisely the maximum value \( (p_{\text{max}}) \) of the combustion pressure. It can only be predicted with a certain probability, which can be assessed using its point or interval estimation,
- \( K_3 \) – in each work cycle of the engine taking place in a given cylinder we cannot predict precisely the maximum value \( (T_{\text{comb}}) \) of the combustion temperature. It can only be predicted with a certain probability, which can be assessed using its point or interval estimation,
- \( K_4 \) – in each work cycle of the engine taking place in a given cylinder we cannot predict precisely the heat generation rate \( (dq/d\alpha) \) during fuel combustion at an arbitrary angular crankshaft position. It can only be predicted with a certain probability, which can be assessed using its point or interval estimation.

The above hypothesis can be verified using the non-deductive inference method, bearing the name of reductive inference, which bases on the following scheme:

\[
K_1, K_2, K_3, K_4 \implies H
\]  

(3)

Verification of the hypothesis \( (H) \), done via experimental examination of the truth of the above-named consequences \( (K_i, i = 1, 2, 3, 4) \), requires acceptance of the truth of the following syntactic implication:

\[
H \implies K_i (i = 1, 2, 3, 4)
\]  

(4)

The logical interpretation of the inference scheme (3) is the following: if experimental verification of consequence \( K_i (i = 1, 2, 3, 4) \) has confirmed its truth, then if the implication (4) is true, then the hypothesis \( H \) is also true and can be accepted.

The inference carried out in accordance with the scheme (3) does not lead to sure conclusions, but only to probable ones.

It results from the above considerations that during the operation of the engine its thermal and mechanical loads [2, 4, 9, 11, 13, 15, 17], and, consequently, the loads of its crankshaft-piston assemblies at an arbitrary time \( t \) should be considered random variables. As a further consequence, the thermal load process and the mechanical loads process should be considered stochastic processes, \( \{Q_t(t); t \geq 0\} \) and \( \{Q_m(t); t \geq 0\} \), respectively. These two processes compose the stochastic process of total load \( \{Q(t); t \geq 0\} \) taking place during engine operation. Hence the total load \( Q(t) \) of crankshaft-piston assemblies of an arbitrary internal combustion engine during its operation can be presented as:

\[
Q(t) = Q_t(t) + Q_m(t)
\]  

(5)

where:

\[
Q_t(t) = Q_{t\text{c}}(t) + Q_{t\text{D}}(t)
\]

In these formulas:

- \( Q_{t\text{c}}(t) \) – total load resulting from engine action \( (D) \) at time \( t \) (1),
- \( Q_{t\text{D}}(t) \) – thermal load carried by engine elements during its operation at time \( t \),
- \( Q_{m}(t) \) – mechanical load connected with the presence of gravity forces and the appearance of gas, friction, and inertia forces at time \( t \),
- \( Q_{t\text{D}}(t) \) – thermal load resulting from the internal energy increase \( \Delta U(t) \), at time \( t \), of the elements through which part of thermal energy \( E_p(t) \) penetrates, thus generating their load \( Q_{t\text{D}}(t) \),
- \( Q_{t\text{c}}(t) \) – load of engine elements composing work spaces (combustion chambers) and other elements as a result of transmission of the dissipated (lost) energy to the environment at time \( t \) (Fig. 1).

When studying these processes for an arbitrary time \( t \), the objects of interest are random variables \( Q_{t\text{c}}, Q_{t\text{D}} \) and \( Q_{t\text{D}} \) which are, respectively, the values of the processes \( \{Q_{t\text{c}}(t); t \geq 0\}, \{Q_{t\text{D}}(t); t \geq 0\} \) and \( \{Q_{t\text{D}}(t); t \geq 0\} \).

In this interpretation the load is a result of the abovementioned energy conversion in each work space of the engine during its operation (Fig. 1). The thermal load leads to both linear and volumetric wear of tribological systems of the engine. Significant load and wear mainly concerns cylinder liners, cylinder head, and piston rings. They are extremely high in case of cylinder liner walls in the combustion chambers and piston crowns, which are water cooled [13, 17]. Remarkable loads are also recorded in outer layers of bushes and journals of both main and crankshaft bearings. They are caused by friction effects in bearings of this type. An additional load of these bearings, which can affect the rate of their wear, is mechanical load observed during detonation combustion. In general, the wear, both linear and volumetric, of crankshaft-piston assembly components which results from their load is a random process, as the loads of these components depend on the course of the combustion process in the engine work spaces.

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COMBUSTION PROCESS AS THE CAUSE OF LOAD AND WEAR OF CRANKSHAFT-PISTON ASSEMBLY

The combustion process, especially in the ship’s main engine, should be considered a main cause of load and wear of elements composing its crankshaft-piston assembly. This wear, in both linear and volumetric form, is the cause of failures, frequently premature, of such important system components as pistons and main and crankshaft bearings. Both the load itself and its effects, in the form of wear and damages of crankshaft-piston assemblies, in this type of engines reveal random nature. The reason of this randomness is mainly a random course of fuel combustion in the work spaces of the abovementioned engines. This effect is extremely important in pressure charged engines, as engine charging enables to combust larger fuel doses in engine work spaces, which proportionally increases the average indicated pressure and, as a further consequence, the average useful pressure [2, 13, 15]. All this leads to higher torque (1) and higher engine power, without changing its rotational speed and/or cylinder displacement. The thermal load is also increased, without proportional increase of the maximum combustion pressure. That means that the mechanical load of the engine increases to a lesser degree, as compared to the thermal load. Smaller increase of the mechanical load results from the fact that charging the Diesel engine decreases the ignition delay time. This leads to softer engine operation, as the combustion process begins with rapid oxidation of very small fuel portions. In this situation the maximum combustion pressure \( p_{\text{max}} \) increases more slowly than the average indicated pressure \( p_{\text{m}} \) and the average useful pressure \( p_e \). In Diesel engines the pressure build-up ratio for pressures generating thermal and mechanical stresses [2] can be defined as:

\[
\sigma_p = \xi \frac{p_{\text{max}}}{p_e}
\]  

(8)

where:

\( \sigma_p \) – pressure build-up ratio,

\( p_{\text{max}} \) – maximum pressure in the engine work space (cylinder)

\( p_e \) – average useful pressure in the engine cylinder

\( \xi \) – pressure proportionality factor (\( \xi < 0,5 \)).

Formula (8) shows that increasing the average useful pressure \( p_e \) does not result in the same increase of the maximum combustion pressure \( p_{\text{max}} \) – for instance the appearance of a twice as high value of \( p_e \) does not double the value of \( p_{\text{max}} \) [2, 13, 15]. As a result, the mechanical stresses coming from mechanical loads are lower than those generated by thermal loads. In the Diesel engine the value of the arising maximum pressure, and consequently the maximum mechanical load of the crankshaft-piston assembly, can be limited by gradual injection of the fuel dose to the combustion chamber, done in such a way as to prevent rapid self-ignition of the entire fuel dose. This action is necessary as the oxidation rate of hydrocarbons increases significantly with the number of carbon (C) atoms in the fuel molecule, and those conditions make the appearance of the detonation combustion more likely. The intensity of detonation combustion increases for higher compression ratio \( \varepsilon \). In fact, in Diesel engines the compression ratio is much higher than the level at which the detonation combustion can theoretically take place, but it does not appear in real engine operation. The reason is that in those engines the macro- and microstructure of the fuel-air mixture at the initial time of self-ignition is far from ideal, which leads to insufficient amounts of oxygen in some areas of the combustion chamber and the creation of hydrocarbon peroxides in the first self-ignition centres, and to disintegration of \( C_n \) molecules into molecular fractions, characterised by lower oxidation rate [2]. Therefore, in authors opinion, the following hypothesis \( H \) can be formulated: „despite a high combustion ratio \( \varepsilon \), the detonation combustion does not take place, in general, in a Diesel engine because in engines of this type the fuel-air mixture at the initial time of self-ignition has the macro- and microstructure which is far from ideal, due to the presence of areas in the combustion chamber with insufficient amounts of oxygen, which leads to the formation of hydrocarbon peroxides in first chemical reactions, accompanied by disintegration of \( C_n \) molecules into molecular fractions characterised by lower oxidation rate". This hypothesis explains why the detonation combustion does not take place in the work spaces of correctly working Diesel engines. According to this hypothesis, it is highly unlikely that the flame front appearing just after the ignition delay will move at supersonic speed and thus create an exhaust gas shock wave, with the resultant excessive load of crankshaft-piston assembly bearings. However, combustion similar to the detonation combustion can appear, which is also very dangerous to the durability of bearings. The term proposed by the author for this type of combustion is quasi-detonation combustion. It is characterised by such a rate of oxidation of combustible fuel particles which does not lead to the formation of a shock wave, but the pressure build-up ratio \( \varphi_m \) in this combustion exceeds 1MPa/1°OWK (1° of crankshaft revolution) [15]. This type of combustion can appear when the injection system works incorrectly, causing the ignition of the entire dose of the injected fuel. In this situation the phenomenon of flame front spreading at a speed nearing the speed of sound can be observed. The accompanying mass of the created exhaust gas may be characterised by high rates of temperature build-up \( \varphi_T = \frac{dT}{d\alpha} \) and pressure build-up \( \varphi_p = \frac{dp}{d\alpha} \). In those conditions the pressure build-up rate \( \varphi_p \) can exceed 1,2MPa/1°OWK.

Detonation combustion in Diesel engines can be avoided, for a given fuel, by decreasing the rate of injection and, consequently, fuel atomisation in engine cylinders. The lower limit of decreasing the fuel atomisation rate is the appearance of high opacity of the exhaust gas [2, 13, 15].

During engine idling or partial load, large amounts of hard carbon deposits form as a result of insufficient atomisation [13, 17]. These deposits, even after a relatively short time, can worsen the tightness of the exhaust valves and be a source of excessive pollution of the combustion chamber. The
polluted combustion chamber has a smaller volume, which leads to the increase of the compression ratio ($\varepsilon$), pressure ($p_c$) and temperature ($T_c$) at the end of compression of the fresh fuel charge. Larger $\varepsilon$ and, consequently, larger $p_c$ and $T_c$ may decrease the earlier adjusted ignition delay, and thus quicken the self-ignition in the cylinder. As a result, especially in the engine full-load mode, the temperature and pressure of the exhaust gas in the combustion chamber are high and characteristic for quasi-detonation combustion. In those combustion conditions, carbon reach particles can be generated which delay the course of combustion and lead to the appearance of soot in the exhaust gas. [2].

In the Diesel engine, critical loads are thermal loads which generate mechanical stresses, for this reason bearing the name of thermal stresses. This is because a larger fuel dose leads to the formation of larger heat flux in the cylinder, with the resultant larger heat flux density and higher temperatures of the exhaust gas generated in fuel combustion. The amount of heat to be dissipated per unit time with respect to the combustion chamber surface area increases, as the temperature of the water used to cool the engine remains unchanged. That means that the thermal loads generated in combustion chamber elements by temperature gradients increase to the same extent. An unfavourable factor which additionally increases this load is the fact that the highest temperature increase is recorded when the work space of the cylinder has the volume equal to that of the combustion chamber. In this situation the heat transfer surface is the smallest and changes most slowly, as a result of which the piston crown, the upper part of the cylinder liner, and the inner wall of the cylinder head are the thermally heaviest loaded elements. [2, 13, 17]. Moreover, changes of thermal state of the engine, its wear, and incorrect adjustment of the injection apparatus can be a reason for partial (incomplete, imperfect) combustion which emits large amounts of harmful compounds [1, 12, 13]. The above considerations suggest that it is advisable to analyse such combustion process states as: $s_1$ – full combustion (complete and perfect), $s_2$ – imperfect combustion (with small amount of emitted harmful compounds), $s_3$ – partial combustion (imperfect and incomplete – with large amount of emitted harmful compounds), $s_4$ – quasi-detonation combustion. Evidently, during engine operation the above states take place with different probabilities. Assessing the probabilities for each of these process states to take place requires creating a combustion process model, the values of which are the above states $s_i$, $i = 1, 2, 3, 4$ [3, 7, 10, 14].

MODEL OF COMBUSTION PROCESS IN ENGINE CYLINDERS

The analysis of the course of load and combustion processes shows that their states can be determined in such a way that the duration time of an arbitrary state existing at time $\tau_n$ and the state which will be available at time $\tau_{n+1}$ do not depend stochastically on states which earlier took place, nor time intervals of their existence [5, 6, 7]. That means that a semi-Markov model can be created for each of these processes [6, 8, 10].

It results from the theory of semi-Markov processes that creating a model of an arbitrary real process in the form of the semi-Markov process is possible once we describe its initial distribution $P_i = P_i(t = 0)$ and the functional matrix $Q(t)$ [5, 6, 7, 8, 9, 10, 14].

Taking into account specific nature of the combustion process in work spaces (cylinders) of Diesel engines we can assume that the combustion process $\{X(t): t \geq 0\}$ with the set of states $S = \{s_1, s_2, s_3, s_4\}$ has the following initial distribution [7, 10]:

$$P_i = P_i(t = 0) = \begin{cases} 1 & \text{for } i = 1, \\ 0 & \text{for } i = 2, 3, 4 \end{cases}$$

and the functional matrix

$$Q(t) = \begin{bmatrix} 0 & Q_2(t) & 0 & 0 \\ 0 & 0 & Q_3(t) & 0 \\ 0 & Q_4(t) & 0 & Q_1(t) \\ Q_1(t) & Q_2(t) & 0 & 0 \end{bmatrix}$$

where:

$$Q_{ij}(t) = P_t\{X(\tau_{n+1}) = s_j, X(\tau_n) = s_i\}$$

The initial distribution (9) results from the fact that after the production cycle the internal combustion engine should be ready to use [6, 7, 10]. Then the combustion process $\{X(t): t \geq 0\}$ (taking place in individual cylinders) should take such a course as to ensure the existence of state $s_i$ of this process. The appearance of the remaining states $s_j \in S (i = 2, 3, 4)$ is connected with the decreased engine serviceability, first of all with incorrect operation of the injection apparatuses during engine operation.

The functional matrix (10) and the resultant graph of process state changes $\{X(t): t \geq 0\}$, shown in Fig. 2, reflect rational use of the engine, which has place when the user, after detecting any of engine states $s_j$, $s_i$ or $s_u$, undertakes a relevant technical action to repair the engine and thus restore its state $s_i$. When the state $s_i$ is detected by the diagnosing system, the engine user should eliminate causes of its appearance and restore the engine to the state $s_i$, i.e. to full (complete and perfect) combustion in cylinders. Such an action is not always possible in practice, hence the appearance of the state $s_i$ is quite frequent. It that situation the only rational action to be performed by the engine user is that leading to technical service which will allow to move the process back to the state $s_i$.

An important characteristic of the combustion process $\{X(t): t \geq 0\}$ in Diesel engine cylinders, which can be used to assess its course, is a set of conditional probabilities of process passing to particular states in the time not longer than $t$.

$$P_j(t) = P_t\{X(t) = s_j, X(0) = s_i\}, (i, j = 1, 2, 3, 4; i \neq j)$$
This formula describes probabilities of passing from state $s_i \in S$ at time $t = 0$ to state $s_j \in S$ at time $t \neq 0$.

$$
\begin{align*}
\mathbb{P}_j &= \lim_{i \to \infty} \mathbb{P} \{X(t) = s_j\}; \quad s_j \in S, j = 1, 4
\end{align*}
$$

which is operationally solved using the Laplace-Stieltjes transformation. In formula (12) $G(t)$ is the distribution function of the random variable $T_j$ representing the time duration of state $s_j$ ($j = 1, 2, 3, 4$) of the Markov chain $\{X(t)\}$ placed into the process $\{X(t)\}$ fulfills the equation system [7, 10]:

$$
\begin{align*}
\mathbb{P}_j &= \frac{\pi_j E(T_i)}{\sum_{i=1}^{n} \pi_i E(T_i)}, \quad j = 1, 2, 3, 4
\end{align*}
$$

where the limiting distribution $\pi_j, j = 1, 2, 3, 4$ of the Markov chain $\{X(t)\}$: $n = 0, 1, 2, ...$ placed into the process $\{X(t)\}$ fulfills the equation system [7, 10]:

$$
\begin{align*}
\begin{pmatrix}
0 & p_{12} & 0 & 0 \\
0 & 0 & p_{23} & 0 \\
0 & p_{32} & 0 & p_{34} \\
p_{41} & p_{42} & 0 & 0
\end{pmatrix}
= \begin{pmatrix}
\pi_1, \pi_2, \pi_3, \pi_4
\end{pmatrix}
\end{align*}
$$

From equations (16) and (17) the following form of the limiting distribution of the process $\{X(t)\}$ is obtained:

$$
\begin{align*}
P_1 &= \frac{P_{23} P_{34} E(T_1)}{M}; \quad P_2 = \frac{E(T_2)}{M}; \\
P_3 &= \frac{P_{23} E(T_3)}{M}; \quad P_4 = \frac{P_{23} P_{34} E(T_4)}{M}
\end{align*}
$$

and: $P_1, P_2, P_3, P_4$ — probabilities that the process $\{X(t)\}$ will reach the state: $s_i, s_j; \quad i, j = 1, 2, 3, 4$; $i \neq j$.

The probability $P_i$ can be considered an indicator of correct fuel combustion in engine cylinders and, consequently, correct operation (work) of the engine, while the probabilities $P_j (j = 2, 3, 4)$ will indicate incorrect fuel combustion in the cylinders and incorrect engine operation.

Rational action of the engine user should aim at preventing the appearance of state $s_j$ (then $P_j = 0$) and take care that the time durations of states $s_i$ and $s_j$ of the fuel combustion process are as short as possible ($P_i \approx 0$ and $P_j \approx 0$). This action requires relevant control of the combustion process during engine operation, for instance by following concepts presented in [1, 7, 12, 13], which will lead to the fuel combustion model $\{X(t)\}$ with the graph of state changes shown in Fig. 2.
The presented probabilistic model of fuel combustion process in Diesel engine work spaces (cylinders) can be used for assessing the scale of adaptation of certain types of engines to complete and perfect, as well as detonation-free fuel combustion during relatively long time intervals of engine operation (work), and for assessing correctness of use of these engines in given operating conditions [1, 5, 7, 12, 13]. The model is a four-state model, worked out in the form a semi-Markov process, discrete in states and continuous in time. The values of this process are states: $s_1$, $s_2$, $s_3$, $s_4$ having the following interpretations: $s_1$ – full (complete and perfect) combustion, $s_2$ – imperfect combustion, $s_3$ – partial (incomplete and imperfect) combustion, $s_4$ – quasi-detonation combustion. Generally, it can be assumed that the lower the probability $P_i = P(s_i)$, the less correctly the engine is used. The probability $P_i$ can be interpreted as the probability of correct (faultless) engine operation.

The procedure should consist in evaluating the numbers $p_{ij}$, No 3/201656 POLISH MARITIME RESEARCH

The probabilities given by formulas (18) can be considered indices of Diesel engine reliability. Calculating exact values of these probabilities requires relevant field tests which would allow estimating the number of state changes in the presented model of fuel combustion in engine cylinders. The procedure should consist in evaluating the numbers $n_{ij}$, within a sufficiently long time interval $[0, t]$ and obtaining realisations $T_m$, $m = 1, 2, ..., n_{ij}$ of random variables $T_m$. This would make it possible to assess the distribution functions $F_i(t)$, the probabilities $p_{ij}$, the expected values $E(T)$, and, finally, the probabilities that the combustion process stays in a given state $s_i \in S$ ($i = 1, 2, 3, 4$).

The presented model of fuel combustion process in engine work spaces (cylinders) can be taken in to account when assessing the toxicity of exhaust gases from Diesel engines and the tendency to detonation combustion. This assessment is important, as otherwise it would not be possible to formulate an opinion on pro ecological values of this type of internal combustion engines and on the probability of detonation combustion in their cylinders. Better pro ecological values are confirmed by lower probabilities ($P_i$, $i = 2, 3$), which can be considered probabilistic measures of emissions of harmful compounds contained in the exhaust gas. Low values of these probabilities mean that the emission of these compounds to the environment can be considered permissible. Moreover, a low value of the probability $P_i = P(s_i)$ suggests that the appearance of quasi-detonation combustion in this type of engines is not frequent and will not affect significantly the rate of wear of main and crankshaft bearings, and, consequently, their lifetime.


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THE EFFECT OF FUEL DOSE DIVISION ON THE EMISSION OF TOXIC COMPONENTS IN THE CAR DIESEL ENGINE EXHAUST GAS

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ABSTRACT

The article discusses the effect of fuel dose division in the Diesel engine on smoke opacity and composition of the emitted exhaust gas. The research activities reported in the article include experimental examination of a small Diesel engine with Common Rail type supply system. The tests were performed on the engine test bed equipped with an automatic data acquisition system which recorded all basic operating and control parameters of the engine, and smoke opacity and composition of the exhaust gas. The parameters measured during the engine tests also included the indicated pressure and the acoustic pressure. The tests were performed following the pre-established procedure in which 9 engine operation points were defined for three rotational speeds: 1500, 2500 and 3500 rpm, and three load levels: 25, 40 and 75 Nm. At each point, the measurements were performed for 7 different forms of fuel dose injection, which were: the undivided dose, the dose divided into two or three parts, and three different injection advance angles for the undivided dose and that divided into two parts. The discussion of the obtained results includes graphical presentation of contents of hydrocarbons, carbon oxide, and nitrogen oxides in the exhaust gas, and its smoke opacity. The presented analyses referred to two selected cases, out of nine examined engine operation points. In these cases the fuel dose was divided into three parts and injected at the factory set control parameters. The examination has revealed a significant effect of fuel dose division on the engine efficiency, and on the smoke opacity and composition of the exhaust gas, in particular the content of nitrogen oxides. Within the range of low loads and rotational speeds, dividing the fuel dose into three parts clearly improves the overall engine efficiency and significantly decreases the concentration of nitrogen oxides in the exhaust gas. Moreover, it slightly decreases the contents of hydrocarbons and carbon oxide. In the experiment the contents of nitrogen oxides markedly increased with the increasing injection advance angle for the undivided dose and that divided into two parts. This, in turn, led to the decrease of the contents of hydrocarbons and carbon oxide. Fuel dose division into two and three parts leads to the increase of smoke opacity of the exhaust gas, compared to the undivided dose.

Keywords: exhaust opacity, exhaust emissions, Diesel engine, Common Rail, fuel dose division

INTRODUCTION

A concept of multi-part fuel dose division into three or five separate parts, Multijet system for instance, see Fig.1, has nowadays become a standard solution in contemporary Diesel engine control systems. Engines which make use of the above control strategy include most popular Volvo Penta engines, series D3 to D6, with powers ranging from 100 to 300 kW, and Cummins engines, for instance QSB6.7/QSB7 for Marine, with powers from 184 to 405 kW. These engines, frequently used as basic equipment of contemporary tourist boats, are equipped with high-pressure Common Rail systems and piezoelectric injectors.

Particular injected fuel doses bear the names of: pilot-, pre-, main-, after- and post-injection. The pilot injection, which precedes the main injection by about 0.7 milliseconds, provides time needed for proper fuel mixing with air. The pre-injection decreases the delay time of the main injection in order to decrease the emission of nitrogen oxides (NO), noise, and vibration. The after-injection, taking place within milliseconds after the main injection, aims at afterburning all remaining solid particles, while the post-injection supports keeping proper temperature of the exhaust gas to ensure higher
operating efficiency of the exhaust aftertreatment system. The abovementioned Multijet concept of fuel injection comprises five injection phases, but other systems, which now gain in popularity, divide the fuel dose into 7 or 8 parts [2, 3, 4, 5, 6].

The current state of knowledge suggests that the number of injections per one engine work cycle, the amounts of particular fuel dose parts, and the delay or advance angles of the beginning of injection of particular parts, i.e. in general, the overall volume-time characteristics of the fuel injection process significantly affect the heat release and, consequently, the engine performance, as well as the contents of harmful substances in the exhaust gas, including solid particles (mainly soot), and the noise generated by the engine in operation. The results of examination presented in the article and their analyses aim at confirming this thesis.

METHODOLOGY OF EXAMINATION

The here reported examination aimed at determining the effect of fuel dose division on the exhaust gas composition and smoke opacity, and on engine operation parameters, in particular those referring to the combustion process itself and the generated noise. These issues were partially analysed in other publications by the author [7, 8].

The present examination consisted in measuring basic parameters of engine operation, the chemical composition of the exhaust gas, as well as the indicated and acoustic pressures. The tests were performed in accordance with the pre-established research procedure in which 9 engine operation points were defined for three rotational speeds: 1500, 2500 and 3500 rpm, and three load levels: 25, 40 and 75 Nm. Figure 1 shows the torque characteristic of the examined engine with the marked measuring points.

The measuring points were selected in such a way that the measurements were performed for each engine rotational speed and each load level at different modes of fuel dose division and different corrections of the injection advance angle (IAA) from the factory set values. For instance, seven tests were performed at the rotational speed equal to 1500 rpm and the load equal to 25 Nm. These tests were done, in the below given order, for the following injection characteristics:

1. **3 dose parts**, (pilot-, pre-, and main injection), basic IAA,
2. **2 dose parts**, (pre- and main injection), basic IAA,
3. **2 dose parts**, IAA correction of the main dose part to 4° before TDC,
4. **2 dose parts**, IAA correction of the main dose part to 8° before TDC,
5. **undivided dose**, basic IAA,
6. **undivided dose**, IAA correction to 4° before TDC,
7. **undivided dose**, IAA correction to 8° before TDC,

A graphical scheme or execution of particular test cases for different fuel injection strategies, discussed above, is given in Fig. 2. The injection advance angle referring to the main injection was constant and the same in tests 1, 2 and 5 (marked blue in figures and diagrams). This IAA control resulted from properties of the controller. Having into account that in tests 1 and 2 the fuel dose was divided into three and two parts, respectively, the real angle of fuel injection beginning varied. Switching off, in succession, the pilot and pre-injection phases changed the delay of the fuel injection angle. The IAA correction applied in tests 3 and 6 (marked purple) and tests 4 and 7 (marked yellow) to 4° and 8°, respectively, before TDC aimed to provide opportunities for measurements at similar real fuel injection values as in test 1 for the dose divided into three parts.

For the rotational speed equal to 1500 rpm the engine load was increased to 50 and 75 Nm, and then the measurements were performed following the above described procedure, which was also applied to the cases of rotational speed equal to 2500 rpm and 3500 rpm. At each measuring point the measured parameters were recorded after some time interval, to allow the engine parameters to stabilise.
The measured values were collected in four files. Files 1 and 2 contained data concerning the course of pressure changes in the combustion chamber, which were recorded after each 0.5 °OWK (file 1), and basic data characterising the cycle, such as the indicated pressure and the maximum pressure of the process (file 2). The next file, file 3, contained the data recorded by the code „HAMOWNIA“ which monitored the engine test bed operation, while the last file, file 4, collected the data recorded by the code monitoring the engine operation parameters.

**OBJECT OF EXAMINATION AND RESEARCH RIG**

The object of examination was a modern four-cylinder Diesel engine supplied by the Common Rail system. The cylinder head has two camshafts, and four valves in each cylinder. The valve train is driven via a chain system with direct transmission between the camshafts. The engine is charged from a turbocharger with intake air cooling system, and is equipped with an exhaust gas recirculation system.

The tests were performed in the laboratory owned by the Department of Combustion Engines and Vehicles, University of Bielsko-Biała. The engine was installed on the research rig and connected with the eddy-current brake Schenck W130. The engine and the research rig are shown in Fig. 4.

The engine motion was controlled using a laboratory Diesel engine controller (Fig. 5), which provided full control of operating parameters by proper settings and current monitoring. It also enabled to “block” certain parameters during the tests so that a change of one operating parameter did not provoke automatic correction of the remaining parameters, a situation which frequently occurs in factory controllers. The controller was built based on the experience gained in earlier research works [9, 10] done in the Division of Combustion Engines, University of Bielsko-Biała.

**RESULTS OF EXAMINATION**

The tests performed on the engine test bed in accordance with the research programme presented in Chapter 2 enabled to collect results which will be graphically illustrated in the diagrams and discussed in this Chapter. Due to a large number of performed tests (7 tests for each of 9 engine operation points, i.e. 63 tests in total), only the results obtained for two selected engine operation points (14 tests) are presented and discussed. In the selected cases the fuel dose was divided into three parts and injected at the factory set control parameters. For point 1 the rotational speed was equal to 1500 rpm and the load was equal to 25 Nm, while for point 2 they were equal to 2500 rpm and 40 Nm, respectively. These points can be found in the Multijet injection system control strategy scheme shown in Fig. 1.

Figures 6+10 show bar graphs of the exhaust gas composition, smoke opacity, and overall engine efficiency as functions of the controlled advance angle of the main injection: with the basic value approximately equal to 1° before TDC and the accelerated values equal to 4° and 8° before TDC, and the dose division characteristic for the selected two engine operation points.

Figure 6 shows the effect of fuel dose division and injection advance angle changes on the contents of nitrogen oxides (NOx) in the exhaust gas. As mentioned in the examination procedure, the value of the injection advance angle refers to the injection of the main dose part, therefore switching off successive dose parts, i.e. the pilot and pre-injection (represented by blue bars) shifts down the beginning of the combustion process, as compared to the division into three parts. Despite a much shorter injection time for the fuel dose divided into two parts and the undivided dose, no increase of the combustion dynamics is observed, which results in keeping the NOx contents at a similar level to that characteristic for the dose divided into three parts. For the rotational speed 1500 rpm, a slight increase by 6 % in the NOx content is observed, while for 2500 rpm this content decreases by 7 %.

Despite a relatively early real beginning of fuel injection, the dose division into three parts confirm mild nature of the heat emission process, which positively affects the contents of nitrogen oxides for the both analysed engine operation points. We can also conclude that for this control mode the overall engine efficiency will be higher than for the dose divided into two parts and undivided.

Increasing the injection advance angle (purple and yellow bars) intensifies the process dynamics, as a result of which the maximum pressure and temperature values also increase. As a further consequence, the NOx content in the exhaust gas...
clearly increases, and the scale of this increase is similar in the both analysed cases. The maximum NOx content amounts to 425 ppm for the undivided fuel dose, the TDC equal to 8° and the rotational speed equal to 1500 rpm, and is higher by 97% by that recorded for the fuel dose divided into three parts.

The nature of changes of the content of hydrocarbons (HC) in the exhaust gas, which can be observed in Fig. 7, is quite opposite to that observed for NOx. Switching off the pilot and pre injection phases results in significant increase of the HC content, by 87% for the rotational speed 1500 rpm and by as much as 102% for 2500 rpm. On the other hand, correcting IAA decreases the HC content. Its values obtained after correction (yellow bars) are at the level approximately higher only by 10 ± 20 % as compared to the fuel dose divided into three parts.

A similar trend to that recorded for HC can be observed when analysing the content of carbon oxide (CO) in the exhaust gas. However now, this trend is not as clear and regular as for HC. For the rotational speed 1500 rpm and the fuel dose divided into two parts, no clear trend can be seen and the recorded changes oscillate about the level of 220 ppm. For the same rotational speed and the undivided dose, the CO value for the basic IAA is higher by as much as 200% (!!!), compared to the dose divided into three parts. Smaller and more regular changes are observed for the rotational speed 2500 rpm. Here some analogy to the above discussed HC content changes can be observed. After correcting IAA to 8° before TDC, the CO values recorded for the dose divided into two and three parts were similar to each other, or even smaller when the dose was undivided.

Smoke opacity changes shown in Fig. 9 as a function of fuel dose division and IAA changes illustrate the unfavourable effect of fuel dose division on smoke opacity. This effect was clearly
observed in both analysed engine operation cases. Dividing the fuel dose into two or three parts dramatically increases smoke opacity, compared to the undivided dose. Out of the two analysed cases, the dose division into two parts looks slightly more favourable. As for IAA control, for the undivided dose its effect is not clear, which can result from very small opacity values and limited sampling accuracy of the opacimeter. For fuel dose division into two parts, increasing IAA leads to the increase in smoke opacity by 20%. The recorded values are lower by nearly 30% for the rotational speed 1500 rpm and by 36% for 2500 rpm, as compared to the fuel dose divided into three parts. For the undivided dose, the smoke opacity is nearly three times as low for the rotational speed 1500 rpm and twice as low for 2500 rpm as that recorded for the fuel dose divided into three parts.

As the reference to the analysed changes of exhaust gas composition and smoke opacity as functions of fuel dose division and IAA changes, the bar graph of engine efficiency changes as a function of the above parameters is shown in Fig. 10.

The engine supplied with the fuel dose divided into three parts reveals slightly higher overall efficiency for the two analyses engine operation points than when it is supplied with the fuel dose divided into two parts or undivided. Even increasing IAA cannot compensate the efficiency drop caused by switching off the pilot and pre-injection phases.

CONCLUSIONS

The here reported research has revealed significant effect of fuel dose division on the exhaust gas composition and engine efficiency. The analysis of the obtained results enabled to state clearly that the number of parts into which the fuel dose is divided in one engine work cycle and the injection advance angles, both parameters composing the “volume-time” characteristic of the injection process, significantly affect the combustion process and, as a further consequence, the engine performance. They also affect the contents of harmful compounds and solid particles (mainly soot) in the exhaust gas. The above statement is confirmed by conclusions resulting from the performed analysis:

- within the range of low rotational speeds and small engine loads, the engine revealed very high sensitivity to the introduced control changes, which was manifested itself in significant changes in exhaust gas composition,
- the NOx content in the exhaust gas clearly increased with the increase of the injection advance angle for the fuel dose divided into two parts, and even more for the undivided dose,
- on the other hand, increasing the injection advance angle resulted in the decrease of CO and HC contents in the exhaust gas. This enabled to reach the levels which were higher by slightly more than ten percent for higher IAA, or even lower in some cases (CO), as compared to the contents of these compounds when dividing the dose into
three parts,

- bearing in mind good oxidising properties of the catalyst, manifested by negligible CO and HC contents in the exhaust gas behind it, the optimal injection advance angle for the fuel dose divided into two parts and the undivided dose (needed at high engine rotational speeds and loads) can be selected taking into account the level of NOx content in the exhaust gas and, to a lesser degree, the smoke opacity,

- fuel dose division significantly affects the level of smoke opacity in all examined engine operation states, in particular when the fuel dose is divided into three parts. This makes basic limitation for fuel dose division at higher engine rotational speeds and loads,

- within the low speed and load regime, dividing the fuel dose into three parts clearly and positively affects the contents of all analysed compounds of the exhaust gas, and improves the overall engine efficiency, at the same time keeping the smoke opacity at a permissible level. Therefore this mode of injection process control is sensible and recommendable within the engine low speed and load operation regime.

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**POLAND**
TECHNO-ECONOMIC APPROACH TO SOLAR ENERGY SYSTEMS ONBOARD MARINE VEHICLES

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ABSTRACT

The world is facing the challenge of continuously increasing energy consumption. At the same time, the energy resources are getting scarcer. Despite a sudden significant drop of fuel prices worldwide, research activities that focus on reducing the dependence on fossil fuels as a traditional source of energy still have the preference in the field of shipping industry. The use of clean and renewable energies, such as solar energy for instance, is proposed as a method to improve the ship efficiency. Ships can get the benefits from solar energy due to the fact that most of their upper decks are always exposed to the Sun, especially in sunny water regions. The present paper discusses the effectiveness and challenges of installing solar panels for auxiliary power production on board a ship. As a case study, the research evaluates both economic and environmental benefits resulting from implementing such concept aboard a research vessel.

Keywords: Environmental issues, fuel saving, solar energy, photovoltaic cells, ship emissions

INTRODUCTION

If we look closer around us, we will find that any source of energy, even fossil fuels, is mainly based on the Sun. Plants depend on the Sun to make food, animals eat the plants, and both of them end up becoming key ingredients for fossil fuels. Without the Sun, nothing would exist on this planet (McLamb, 2011). The Sun is a remarkable source of renewable energy in the form of heat and light which can be converted into electricity. However, despite their extended use at main land applications, the presence of solar energy systems in modern marine technology remains limited. They mainly work as suppliers to small lighthouses, buoys, and chargers for batteries of small sailing yachts. However, the rising transport expenses due to fuel prices, the increasing restrictions of Carbon dioxide (CO2) and Nitrogen oxides (NOx) emissions due to new ecological policies, and, finally, the general need for more eco-friendly transportation modes are the reasons that force marine companies to re-examine systematic use of PV systems on large vessels (Kobougias et al., 2013).

The fundamental aim of this paper is to highlight various solar power units that could be utilised aboard ships. In addition, the paper discusses and assesses the effectiveness of using solar energy as an auxiliary power supply in order to reduce fuel consumption and its emissions. Furthermore, the paper studies certain challenges, as well as economic and
environmental issues concerning possible applications of this concept aboard a hydrographic surveying research vessel working in the Red Sea water region.

BENEFITS AND BARRIERS OF SOLAR ENERGY AGAINST CONVENTIONAL MARINE POWER PLANTS

From the view point of availability, most of conventional marine power plants depend on fossil fuels, which are expected to be exhausted during the few coming decades, while solar energy is considered a renewable and continuous source of energy that is supplied by the Sun to everywhere on the Earth (Seddiek and Elgohary, 2014). Undoubtedly, the conventional marine fuels have worked effectively over the past decades, especially with respect to safety, performance and adaptability. But due to huge amounts of harmful emissions from ships, strict emission regulations were introduced by the International Maritime Organization (IMO) during the last few years to overcome this problem (Seddiek, 2015). On the other side, solar energy produces no emissions, i.e., it is environmentally safe. Moreover, unlike conventional marine power plants; solar energy systems are noiseless, which is an important factor especially on passenger ships. Lastly, solar energy systems can act as an ideal power source, independent of the ship’s electromechanical systems (Hussein and Ahmed, 2014).

Despite the aforementioned benefits, low efficiency of the solar power system is a main barrier that limits its widespread use. The average thermal efficiency of a conventional marine engine is about 40 per cent, while the solar PV system efficiency is about 15 to 20 per cent (de Castro Nóbrega and Rössling, 2012). Performance restrictions on solar PV system components such as batteries, inverters, and other power conditioning systems are another barrier. Additionally, since the solar power system is always exposed to the weather, water may seep through tiny cracks in the glass of the PV unit, especially during rainy season, causing dust and algal-growth to accumulate along the lower section of the panels. All this constitutes major problems in the operation of PV systems. Furthermore, low density of the produced power makes it difficult to depend on solar power as a main source of electric power aboard ships (Mahmud et al., 2014).

ESSENTIAL REQUIREMENTS FOR A SUCCESSFUL SOLAR-POWERED SHIP

Studying the applicability of solar power systems aboard ships indicates that there are some factors that may affect this applicability. These factors include: (1) availability of high solar radiation along the ship route, (2) existence of adequate deck area exposed to the Sun, (3) availability of a suitable grid-connected PV solar power system, (4) techno-economic selection of available solar panels, and (5) scientific preparation of the system layout.

SOLAR RADIATION DISTRIBUTION

Data on the solar radiation distribution in a given water region are of vital importance when assessing the possibility of using any solar energy system as a power source in that region. Such data are prerequisites for the design of any solar power conversion system. The solar radiation received at the Earth’s surface is subject to many years of observation (possibly not less than 20 years) that must be attained in order to get a precise estimation of the long-term distribution and availability. There are many places in the developing countries that do not have necessary facilities for accurate and continuous measurements of solar radiation, hence it is necessary to use some experimental methods based on easily-measured meteorological parameters such as: relative humidity, temperature, duration of bright sunshine, rainfall periods and cloudiness (Paulescu et al., 2012). The long-term average of the annual sum of radiation worldwide ranges from 700 to 2700 kW.hr/m² while the daily sum is from 2.0 to 7.5 kW.hr/m². In particular, it is found from the radiation map of Arab countries that the annual and daily sums of radiation range from 1600 to 2400 kW.hr/m² and from 4.4 to 6.6 kW.hr/m², respectively, as shown in Figure 1.

![Figure 1: World Solar radiation distribution (Paulescu et al., 2012)](image)

SUN-EXPOSED DECK AREA

The available sun-exposed deck area aboard a ship plays a great role in the amount of absorbed solar energy. The available area is a function of several factors, such as ship type, ship dimensions, and deck machinery arrangement. The larger the number of solar panels that can be installed, the more sunlight they can collect and, consequently, the faster the solar energy is converted into electricity that can be stored in batteries. Large deck area is also important in other situations, such as cloud cover, or low-angled and low-intensity light in winter. The charging time can vary from 4 to 16 hours of sunlight for one battery, depending on the surface area and light conditions.

Solar energy can be calculated according to the geographical position, solar panel area, and solar panel efficiency. Generally, the amount of solar energy which can be absorbed is estimated
as follows:

\[
\text{Amount of Solar Energy} = ASE \times P.A \times \mu
\]  

(1)

where, \((ASE)\) is the average solar energy per unit area (kW/m²), \((P.A)\) is the solar panel area (m²), and \((\mu)\) is the solar panel efficiency.

GRID-CONNECTED PV SOLAR POWER SYSTEM

A grid-connected PV solar power system consists mainly of solar panels, inverter, battery bank, and other necessary electric devices. Figure 2 describes a simple model of a grid-connected PV system which can be installed on board a ship. As shown in the figure, the multiple solar panels 1 are connected together to make up a solar array 2 which is responsible for producing DC (direct current).

Figure 2: Simple model of grid-connected PV solar power system that can be installed on board a ship

It is noteworthy that the number of solar panels that can be used depends on the storage capacity of the used batteries. The produced DC is then transferred to the combiner box 4 via electric wires 3. The electric current flows across the disconnect switch 5 to the charge controller 6, which controls the current coming from the solar panels and prevents batteries from overcharging (SolarGIS, 2015). The electricity then goes to multiple batteries composing the battery bank 7. The batteries use and store DC. They have low voltage output, usually in the range of 12-24 volts. As most of the appliances on board ships operate on 220V AC, the inverter 7 is needed in the system to convert DC into AC. The shunt B is used to measure the electric current passing between the battery bank 7 and the inverter 9. The produced AC is supplied to the designated electric consumers 13, to provide lighting, for instance. Once the quantity of solar energy becomes low; the automatic Genset starter 10 is activated to start up the electric diesel generator 11. The selector 12 is used to switch between the current coming from the inverter and that supplied by the generator.

IMPACT OF THE WEIGHT OF SOLAR ENERGY SYSTEM ON SHIP STABILITY

There is no doubt that adding extra weight aboard the existing ship will result in certain location changes of its centre of gravity, which will probably affect its stability. It should be noted here that the added weight of the solar panels, which need to be placed at the upper decks exposed to the Sun, will result in a vertically upward shift in the ship’s centre of gravity. However, at the same time the heaviest component of the solar power system, which is the battery bank, is usually secured properly at the bottom of the ship, near the keel, which brings down the ship’s centre of gravity. All this is expected to result in negligible effect on the ship’s intact stability (Rolland, 2013; Moustafa and El-bokl, 2014; Santosa and Utama, 2014), and Raj, 2014).

POTENTIAL FOR MARINE APPLICATIONS OF SOLAR POWER SYSTEMS

In order to study the applicability of solar power system aboard a ship, a great amount of important information needs to be collected and estimated. Examples of such data that need to be collected and/or estimated include: ship’s principle particulars, deck layouts, fuel consumption, the initial, operating and maintenance costs of the diesel generator, the required solar power for feeding batteries, market prices of solar panels, among others. The consumed auxiliary power of the vessel should also be taken into consideration in order to determine the suitable number of solar panels that are capable of supplying the amount of power similar to that supplied by the marine diesel generator. In the reported project, solar panels are used to supply 24V DC power to the battery bank, from which it is used to run electronic equipment such as alarms, lights, emergency lights, radio navigational aids, navigational lights, and other emergency loads on board the ship. The collected data are used to analyse the cost of the implemented solar power system and then compare it to the cost of the consumed fuel from the Gensets. The environmental impact of the implemented solar power system is also analysed.

ECONOMIC ANALYSIS

The economic aspect is one of the main indicators that are to be taken as the evidence of possible benefits from installing the solar power system aboard the ship as auxiliary power supply for some electrical instruments in a way to reduce the fossil fuel consumption and hence save its cost and avoid emissions. The annual net cost savings resulting from implementing the solar power aboard the ship can be estimated from the following formula:

\[
\text{Annual Net Cost Savings} = \text{AAEPC} - \text{AAC}
\]

The AAEPC is the on board Annual Auxiliary Engine Power generation Cost for the same electric load, either during sailing or at berth. The AAEPC (in US$/year) is the sum of fuel cost, maintenance cost, and operating cost, and it is calculated from the following equation:

\[
\text{AAEPC} = \text{P}_{\text{aux}} \times t \times c_{\text{f}} \times f_{\text{c}} \times 10^{-6} + \sum_{d=1}^{n} c_{d} M_{d} + \sum_{b=1}^{m} c_{b} O_{b}
\]  

(3)

where, \(P_{\text{aux}}\) is the Diesel engine power (kW), \(t\) is the time of connection to batteries (hr/year), \(c_{\text{f}}\) is the specific Diesel fuel oil consumption (g/kW.hr), \(f_{\text{c}}\) is the Diesel fuel oil cost (US$/tonne), \(C_{d}M_{d}\) is the auxiliary onboard engine maintenance cost (US$/year), and \(C_{b}O_{b}\) is the auxiliary onboard engine operating
cost (US$/year). Symbols y and w refer to maintenance and operating items, respectively.

The AAC is the Annual Average Cost of the implemented solar energy system, which can be estimated from the following equation:

$$AAC = TUC \times CRF$$

where TUC is the Total Unit Cost of the solar energy system (US$), and CRF is the Capital Recovery Factor, given by:

$$CRF = \frac{i(1+i)^N}{(1+i)^N - 1}$$

Here, N is the expected number of years of ship operation after implementing the solar panel system, and i is the annual interest rate.

The [TUC] of any solar energy system depends on three main variables, which are: the initial cost, the operating cost, and the maintenance cost. The initial cost includes, in addition to the purchase and installation cost of the system, various sub-systems necessary for its effective operation. The operating and maintenance costs are difficult to quantify because they depend on a large number of variables, such as local labour cost, life-time of the equipment, and operating time. Based on the data collected from actual projects utilising the same solar panels, and on quotations obtained from solar panel manufacturers; the value of initial, operating and maintenance costs can be estimated from the following equation (Moustafa and El-bokl, 2014); (Sulaiman et al., 2011); (Spagnolo et al., 2012), and (Ren et al., 2013):

$$TUC = SP_{CC} \cdot N_m \left[ 1 + \text{ins}_{sp} \right] + \sum_{i=1}^{N_{PG}} P_G + \sum_{j=1}^{N_{O&M}} O&M$$

where, $SP_{CC}$ is the cost of one solar panel in (US$); $N_m$ is the number of solar panel modules; $\text{ins}_{sp}$ is the installation cost percentage; $P_G$ is the power grid system cost in (US$); $O&M$ is the operating and maintenance cost in (US$); $i$ is the total number of components of the power grid system, and $j$ is the total number of various operating and maintenance cost items.

The costs of both the solar array and the power grid need to be estimated separately, because they are independent of the system specification, as follows:

Estimation of solar array cost

The cost of solar array depends mainly on the number of solar panels ($N_m$). This work presents two different methods for the calculation of the number of solar panels, which depend on the stage of applying solar energy concept aboard. In case of new build ships, $N_m$ is proportional to the required electric load to be covered. The calculations presented in (Soufi et al., 2013), and (Wang and Nehrir, 2008) can be used to determine the number of solar modules which will be able to provide the required electricity, as follows:

$$N_m = \frac{E_t \cdot \text{PSI} \cdot \mu_{PV}}{G_{av} \cdot TCF \cdot \mu_{T} \cdot \mu_{PV}}$$

where, $E_t$ is the average daily required electric load in (Watt/day), PSI is the peak solar intensity at the Earth’s surface (kW/m²), $\mu_{PV}$ is the PV (panel) efficiency, which is about 0.15 - 0.20 per cent (Kobougias et al., 2013), and (Mahmud et al., 2014), $G_{av}$ is the average solar energy input per day, $TCF$ is the temperature correction factor, which is about 0.8 (Soufi et al., 2013), $PV_{pp}$ is the maximum power of the selected solar panel, $\mu_{T}$ is the overall solar panel efficiency, $\mu_{inv}$ is the inverter efficiency, which is up to 0.8 - 0.85 per cent (Ahmad and Khan, 2012), and $\mu_{b}$ is the battery efficiency, which is about 70 - 80 per cent (Patil et al., 2013).

For the already existing ships, $N_m$ depends on the available area exposed to the Sun and suitable for solar array installation ($A_{av}$) in (m²) as well as on the selected solar panel area available in the market ($A_{sp}$) in (m²). It can be calculated as follows:

$$N_m = \frac{A_{av}}{A_{sp}}$$

Consequently, the electric load which can be provided by the solar energy system ($E_{SE}$) in (Watt.hr/day) may be formulated as follows:

$$E_{SE} = PV_{pp} \cdot N_m \cdot \mu_{B} \cdot \mu_{inv} \cdot T_S$$

where, $T_S$ is the average time during which the solar array is expected to be subjected to solar energy.

Estimation of power grid cost

The cost of power grid system includes the cost of batteries, inverter, and charge controller. The cost of batteries can be estimated as follows:

$$\text{Batteries cost} = N_B \cdot B_C$$

where $B_C$ is the cost of one battery, $N_B$ is the total number of batteries needed to supply the required current, which may be calculated as presented in (Moustafa and El-bokl, 2014), and (Bhatt and Verma, 2014) as follows:

$$N_B = \frac{E_t \cdot N_d}{V_B \cdot B_{cap} \cdot \text{DoD}}$$

Here $E_t$ is the electric load provided by the solar system depending on the ship’s situation, i.e. already existing or newly built, $N_d$ is the expected number of days without sunshine, $V_B$ in (volts) and $B_{cap}$ in (A.hr) are the voltage and capacity of the selected battery, respectively, and DoD is the battery depth of discharge. The costs of inverter and charge controller are taken as specific power in (US$/Watt) (Hussein and Ahmed, 2014), and (Moustafa and El-bokl, 2014).

Finally, the annual operation and maintenance (O&M) cost of the system can be either estimated according to data collected from the established units, companies and catalogues, or taken as a percentage of the capital cost and interest charge (Bhatt and Verma, 2014).
ENVIRONMENTAL ANALYSIS

The amount of emissions released from the ship’s diesel generator depends mainly on the emission factors of the used fuel, which vary largely among different engines and fuels. On the other side, the emissions released by the Solar PV system are considered equal to zero as there is no burned fuel. To evaluate gains of using the concept of Solar PV system as a green alternative to the auxiliary diesel engines, it is very important to estimate the quantity of the exhaust gases emitted from the auxiliary diesel engines. These amounts are estimated during the ship’s sailing and berthing. The quantity of emissions depends mainly on the main emission factors, the consumed electric load, and the working hours. Table 1 summarizes the main emission factors of the diesel generator (Seddiek et al., 2013).

Table 1: Summary of main emissions factors of Diesel generator

<table>
<thead>
<tr>
<th>Emission Gases</th>
<th>CO₂</th>
<th>CO</th>
<th>NOₓ</th>
<th>PM₁₀</th>
<th>SOₓ</th>
<th>HC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Emission Factor (g/kW, h)</td>
<td>698</td>
<td>1.68</td>
<td>13.43</td>
<td>0.55</td>
<td>2.56</td>
<td>0.53</td>
</tr>
</tbody>
</table>

As the solar energy system has zero emissions, the amount of emissions that can be deducted as a result of implementing the solar energy system aboard ships (E_deduct) can be expressed as follows:

\[
E_{\text{deduct}} = E_f \times t_s \times P
\]  
(13)

where, \(E_f\) is the emission factor in (g/kWh), and \(P\) is the power saved when using the solar power system in (kW).

CASE STUDY

To demonstrate the use of the abovementioned equations to study the feasibility of implementing the solar energy aboard an existing vessel, the authors have selected the research boat “Al-Azizi” owned by King Abdulaziz University and working for hydrographic surveying in the Red Sea water region. The boat has a flat-top deck structure that increases the area available for installing the PV solar array. Figure 3 shows a photo of this boat, while Table 2 collates its characteristics.

SELECTED SOLAR SYSTEM

As the present case study refers to the existing vessel; the amount of electric load provided by the solar system using multi-crystal PV solar panel model (KD260GX-LFB2) can be estimated using the deck area exposed to the Sun which is available for solar array installation aboard the vessel. The specifications of the selected solar panel are shown in Table 3.

Table 2: Characteristics of the research boat

<table>
<thead>
<tr>
<th>Ship Type</th>
<th>Research Boat</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum speed</td>
<td>14 knots</td>
</tr>
<tr>
<td>Transit speed (medium load)</td>
<td>12 knots</td>
</tr>
<tr>
<td>Length overall</td>
<td>45.5 m</td>
</tr>
<tr>
<td>Beam</td>
<td>9.5 m</td>
</tr>
<tr>
<td>Draft at full load</td>
<td>3.2 m</td>
</tr>
<tr>
<td>Depth</td>
<td>3.8 m</td>
</tr>
<tr>
<td>Main generator</td>
<td>2 x Cummins QSM11</td>
</tr>
<tr>
<td>Average electrical power required during the cruise</td>
<td>21 kW</td>
</tr>
<tr>
<td>Average electrical power required during the berthing</td>
<td>5 kW</td>
</tr>
<tr>
<td>Area available for solar array installation (A_{ext})</td>
<td>90 m²</td>
</tr>
<tr>
<td>Average time of exposure to the Sun</td>
<td>8 hrs</td>
</tr>
</tbody>
</table>

Table 3: Solar panel specifications

<table>
<thead>
<tr>
<th>Model</th>
<th>KD260GX-LFB2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Size</td>
<td>166* 99* 4.6 cm</td>
</tr>
<tr>
<td>Weight</td>
<td>20 kg</td>
</tr>
<tr>
<td>Power rating (P_{pp})</td>
<td>260 Watt</td>
</tr>
<tr>
<td>Open circuit voltage (V_{oc})</td>
<td>38.3 volt</td>
</tr>
<tr>
<td>Short circuit current (I_{sc})</td>
<td>9.09 Amps</td>
</tr>
<tr>
<td>Voltage at P_{max} (V_{mp})</td>
<td>31 volt</td>
</tr>
<tr>
<td>Current at P_{max} (I_{mp})</td>
<td>8.39 Amps</td>
</tr>
<tr>
<td>Solar panel cost</td>
<td>322 USS</td>
</tr>
</tbody>
</table>

The data related to the selected battery, inverter, and charge controller are listed in Table 4.

The data collected from the hydrographic surveying research vessel regarding the solar system and the fuel prices have been used to estimate both the annual solar system installation cost (AAC) and the annual operating cost of the auxiliary engine, as shown in Figure 4. Giving a period of about 10 years as a life-cycle of the solar energy system, the installation cost is expected to be recovered within the first 3 years, which is a short period compared to the expected ship age.
Table 4: Characteristics of battery, inverter, and charge controller

<table>
<thead>
<tr>
<th>Item</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>B_cap</td>
<td>300 Ah</td>
</tr>
<tr>
<td>N_2</td>
<td>2</td>
</tr>
<tr>
<td>DoD</td>
<td>0.8</td>
</tr>
<tr>
<td>V_B</td>
<td>12</td>
</tr>
<tr>
<td>M&amp;O</td>
<td>0.5% of total</td>
</tr>
<tr>
<td></td>
<td>cost</td>
</tr>
<tr>
<td>inscp</td>
<td>10%</td>
</tr>
</tbody>
</table>

Figure 4: Installation and Operating Costs

Figure 5 presents several scenarios considered in the economic analysis in the case of applying the proposed system on board the research boat "Al-Azizi". It is clear that due to the low cost of fuel inside the Kingdom of Saudi Arabia, the money back period will be longer than if it is applied in another country with higher fuel prices. For a period of 10 working years, the proposed system can achieve a fuel saving cost of about 18500 and 5600 ($/year) in case of international and local fuel prices, respectively.

Figure 5: Various scenarios demonstrated in the economic analysis

Figure 6 shows the annual quantities of emission reduction when applying the solar energy concept aboard the research boat. The results show that there is a possibility of achieving a total reduction in ship's emissions of about 21.5 tonnes per year.

Figure 6: Environmental Benefits of Solar Concept

This amount is valuable in terms of ship's emissions reduction, especially with the new legislations issued by the International Maritime Organization, aiming to apply either Energy Efficiency Design Index (EEDI) or Ship Energy Efficiency Management Plan (SEEMP) for eliminating greenhouse gases. It should be noted that the results regarding both economic and environmental benefits, which are presented in this work, can vary from one boat to another or from one country to another, but there is no doubt that each time the results will be eco-friendly.

CONCLUSIONS

Besides its domestic uses, solar power can be utilised to supplement the auxiliary power usually produced by the boat's Diesel generator. As a consequence, it will be capable of minimising both air pollution and fuel cost. The PV solar panel appears to be a suitable solar power unit for boats. The present paper shows that until now, there are some challenges that need to be overcome before integrating solar energy on a ship. The sufficient area of the deck space exposed to the Sun is believed to be a major constraint. In addition, more attention should be paid to increase the durability of the PV solar panel through increasing its efficiency and decreasing its cost (perhaps by mass production). As a numerical study, the research boat "Al-Azizi", owned by King Abdulaziz University and working in the Red Sea water region, was selected as a case study to demonstrate the benefits of the solar power system. The study has revealed that, economically for a period of 10 working years, the proposed system can achieve a fuel saving cost of about 18500 and 5600 ($/year) in case of international and local fuel prices, respectively. Moreover, the present paper shows that the use of solar energy to contribute to the total energy supply on a ship is a smart beginning for designing an environmentally friendly ship. The results show that there is a possibility of achieving a total reduction in ship's emissions...
of about 21.5 tonnes per year. Finally, despite the prediction of constant fuel prices at a low level, the world’s concern regarding ship emissions will accelerate the process of using solar power systems on-board ships in the near future, especially with the new legislations issued by the International Maritime Organization.

ACKNOWLEDGEMENT

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MEASUREMENT AND ANALYSIS OF COASTAL WAVES ALONG THE NORTH SEA AREA OF CHINA

Shu-zheng SUN
Hui LI
Hui SUN
College of Shipbuilding Engineering, Harbin Engineering University, China

ABSTRACT

Taking advantage of coastal wave environment to carry out scaled ship model test is an effective testing technology for ship performance. In this paper, the method of spectral analysis is adopted to calculate the significant wave height, period, wave spectrum and some other parameters of some places along the North Sea area of China. The measured wave spectrum and the ocean spectrum are handled into non-dimensional form to evaluate their similarity. The influence of wind direction and tide on coastal waves was analyzed. And the results indicate that the coastal wave spectrum is similar to the ocean spectrum under some specific conditions.

Keywords: coastal wave, measurement, buoy, spectral analysis

INTRODUCTION

Sea wave research has significance on marine military activities, the design of marine structures, the construction of marine projects, marine development, shipping transportation, marine fishing, etc. One important method of sea wave research is to measure actual sea wave environments. Wave observation technologies have developed rapidly in recent years, with a great variety of wave observation devices having been invented according to different working principles, such as wave pole-type, pressure-type, acoustics-type, gravity-type, remote-sensing wave device, etc. Users can employ different devices according to different demands so as to be adapted to specific sea situations [1].

Compared with environments of distant oceans, inshore waters are relative shallow with relative small waves, which makes the waters good for carrying out scaled self-propelled testing of ship models. Meanwhile, with great influence caused by meteorology, tides, etc., sea wave environments in inshore waters have more uncertain factors. Therefore, measuring actual environment of inshore sea waves, adopting spectral analysis method to calculate sea wave spectrum and to make statistics on wave elements, discussing the similarity between sea wave spectrum in inshore covert waters and actual ocean spectrum, and making research on the environmental characteristics of wind and wave in inshore waters, will have significance on utilizing inshore waters to carry out scaled ship model testing as well as research on ship adaptability to environment which are under actual sea wave environment [2].

This article carries out measurement on a number of inshore sea wave environments of the Yellow Sea and the Bohai Sea by utilizing buoy-typed wave height gauge, makes analysis on the measured sea wave data by adopting spectral analysis method, acquires statistics of wave elements, explores the similarity of actual wave spectrum and ocean spectrum, and makes
analysis on the influence of environmental conditions on sea wave elements, such as water depth, meteorology and tides.

ACTUAL WAVE MEASUREMENT AND SPECTRUM ANALYSIS METHODS

THE DESIGN OF WAVE HEIGHT METER

This article utilizes a self-made buoy-typed wave height meter to carry out wave measurement. The gauge is small, lightweight, low in cost, convenient for carrying, and easy for operating. The wave height meter adopts the principle of acceleration of gravity to conduct the ocean wave measurement. When the ocean wave performs a porpoising with the change of the wave surface, the vertical acceleration sensor installed in the buoy will output a time history signal which reflects the accelerated speed of the porpoising of the wave surface. After conducting a Fourier transform and a quadratic integral processing on the signal, we can get various kinds of statistical height values of the wave surface and its corresponding period of the wave [3, 4].

The buoy has a spherical overall shape, which synthetically considers its resistance to capsizing and its motion response. The wave height meter chooses a ball of 0.4m diameter as the buoy body of it. The material of the body is fiberglass, while the material of the hatch cover and the hatch flange is stainless steel. Install the acceleration sensor at the center-of-gravity position of the body, install the ballast and fix it at the bottom, and keep the waterline at the maximum-section area, where the rubber damper should be installed. The design plan of the wave height meter is shown as Figure 1 and Figure 2. The parameters of the acceleration sensor are shown in Table 1.

![Fig.1 Plan of the wave height meter](image1)

![Fig.2 Photo of the wave height meter](image2)

<table>
<thead>
<tr>
<th>Type</th>
<th>Acceleration Range [g]</th>
<th>Frequency Response [Hz]</th>
<th>Supply Voltage [V]</th>
</tr>
</thead>
<tbody>
<tr>
<td>KISTLER 8303A2</td>
<td>-2~+2</td>
<td>0-150</td>
<td>9~20</td>
</tr>
</tbody>
</table>

Conversion:1g = 9.80665m/s²

ESTIMATING THE OCEAN WAVE SPECTRUM WITH THE CORRELATION FUNCTION METHOD

This paper uses the correlation function method to estimate the ocean wave spectrum. This method firstly calculates the correlation function and obtains the sampling interval $\Delta t$ ($\Delta t = 0.5 \sim 2$). The correlation function is

$$R(\nu \Delta t) = \frac{1}{N-\nu} \sum_{n=0}^{N-1} x(t_n + \nu \Delta t)x(t_n)$$

$$r = \nu \Delta t, \nu = 0,1,2,\cdots,m$$

And then it will estimate the rough value of spectrum, which is defined $L_n$ as the rough value of spectrum corresponding to the frequency $f_n$

$$L_n = \frac{2 \Delta t}{\pi} \left[ \frac{1}{2} R(0) + \sum_{\nu=1}^{\infty} R(\nu \Delta t) \cos \frac{\pi \nu}{m} \right]$$

$$+ \frac{1}{2} R(m \Delta t) \cos \frac{\pi n}{m}$$

Finally it will smooth the rough value of spectrum, making different $R(\nu \Delta t)$ have different weights. It usually adopts the Hamming smooth processing, and the spectrum value after smooth processing is

$$S(\omega_n) = 0.23L_n^{m-1} + 0.54L_n + 0.23L_{n+1}$$

When calculating the wave height based on the ocean wave spectrum, the rth order spectral moments should be calculated first

$$m_r = \sigma^2 = \int_0^{\infty} \omega^r S(\omega) d\omega$$

The significant wave height and the average period are respectively

$$H = 4\sqrt{m_0}, T = 2\pi \sqrt{m_0/m_2}$$
Common Ocean Spectrum Formula

(1) ITTC Spectrum

\[ S_f(\omega) = \frac{A}{\omega_0^2} e^{-\frac{g^2}{\omega^2 \omega_0^2}} \]  

Here, when \( A = 8.10 \times 10^{-3} \), \( B = 3.111(h_{1/3})^2 \), it is the ITTC single parameter spectrum; and when \( A = 173(h_{1/3})^2 / T_0^4 \), \( B = 691 / T_0^4 \), \( T_0 = 2\pi m_0 / \omega_0 \), it is the ITTC dual parameter spectrum.

(2) China Sea Spectrum

\[ S_f(\omega) = \frac{0.74}{\omega^2} \exp \left[ -\frac{g^2}{U^2 \omega^2} \right] \]  

Here, \( \omega \) — wave frequency; \( U \) — velocity of wind, \( g = 9.81 \text{m/s}^2 \). If the significant wave height \( h_{1/3} \) is known, the velocity of wind \( U \) can be calculated by:

\[ U = 6.28 \sqrt{h_{1/3}} \]  

(3) Ochi-Hubble Six-parameter Spectrum

Some of the actual wave spectra often show the bimodal shape, and sometimes a third peak will appear, which is much smaller than the other two peaks. Ochi and Hubble proposed a six-parameter spectrum formula, which divided the spectrum into two parts: low frequency and high frequency, with each part using three parameters—signified in the forms of significant wave height \( H_s \), spectral peak frequency \( \omega_m \) and shape parameter \( \lambda \), namely,

\[ S_f(\omega) = \frac{1}{4} \sum_{\lambda} \left[ \left( \frac{4 \lambda + 1}{4} \right)^{\partial_m^4 / 4 \lambda^4} \right] \Gamma(\lambda) \exp \left[ -\left( \frac{4 \lambda + 1}{4} \right)^{\partial_m^4 / 4 \lambda^4} \right] \]  

Where, 1, 2 respectively represent the low frequency and high frequency in the formula. In the formula, there are totally six parameters \( H_s, H_{1/2}, \omega_m, \omega_m, \lambda, \lambda \), which according to the measured spectral forms can make the difference value between the theoretical spectrum and the experimental spectrum smallest, namely, with each change of the parameter [5-8].

THE SIMILARITY ANALYSIS OF MEASURED SPECTRUM AND OCEAN SPECTRUM

When discussing the problems that whether the wave spectrum in the covered coastal water and the actual ocean are similar or not, they are often transformed into the problems that whether the spectral shapes and the main characteristics of the two are similar or not. We make a nondimensional comparison between the wave spectrum in the measurement sea area and the ocean spectrum, the nondimensional number of spectrum adopts the wave elements as the parameters, usually expressing with \( S(\omega) \cdot \omega / m_0 \) or \( S(\omega) \cdot \omega_m / m_0 \), among which \( \omega \) is the average circular frequency of spectrum, \( \omega_m \) is the peak frequency and \( m_0 \) is the spectral area [9].

The manuscript should be written in English in a clear, direct and active style. All pages must be numbered sequentially, facilitating in the reviewing and editing of the manuscript.

ANALYSIS OF TEST DATA

Before actual ocean wave measurement, the wave height meter needs to be calibrated. The calibration is carried out in the ship model towing tank in Harbin Engineering University. The irregular wave theoretical spectrum generated by the wave maker is well consistent with the wave spectrum measured by the wave height meter.

The experiment group measures the coastwise wave at Rongcheng Jinghaiwei, Dalian Xiaoheishi, Dalian Xiaoqing Island, Huludao Harbor and other places in the Yellow Sea and the Bohai Sea. The wave data are measured by the above-mentioned buoy wave height meter and are collected every 0.05 second. Each collection lasts for more than 10 minutes and the sampling frequency is 20Hz. Information like the tide, the weather condition, the distance from the measuring spot to the coast and the water depth are recorded synchronously. The map of the sea area being measured is shown in Figure 3. Like the acceleration time duration of wave surface heaving was shown in Figure 4, and Figure5 shown is the process of releasing the buoy from the boat.

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

Figure 6 shows the wave spectra of Huludao sea area at the high water period. On the very day when the test is carried out at Huludao Harbor (40° 43’ N, 121° 00’ E), the parameters of the tide, wind and water depth, and etc., are as follows:

On the very day, the time of tide: 01:05 with a height of 76cm; 9:39, 286cm; 16:02, 80cm. The tidal datum plane is 158cm beneath the average sea level, in the time zone: GMT+8; There is a southerly wind above the sea, with a wind speed of about 5m/s; and a water depth of about 8 meters in the test sea area. The significant wave heights \( H_{1/3} \) of the three groups of wave data recorded are respectively 0.14m, 0.153m and 0.161m.
Figure 7 indicates the wave spectra of the coastal flow phase at Xiaoping Island sea area in Dalian City. On the very day when the test is carried out at Xiaoping Island (38° 49´ N, 121° 32´ E ), the parameters of the tide, wind and water depth, and etc. , are as follows:

On that day, the time of tide: 6:33 with a height of 51cm; 11:57, 323cm; 18:33, 46cm. The tidal datum plane is 163cm beneath the average sea level, in the time zone: GMT+8; with a southerly wind above the sea, and a wind speed of about 4m/s; and with a water depth of about 10 meters in the tested sea area waters. The statistical significant wave heights (H_{1/3}) are respectively 0.279m, 0.264m, and 0.251m.

Figure 8 indicates the wave spectra of the coastal flow phase at Xiaoheishi sea area in Dalian City. On the very day when the test is carried out at Xiaoheishi sea area (38° 51´ N, 121° 15´ E), the parameters of the tide, wind and water depth, and etc., are as follows:

On that day, the time of tide: 2:12 with a height of 100cm; 13:48, 231cm; 19:36, 121cm. The tidal datum plane is 145cm beneath the average sea level, in the time zone: GMT+8; with a southerly wind above the sea, and a wind speed of about 4m/s; and with a water depth of about 10 meters in the tested sea area waters. The statistical significant wave heights (H_{1/3}) are respectively 0.31m, 0.24m.

Figure 9 indicates the wave spectra of the coastal flow phase at Jinghaiwei sea area in Rongcheng City. On the very day when the test is carried out at Jinghaiwei sea area (37° 10´ N, 122° 25´ E), the parameters of the tide, wind and water depth, and etc. , are as follows:

On that day, the time of tide: 6:56 with a height of 61cm; 12:46, 236cm; 19:02, 66cm. The tidal datum plane is 151cm beneath the average sea level, in the time zone: GMT+8; with a northernly wind above the sea, and a wind speed of about 5m/s; and with a water depth of about 15 meters in the tested sea area waters. The statistical significant wave heights (H_{1/3}) are respectively 0.56m, 0.49m.
Figure 10 shows the nondimensional spectrum findings of measured ocean wave and some theoretical ocean wave spectrums.

(a)

(b)
From Figure 10, we know that:

1. According to the central frequency, frequency range, energy distribution, the situation of single-peak and multi-peak of the measurement spectrum, an appropriate ocean wave spectrum that is similar to the measurement spectrum can be found, which illustrates that the coastal ocean wave spectrums of the Yellow Sea and the Bohai Sea are similar to the ocean wave spectrum;

2. The central frequencies of ocean wave spectrum in different sea areas of the Yellow Sea and the Bohai Sea differ greatly and are similar to different ocean spectrums, which are mainly similar to ITTC single parameter spectrum and China sea spectrums, while some Double-peaked spectrums caused by multi wind fields are generally similar to Ochi-Hubble spectrums.

CONCLUSION

Buoy wave height meter which is made by adopting the gravity acceleration principle can effectively measure sea wave; the measurement and analysis of sea wave in coastal waters of the Yellow Sea and Bohai Sea show that conditions including weather and tide have significant effect on coastal wave parameters. Under suitable environmental conditions, features of wave spectra in covered coastal waters are similar to those of ocean wave spectra.

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EXPERIMENTAL AND SIMULATIVE STUDY ON ACCUMULATOR FUNCTION IN THE PROCESS OF WAVE ENERGY CONVERSION

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Mechanical Engineering Institute, Shandong University, Jinan, Shandong, China

ABSTRACT

In this paper, a floating-buoy wave energy converter using hydrostatic transmission system is studied. The entire work progress of wave energy power generation device is introduced, and the hydraulic transmission principles are emphasized through the simulation to verify the feasibility of design principle of hydraulic transmission system. The mathematical model of the accumulator is established and applied in the AMEsim simulation. The simulation results show that the accumulator plays an important role in the wave power hydraulic transmission system and that the correct configuration of accumulator parameters can improve the rapidity and stability of the system work. Experimental results are compared with the simulation results to validate the correctness of the simulation results. This would provide a valuable reference to the optimal design of wave power generation.

Keywords: wave energy converter, AMESim simulation, accumulator, experimental verification

INTRODUCTION

With the development of science and technology, and the increase in the level of productivity, the human demand for energy grows daily thus causing the intense consumption of non-renewable energy, such as petroleum, natural gas and coal resources, and resources have become increasingly depleted[1]. Fossil energy consumption also caused a straight rise in carbon dioxide emissions. In order to ease the pressure on the energy crisis, it is imperative to carry out new energy research to develop and utilize clean energy [2]. It is urgent to establish an energy structure dominated by clean, renewable energy and gradually replace fossil fuel based energy structure. The utilization of renewable energy is related to human survival and the future of mankind [3]. Wave energy technology is a useful solution to solve energy and environmental issues. The hydraulic transmission is one of the important components [4].

Wave energy power generation technology is through the wave energy absorption and conversion device to convert the wave energy into electric energy [5]. Marine technology is becoming outdated, which shows a good prospect for the full use of the ocean in the new century. But so far, the use of it also faces many challenges. The output of the ocean energy generator is unstable, and the change of the wave can cause the mechanical power of the energy absorbing mechanism to change. The output power of the generator then fluctuates and the power quality is reduced. Therefore, the improvement of the stability of the output power quality of the generator has become an important problem in wave power generation.
Usage of the hydraulic drive system to replace the traditional mechanical transmission system can reduce the energy fluctuation caused by the irregular change of front end input [7]. The ability to control the hydraulic oil as the transmission medium is greater than gear boxes and other mechanical transmission device because it is a flexible medium, which easily realizes the stepless speed change to ensure generators to maintain in the vicinity of the synchronous speed to further improve the stability of power system.

At present, most wave energy converters use hydraulic energy transmission, so the hydraulic system has great promise in the research of wave energy converters [8]. A series of studies of hydraulic power take-off in different wave energy converter models have been conducted by Bjarte-Larsson and Falnes (2006) [9], Hals et al. (2007) [10], Yang et al. (2009) [11] and others. Yang, Hals and Moan [12] studied on a heaving-buoy wave energy converter equipped with a high-pressure hydraulic power take-off machine. The mathematical model was designed to investigate the wear damage in the hydraulic machine. Falcão [13] focused a study on oscillating-body wave energy converters with hydraulic power take-off and gas accumulator. The results show that the hydraulic system with accumulators can better transform the wave energy with a stable output power. However, there is no perfect solution to stabilize the output power [14], and the comparison analysis of semi-physical experiment to simulation results is not fully expressed.

In this paper, study on the output pressure and voltage characteristics of power generation system and the response characteristics and stability of the energy conversion system with the effect of the accumulator. The first section introduces the working principle and the hydraulic transmission system of wave energy converter. The second section describes the construction of the AMESim simulation model based on the design principle, the simulation parameters are established by the theoretical calculation, and the simulation results show that the hydraulic system with accumulator can make the output power stable [15].

In the third section, according to the simulation analysis results, the semi-physical experiments are carried out and compared with the simulation results. This paper further demonstrates that the accumulator can improve the power output characteristics of the machine greatly, and the correct configuration parameter of storage can improve the stability and rapidity of output energy [16].

Fig. 1 is the new type of wave energy power generation device. The device adopts the hydraulic transmission system, which can convert the unstable wave energy into a stable power output. This paper takes the hydraulic conversion system as the research object, which is the core technology used in the wave energy power generation system.

**WORK PRINCIPLE AND COMPOSITION OF WAVE ENERGY CONVERTER**

The whole wave energy generating device is composed of a wave energy collection system, hydraulic transmission system, energy storage system, power output and data acquisition system. Fig. 2 is the schematic diagram of floating wave power generation device.

The wave energy collection system can convert the wave energy into mechanical energy of float. The movement of float can convert the mechanical energy to hydraulic energy, which travels through the hydraulic system and drives the motor to rotate, thus driving the generator to work and output power stably.

**HYDRAULIC TRANSMISSION PRINCIPLE**

The hydraulic drive system is the most important part in ensuring the output power of device, which can buffer the impact of the irregular wave energy. Fig. 3 is the working principle diagram of the hydraulic transmission system through the hydraulic cylinder of double rod to improve the output efficiency of hydraulic transmission system. It can ensure that the hydraulic energy discharges continuously whether the float movement is upward or downward. The hydraulic rectification module is composed of the one-way valve to ensure that the hydraulic output is in the same direction, so that the motor rotates in the same direction always. By matching
the parameters of each part of the hydraulic drive system reasonably can guarantee the output power more stable.

**SIMULATION AND ANALYSIS**

**THE ESTABLISHMENT OF MODEL**

Fig. 4 is the simulation model established using AMESim software according to the working principle of the hydraulic transmission system. By setting the amplitude and period of the input signal, movement state of float under different wave conditions can be analogized.

![Simulation model](Image)

**Fig. 4. Simulation model**

The main parameters of the model are set up as shown in Tab. 1:

*Tab. 1. Main parameters of the model*

<table>
<thead>
<tr>
<th>Main model</th>
<th>Settings parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>The total weight of the Moving part of float</td>
<td>15000 kg</td>
</tr>
<tr>
<td>Piston diameter</td>
<td>180 mm</td>
</tr>
<tr>
<td>Diameter of piston pole</td>
<td>140 mm</td>
</tr>
<tr>
<td>Hydraulic cylinder stroke</td>
<td>5 m</td>
</tr>
<tr>
<td>Hydraulic motor</td>
<td>speed: 1000 rev/min; displacement: 160 mL/r</td>
</tr>
<tr>
<td>Electrical generators</td>
<td>three-phase permanent synchronous generators</td>
</tr>
<tr>
<td>Pistons</td>
<td></td>
</tr>
<tr>
<td>Dampers</td>
<td></td>
</tr>
<tr>
<td>Barrels</td>
<td></td>
</tr>
<tr>
<td>Hydraulic fluid</td>
<td></td>
</tr>
<tr>
<td>Flow sensor</td>
<td></td>
</tr>
<tr>
<td>Pressure sensor</td>
<td></td>
</tr>
<tr>
<td>Speed regulator valve</td>
<td></td>
</tr>
<tr>
<td>Accumulator</td>
<td></td>
</tr>
<tr>
<td>Relief valve</td>
<td></td>
</tr>
<tr>
<td>Voltage sensor</td>
<td></td>
</tr>
<tr>
<td>Oil tank</td>
<td></td>
</tr>
</tbody>
</table>

**VALIDATION OF SIMULATION MODEL**

The simulation system is operated to verify whether the built simulation model is correct. Fig. 5 is the error contrast between the actual motion curve and the desired curve.

![Expected curve vs Actual curve](Image)

**Fig. 5. Contrast displacement curve of float and the expected value**

Fig. 5 suggests when the system started to run the oil pressure was not established, so the displacement of float is not demonstrated in the target displacement. After one cycle, the float displacement curve is in conformity with the expectation curve. It executes the sinusoidal movement according to the amplitude of 5 m cycle 4 s schedules. The trajectories of float in the allowable error range that verifies the establishment of the simulation model is correct and can be used to further explore the role of each part of the hydraulic system and study the effect of different parameters on the system output stability and velocity.

**THE SIMULATION ANALYSIS OF ACCUMULATOR**

The accumulator has the effect of peak clipping and valley filling in hydraulic transmission systems, and it can cushion the impact of irregular wave energy, as well as guarantee the stability of hydraulic transmission. It plays a key role in stabilizing the output power of the system [17]. First, to explore the working state effect of the accumulator parameters for the system, a mathematic model of the accumulator is built, and the bottom area of the wallet is $A_s$. The quality of the oil into the oil chamber is $m_o$. Ignore the elastic modulus of hydraulic oil the force analysis equation of hydraulic oil in the oil chamber is [18]:

$$
\left(p - p_o\right)A_s = \frac{m_o \frac{dV_o}{dt} + B_a \frac{dV_a}{dt} + C_a \frac{dV_a}{dt} + k_a V_a}{A_s}
$$

In Eq. (1):

- $p$ - Oil cavity pressure near the oil outlet end;
- $P_o$ - Chamber pressure;
- $B_a$ - The damping coefficient of oil cavity;
- $k_a$ - The spring stiffness of the gas;
- $C_a$ - The damping coefficient of the gas;
- $V_a$ - Accumulator volume.
Assuming \( \left( P_1, V_1 \right) \) is pressure and volume state of a certain working point, \( \left( P_a, V_a \right) \) is the working state of the accumulator on any point in a chamber, by the gas formula one can get:

\[
PV_1^k = P_aV_a^k
\]  

(2)

By applying derivation to point \( \left( P_1, V_1 \right) \) in Eq. 2 and omitting the higher order terms one can get:

\[
\frac{dP}{dt} = \frac{dP_1}{dt} \frac{dV_a}{dt}
\]  

(3)

The hydraulic oil flow rate of accumulator inlet valve is:

\[
q_4 = -\frac{dV_a}{dt}
\]  

(4)

Combine Eq. 3 and 4 one obtain:

\[
P = kP \frac{dV}{dt} - q.
\]  

(5)

Replace \( q_4 \) in Eq. 5 using Eq. 4, then substitute into Eq. 1 to form the Laplace transform:

\[
G(s) = \frac{P(s)}{Q(s)} = \frac{1}{A_e^2} \left[ m_2s^2 + \left( B_e + C_e \right)s + \left( k_e + \frac{kP \cdot A_e}{V_0} \right) \right]
\]  

(6)

If the flow of accumulator inlet valve is \( q \), the oil cavity fluid changes and storage volume change can be calculated from Eq. (7):

\[
q = -\frac{\Delta V}{\Delta t}
\]  

(7)

Then:

\[
q = -\dot{V_a}
\]  

(8)

After making Eq. (8) Laplace transformation then brings into Eq. (6) one can obtain the matheamtical model of accumulator by gas chamber volume as the output:

\[
G(s) = \frac{V_a(s)}{P_a(s)} = \frac{A_e^2}{m_2s^2 + \left( B_e + C_e \right)s + \left( k_e + \frac{kP \cdot A_e}{V_0} \right)}
\]  

(9)

Then one can get:

\[
G(s) = \frac{A_e^2}{\left( k_e + \frac{kP \cdot A_e}{V_0} \right)} \frac{\omega_n^2}{s^2 + 2\zeta \omega_n s + \omega_n^2}
\]  

(10)

where \( \omega_n \) is the undamped natural frequency of accumulator, \( \zeta \) is the gas - oil equivalent damping ratio of accumulator. We can see that the performance of the hydraulic accumulator is associated with the cross section area of circle, air pressure, and air volume [19].

In order to further explore what role the accumulator plays in the wave power hydraulic transmission system and the influence of its main parameters on the stability and rapidity of the system, we carry out the concrete analysis through the AMESim simulation model [20].

When the gas pressure of the accumulator is 10 MPa, the gas volume is 40 L then run the simulation model. Fig. 6 is the motor outlet pressure simulation curves when the accumulator is open and closed.

Fig. 6 suggests when the accumulator is open, the establishing time of motor output port pressure is 1.8 s, the peak pressure is 13 MPa. After 5 s the output pressure stabilized at about 12 MPa, the output pressure of the motor is relatively stable, thus ensuring the stability of synchronous generator output power.

When the accumulator is turned off, the system pressure is created after 0.8 s and the peak pressure is 14 MPa, then the oil pressure continued to decrease rapidly and reduced to 3.9 MPa at 2 s, when the time is 3.5 s the pressure is raised to 14 MPa again, and at 4 s the pressure will be zero, this process cycle is a continuous operation. The maximum pressure fluctuation difference is 14 MPa, the output pressure is not stable, and it will cause difficulty to the subsequent power handling progress.

Through comparative analysis of simulation curves one can get the accumulator can absorb the impact pressure in the system effectively and keep the pressure of hydraulic system stable. It plays an obvious role in hydraulic transmission systems. But the existence of the accumulator also reduces the system response speed and decreases the peak. Therefore, the reasonable matching of accumulator parameters to improve the rapidity and stability of the system is the key of the study.

The air pressure and air volume is the main parameters of accumulator. Next, we will make the specific analysis about their influence on the work state of system.

**PRESSURE OF ACCUMULATOR INFLUENCE ON THE SYSTEM**

The volume of the accumulator is set at 40 L, and the working pressure is 8, 10, 11, 13 MPa, respectively. The output pressure curve of motor is shown in Fig. 7 after running the simulation model.
Fig. 7 suggests when the inflation pressure of accumulator is too small at 8 MPa, the peak pressure of motor is greater than 17 MPa, and the response time is 3 s. When the air pressure in accumulator is 13 MPa, the response time of system is 0.8 s, but the impact pressure is shocked evidently. When the pressure of accumulator is 10 MPa or 11 MPa, the response time is 1.8 s and the output pressure of system is relatively stable, remaining at about 12.4 MPa. The simulation results show that when the filling pressure is 80% or 90% of the actual working pressure, the rapidity and stability of system output is better.

VOLUME OF ACCUMULATOR INFLUENCE ON THE SYSTEM

The setting of the accumulator pressure is 10 MPa and the inflatable volume is 10, 20, and 40, 60 L, respectively. Fig. 8 shows the output pressure curve after running the simulation model.

Fig. 8 suggests when the charging volume of accumulator is 10 L or 20 L; the response time is 1.5 s and 1.7 s, respectively. The peak pressure of system is 7 MPa and 15 MPa. When the charging volume of accumulator is 60 L, the peak pressure is 12.7 MPa, then the buffer effect for system is obvious, but the response time becomes slower for 2 s. When the gas volume is 40 L, the buffer effect for system is medium and the maximum pressure is 13.2 MPa, the response time is 1.8 s. The simulation results show: when accumulator aeration volume is too small, the buffer effect is reduced, the pressure output volatility is obvious; if the inflatable volume is too large the system response will become slow, the cost will increase and the occupied space becomes large.

EXPERIMENTAL RESULT AND ANALYSIS

CONSTRUCTION OF EXPERIMENTAL APPARATUS

For experimental study of the performance of each component and how the settings parameter will bring the influence on the work state of system, setup of the land test bench is shown in Fig. 9: Using the hoist to pull the float along with the buoy moves up and down to simulate the movement of float which acted on by the vertical wave forces. Then the hydraulic energy can be transferred and the system could output power finally.

Fig. 10 is the main part of hydraulic transmission system and data acquisition system, the PC software can collect the real-time data of float displacement, the outlet pressure of motor, the output voltage, and the current parameters of generator. It provides the reference for further analysis and design optimization of the system.
EXPERIMENTAL DATE ANALYSIS

Fig. 11 is the motor pressure curve and output voltage curve of the generator when the accumulator volume is 40 L and the pressure is 10 MPa. After 5 s of float running, the motor output pressure gradually built up, and the maximum peak pressure reached 11.7 MPa, the pressure fluctuation ranges from 8.8 to 11.7 MPa. The generator output voltage peak value is 240 V, and the voltage fluctuation ranges from 180 to 240 V. The change trend of voltage output curve is consistent with the change trend of motor output pressure curve. Fig. 12 is the motor output pressure and generator output voltage curve when the accumulator is turned off. In this case, the fluctuations of output pressure and voltage curves are obvious and its value is discontinuous. The maximum pressure difference is 14 MPa, and the maximum voltage difference is 300 V. The output peak is greater than the condition of accumulator is open; it is identical with the simulation results. When the accumulator is closed, after the float stopping movement, the generator can continue to run 1 s under the force of inertia then stopped immediately. When the accumulator is open, after the float stopping movement, the generator will still continue to generate electricity for about 10 s, thus ensure the continuity of electricity output.

Fig. 13 is the pressure and voltage curves when the volume of accumulator is 40 L and the pressure is 5 MPa. When the float has been running for 6 s, the output pressure of motor is gradually built up, then the maximum pressure peak is 12.5 MPa and the pressure fluctuation ranges from 9.5 to 12.5 MPa. The generator output voltage peak value is 210 V, the voltage fluctuation ranges from 160 to 210 V. For comparison, the pressures and voltage output curves under the condition of same air volume and different air pressure is analyzed. It can be drawn: when the air pressure in the accumulator is smaller, the output peak of pressure is larger, the voltage fluctuation range becomes smaller, and when the float stops motion the generator can output power continuously for a longer time.

CONCLUSIONS

1. The changing trends of experimental data curves and simulation curves are consistent, verifying that the design principle of hydraulic drive system is feasible and the built of simulation model is correct. The study on the hydraulic transmission systems of wave energy converter clarifies the impact of accumulator on the power production. It provides a reliable reference for further optimization.

2. By analyzing the simulation curves and experimental data curves one can obtain that the accumulator plays an enormous role in relieving the pressure impact of system and the correct configuration of accumulator parameters can improve the rapidity and stability of the system work.

ACKNOWLEDGEMENTS

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CHINA
INFLUENCE OF FRICTION STIR WELDING (FSW) ON MECHANICAL AND CORROSION PROPERTIES OF AW-7020M AND AW-7020 ALLOYS

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Gdynia Maritime University, Poland
Wojciech Jurczak
Polish Naval Academy, Poland

ABSTRACT

Friction welding associated with mixing the weld material (FSW - Friction Stir Welding) is an alternative to MIG and TIG welding techniques for Al-alloys. This paper presents experimental results obtained from static tension tests on specimens made of AW-7020M and AW-7020 alloys and their joints welded by using FSW method carried out on flat specimens, according to Polish standards: PN-EN ISO 4136:2011 and PN-EN ISO 6892-1:2010. Results of corrosion resistance tests are also presented. The tests were performed by using the electrochemical impedance spectroscopy (EIS). EIS measurement was conducted with the use of three-electrode system in a substitute sea water environment (3.5% NaCl - water solution). The impedance tests were carried out under corrosion potential. Voltage signal amplitude was equal to 10mV, and its frequency range - 100 kHz ÷ 0.1 Hz. Atlas 0531 EU&IA potentiostat was used for the tests. For the tested object an equivalent model was selected in the form of a substitute electric circuit. Results of the impedance spectroscopy tests are presented in the form of parameters which characterize corrosion process, as well as on Nyquist’s graphs together with the best-fit theoretical curve.

Analysis of the test results showed that the value of charge transfer resistance through double layer, \( R_{ct} \), for the FSW-welded specimen, was lower than that for the basic material, and that much greater difference was found in the case of AW-7020M alloy.

The impedance spectroscopy tests showed that both the FSW-welded joints and basic material of AW-7020M and AW-7020 alloys were characterized by a good resistance against electrochemical corrosion in sea water environment, and that FSW -welded joints revealed a greater corrosion rate.

The performed tests and subject-matter literature research indicate that application of FSW method to joining Al-alloys in shipbuilding is rational.

Keywords: Al-alloys, mechanical properties, electrochemical impedance spectroscopy (EIS), corrosion, FSW, friction welding

INTRODUCTION

Such properties of Al-alloys as: non-magnetism, high value of the relative strength factor \( R_{p0.2}/\rho \), relatively good corrosion resistance in sea water and atmosphere, weldability as well as good fracture toughness (also in low temperatures) have decided on wide range of their application in world shipbuilding [2].

In shipbuilding industry are widely used Al-alloys of 5xxx series whose higher strength properties of parent material are reached as a result of cold working. However influence of temperature during welding such alloys results in strength degradation in heat affected zone (HAZ) down to their properties in mild state, i.e. by abt. 20 ÷ 30 %, which is hard to be estimated precisely. Qualitative changes in load-carrying ability analysis are especially visible for welded aluminium structures, in HAZ [11]. The phenomena are specific for majority of 5xxx series Al-alloys. Al-alloys in the annealing state „O”, raw state „F” as well as the annealing and light-hardening state „H111” do not suffer the above mentioned destruction [8].

In the 1990s high interest started to be paid to the weldable Al-alloys of 7xxx series (Al-Zn-Mg), characteristic of much higher strength properties compared to those of 5xxx series Al-alloys. A disadvantage of Al-Zn-Mg alloys is their dependence to stress corrosion fractures in sea-water environment, especially when total (Zn+Mg) amount exceeds 6% [2]. Due to application of an appropriate heat treatment consisting in solutioning and artificial ageing, resistance of the alloys to environmental destruction reached a satisfactory level. In contrast to Al-Mg alloys, strength properties of welded joints of Al-Zn-Mg alloys reach similar values as those of parent material [1]. Unfortunately, their resistance to corrosion, especially stress corrosion in sea water, drastically drops. One of the most common, high-strength Al-alloys of 7xxx series applied in shipbuilding is 7020 Al-alloy. The above
mentioned dependence to stress corrosion fracture in sea-water environment which occurs first of all in welded joints results in undertaking research towards modification of the alloy. As a result, was developed the alloy marked 7020M whose chemical composition was changed (amount of Zn+Mg was increased to >6.55% at increasing the amount of Cr and Zr) [8].

The joining of aluminium and their alloys by welding is rather difficult due to its specific properties. Principal difficulties which occur during welding Al-alloys result from: a high affinity between aluminium and oxygen and forming a hard-melting oxide, Al₂O₃ (in 2060 °C), high heat conductivity, high thermal expansion of Al-alloys, high casting shrinkage (resulting in post-welding stresses and deformations), rather large strength drops in welding temperatures, losing some alloying components such as Mn, Zn or Li in welding temperatures [8]. The shortly described principal difficulties associated with welding Al-alloys encourage to search for other methods of joining such materials. The friction stir welding (FSW) can serve as an alternative in case of butt joining the plates [7, 10, 12].

The technology of friction welding associated with mixing the weld material (Friction Stir Welding) was developed and patented in 1991 by Welding Institute (TWI), Cambridge, UK. In the method for heating and plasticizing the material, was used a tool with rotary arbour, located in the place of joining the plates pressed tight together. After making the tool arbour rotating, friction heating and plasticizing the material in its direct neighbourhood, slow move of the whole system along contact line, is initiated. FSW is a method of friction welding of such materials in solid state so far as Al-alloys, copper and stainless steels. The main advantage of the method is easiness of producing joints of high, repeatable properties [3,4,5]. Because the friction welding technology is applicable to materials in solid state (below their melting temperatures), strength properties of joints produced this way may be higher than those reached by using the arc welding techniques (MIG, TIG) [13]. This is especially important when to take into account that plates made of Al-alloys are often subjected to heat treatment. Therefore the traditional welding methods, by charging large heat amount, introduce important changes in structure of joined material (especially in its heat affected zone). It causes substantial decrease of strength properties of welded joints, but first of all, a drastic drop in their corrosion resistance (stress and fatigue corrosion), especially in such aggressive environment as sea water.

This work is aimed at determination of influence of friction welding with the use of Friction Stir Welding (FSW) method, on corrosion resistance of AW-7020 and AW-7020M alloys in sea water. Static tension tests were also conducted to determine mechanical properties of joints of the selected alloys welded by using FSW method.

**TEST METHOD**

For the tests EN AW-7020M and EN AW-7020 alloys were selected. The alloys were subjected to T651- heat treatment. Their chemical composition is shown in Tab. 1.

**Table 1. Chemical composition of 7020M and 7020 alloys [9]**

<table>
<thead>
<tr>
<th>No. of melt</th>
<th>Chemical composition [%] of 7020M alloy</th>
</tr>
</thead>
<tbody>
<tr>
<td>507</td>
<td>Zn  5.15, Mg  1.9, Cr  0.16, Zr  0.15, Ti  0.071, Fe  0.27, Si  0.15, Cu  0.08, Mn  0.057, Ni  0.006, rest</td>
</tr>
<tr>
<td>515</td>
<td>Mg  4.81, Zn  1.9, Cr  0.17, Zr  0.12, Ti  0.016, Fe  0.31, Si  0.21, Cu  0.09, Mn  0.06, Ni  0.006, rest</td>
</tr>
</tbody>
</table>

**Type of heat treatment:**

T651: super saturation - heating up to 480°C for 50 min, cooling with hot water of min. 70°C, natural ageing for 6-4 days at 20°C, two-stage artificial ageing - 95°C/3h + 150°C/3h.

**Chemical composition [%] of 7020M alloy**

<table>
<thead>
<tr>
<th>Mg</th>
<th>Mn</th>
<th>Ti</th>
<th>Zn</th>
<th>Cr</th>
<th>Si</th>
<th>Fe</th>
<th>Cu</th>
<th>Zr</th>
<th>Al</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.25</td>
<td>0.18</td>
<td>0.034</td>
<td>5.3</td>
<td>0.14</td>
<td>0.16</td>
<td>0.32</td>
<td>0.05</td>
<td>0.04</td>
<td>rest</td>
</tr>
</tbody>
</table>

**Type of heat treatment:**

T651: super saturation - heating up to 430°C for 45 min, cooling with water of min. 15°C, natural ageing for 6-6 days at 20°C, artificial ageing - 120°C/96h.

Butt joints of plates of the thickness g = 10 mm were made by using FSW method. The plates of both the tested alloys were welded on both sides at the same welding parameters. Schematic diagram of the friction stir welding connected with mixing the weld material (FSW) is presented in Fig. 1, and its parameters – in Tab. 2.

**Table 2. Parameters of FSW welding procedure for 7020M and 7020 alloys**

<table>
<thead>
<tr>
<th>Dimensions of arbour</th>
<th>Tool slope angle Tool rotational speed Vₐ [rpm] Tool linear speed Vₜ [mm/min]</th>
<th>Welding linear speed Vₙ [mm/min]</th>
</tr>
</thead>
<tbody>
<tr>
<td>D [mm] d [mm] h [mm]</td>
<td>αₙ</td>
<td>88,5</td>
</tr>
</tbody>
</table>

**Fig. 1. Schematic diagram of FSW welding procedure [5]**
STATIC TENSION TEST

Static tension tests were performed to determine mechanical properties of plates made of 7020 and 7020M alloys and their joints produced by using FSW method. Flat specimens cut perpendicularly to milling direction were used. The tests were conducted in ambient temperature of +20 °C ± 2. The tension tests were carried out in accordance with Polish standards: PN-EN ISO 4136:2011 and PN-EN ISO 6892-1:2010 with the use of EU-40 testing machine of force range up to 200 kN.

TESTS OF ELECTROCHEMICAL CORROSION RESISTANCE

Measurement of electrochemical corrosion resistance was performed by using the electrochemical impedance spectroscopy method (EIS) in accordance with ASTM G 3 and ASTM G 106 standards [15, 16].

The applied three-electrode system consisted of the following elements: specimen, an auxiliary (polarizing) electrode of platinized titanium and a reference electrode (saturated calomel electrode). The electrodes were placed into a vessel filled with 3,5 % NaCl –water solution. The active surface of the specimens was equal to 1 cm².

During measurement the electrolyte was all the time stirred by means of a magnetic mixer. The specimens were degreased in advance.

Atlas 0531 EU&IA potentiostat was used for performing the tests.

The impedance tests were carried out under corrosion potential. The voltage signal amplitude was equal to 10 mV, and the range of signal frequency changes amounted to 100 kHz ÷ 0,1 Hz.

Corrosion process parameters were determined by using AtlasLab 2.0 and EIS Spectrum Analyser software.

For the tested object an equivalent model in the form of a substitute electric circuit was selected. Its schematic diagram is presented in Fig. 2.

\[
Z = R_s + \frac{1}{R_{ct} + CPE_{dl}(j\omega)^n}
\]  

RESULTS OF THE TESTS

STATIC TENSION TESTS

Tab. 3 shows results of the static tension tests conducted on the investigated alloys and their joints welded by using FSW method.

<table>
<thead>
<tr>
<th>Specimen</th>
<th>$R_m$ [MPa]</th>
<th>$R_{p0,2}$ [MPa]</th>
<th>A [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>7020M</td>
<td>443</td>
<td>397</td>
<td>9,8</td>
</tr>
<tr>
<td>7020M - FSW</td>
<td>437</td>
<td>393</td>
<td>10,2</td>
</tr>
<tr>
<td>7020</td>
<td>372</td>
<td>317</td>
<td>16,0</td>
</tr>
<tr>
<td>7020 - FSW</td>
<td>370</td>
<td>314</td>
<td>15,4</td>
</tr>
</tbody>
</table>

The mechanical properties of FSW-welded joints are almost the same as those of respective parent materials. This is proved by location of fracture which, in all the tested specimens, occurred in parent materials in the distance of about 10 mm apart from weld junction. In the case of friction welding, heat affected zone (HAZ) is very narrow, which may be concluded that the material in the place of fracture was unchanged, i.e. it remained parent. Fig. 3 shows image of an example FSW-welded specimen after static tension test.

TESTS OF RESISTANCE TO ELECTROCHEMICAL CORROSION

The test results of electrochemical impedance spectroscopy (EIS) for the specimens made of parent material of 7020M and 7020 alloys and their FSW-welded joints, recorded during the tests, were analyzed by using EIS Spectrum Analyser software. As a result, were determined corrosion process parameters whose mean values based on five specimens are presented in...
Tab. 4. The parameters determine particular components of the model, i.e. the substitute electric circuit selected in compliance with the applied testing method.

Tab. 4. Parameters of the substitute electric circuit for the corrosion system of 7020M and 7020 alloys and their joints welded by using FSW method

<table>
<thead>
<tr>
<th>Specimen</th>
<th>$R_c$ [Ω cm²]</th>
<th>$R_{ct}$ [Ω cm²]</th>
<th>$CPE_{dl}$ [$\mu$F cm²]</th>
<th>$\beta$ [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>7020M</td>
<td>0.28</td>
<td>1241.4</td>
<td>8.8</td>
<td>0.984</td>
</tr>
<tr>
<td>7020M - FSW</td>
<td>0.45</td>
<td>614.7</td>
<td>10.8</td>
<td>0.996</td>
</tr>
<tr>
<td>7020</td>
<td>0.42</td>
<td>773.01</td>
<td>29.13</td>
<td>0.941</td>
</tr>
<tr>
<td>7020 - FSW</td>
<td>0.66</td>
<td>534.55</td>
<td>21.82</td>
<td>0.985</td>
</tr>
</tbody>
</table>

In the case of AW-7020M alloy value of the resistance of charge transfer through double layer, $R_{ct}$, is twice smaller for the welded specimen than for the parent material, which indicates that charge transfer resistance between material and electrolyte is lower. Differences in $R_{ct}$ values recorded during testing the AW-7020 alloy and its FSW –welded joints were significantly lower (by about 30%), and also the welded specimens revealed a lower value of the parameter in question.

The component of the capacity impedance power exponent $\beta$ for the substitute system, which determines homogeneity of corrosion process occurring on surfaces of the tested specimens, took similar high values (exceeding 0.9) for all the specimens. It shows an activating character of the constant-phase element CPE$_{dl}$ in the conditions of the performed tests. The value of the component of capacity impedance power exponent close to or equal to 1 may show that during the tests uniform corrosion takes place or no diffusion limitations are present during the corrosion process.

The results of the impedance spectroscopy tests are also presented graphically on Nyquist’s diagrams (Fig. 4 and 5), together with the relevant best-fit theoretical curves.

**SUMMARY AND CONCLUSIONS**

The tests of mechanical properties of AW-7020 and AW-7020M alloys have shown that the alloy with the changed chemical composition reveals a higher strength and lower plasticity compared to the basic alloy (7020). For both the tested alloys the joints welded by using the friction stir welding method (FSW) probably show higher strength properties compared to their parent materials. This is proved by the fact that fracture occurred in the place 10 mm distant from the weld. Results of the strength properties recorded during static tension tests are on the same level.

An analysis of the data obtained from the impedance spectroscopy tests revealed that the FSW-welded specimens showed a lower resistance to electrochemical corrosion compared to those made of parent material, and a greater difference was recorded in the case of 7020M alloy.

For all the specimens it was stated that corrosion process is based on activating control.

In the case of AW-7020M alloy, the parameter $R_{ct}$ (charge transfer resistance), the most important from the point of view of electrochemical corrosion resistance, obtains twice greater values for parent material specimen compared to that welded by using FSW method. For AW-7020 alloy the difference between the recorded $R_{ct}$ values for parent material and FSW-welded joint reached value of about 30% in favour of parent material.

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ASSESSMENT OF A BOAT FRACTURED STEERING WHEEL

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ABSTRACT

During regular use of the steering wheel mounted on a boat, two cracks emanating from a fastener hole were noticed which, consequently, caused final fracture of the wheel. To determine the behavior of a boat steering wheel with cracks present, assessment of a fractured wheel was performed. Torque moments of the fasteners were measured prior to removing the steering wheel from the boat. Visual and dye penetrant inspection followed along with the material detection. Besides using experimental procedures, assessment of the fractured wheel was performed using finite element analysis, i.e. stress intensity factor values were numerically determined. Variation of stress intensity factor with crack length is presented. Possible causes of crack occurrence are given and they include excessive values of fastener torque moments coupled with fretting between fastener and fastener hole that was poorly machined. Results obtained by this assessment can be taken for predicting fracture behavior of a cracked steering wheel and as a reference in the design, mounting and exploitation process of steering wheels improving that way their safety in transportation environment.

Keywords: crack, fracture, stress intensity factor, fastener hole, steering wheel

INTRODUCTION

Imperfections in material, manufacturing faults and/or unfavorable service conditions can easily cause flaw appearance in engineering structures. The consequential crack growth can seriously affect integrity of such structures threatening the structures themselves, their users and leading to substantial financial losses. This is particularly important when dealing with possible fracture of components that are used in transportation vehicles.

Catastrophic consequence can occur if components susceptible to crack growth are used in vehicle assembly. Therefore, it is crucial to use fracture mechanics principles to assess design of such components and perform adequate material selection [1, 2, 3].

One of such problems is discussed in this paper. Performance of a steering wheel mounted on a boat was observed. The steering wheel was of an unknown make and mounted on the boat as a replacement for the original equipment manufacturer (OEM) steering wheel. After nine years of regular marine service conditions, multiple cracks emanating from one of the fastener holes were noted. Crack propagation was monitored (only visually, without taking any relevant measurements) in the following period until the final fracture occurred. In order to numerically simulate service conditions, fasteners torque moments were measured prior to removing steering wheel from the boat. Visual inspection of cracked planes followed. Furthermore, chemical material composition and mechanical properties were determined. After that, finite element (FE) analysis was used to describe fracture behavior of the steering wheel. Obtained values of stress intensity factors gave insight into the fracture behavior and possible causes of crack occurrence and growth. Results are valuable as a reference data in the design, mounting, exploitation and for predicting fracture propagation of a steering wheel.
CRACKED STEERING WHEEL

Multiple crack points were noticed on the in-use powerboat steering wheel after about nine years of regular maritime service conditions use. Since the steering wheel is not an OEM part, mounting was different when comparing to factory defaults. Instead of being attached by one fastener, aftermarket wheel uses an adapter to connect with steering column and it is fastened to the adapter by six fasteners. Two cracks started to emanate from one of the fastener holes causing the final fracture of the wheel, Fig. 1. Cracks started as semielliptical cracks tending to straighten their fronts as the crack propagated.

Fig. 1. Cracked steering wheel with a detail of a crack.

Geometry and dimensions of wheel skeleton (plastic lining removed) are given in Fig. 2.

Fig. 2. Geometry and dimensions (in mm) of the steering wheel.

Torque values used to tighten the fasteners were measured prior to detaching steering wheel from the column. Common marking test was performed using dial torque wrench; position of fasteners on wheel surface was marked and fasteners were then loosened and retightened again until the marks were aligned. Torques required to return fasteners to their original positions are taken as the original torques applied to fasteners (Tab. 1).

<table>
<thead>
<tr>
<th>Fastener no.</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M_t$ (Nm)</td>
<td>4.78</td>
<td>5.11</td>
<td>5.20</td>
<td>5.05</td>
<td>4.95</td>
<td>8.65</td>
</tr>
</tbody>
</table>

Measured values show that the fastener mounted at the position no. 6 had relatively high torque value comparing to the other five. This excessive torque value could be due to initial mounting of the steering wheel, but it should not be neglected that it could also be a result of retightening the fastener after it loosened due to crack occurrence. However, regardless of way that is was introduced, excessive torque could initiate fatigue crack growth and final fracture.

A piece of the steering wheel was given for chemical analysis to determine the type of material. Composition of material given in Tab. 2 suggests that the steering wheel was made of aluminum alloy AA 6061.

<table>
<thead>
<tr>
<th>Mg</th>
<th>Si</th>
<th>Fe</th>
<th>Cu</th>
<th>Cr</th>
<th>Mn</th>
<th>Ti</th>
<th>Zn</th>
<th>Al</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.98</td>
<td>0.551</td>
<td>0.498</td>
<td>0.251</td>
<td>0.185</td>
<td>0.055</td>
<td>0.0497</td>
<td>0.0256</td>
<td>97.937</td>
</tr>
</tbody>
</table>

When comparing obtained values with the reference [4], it can be noted that the iron percentage is below the standard value (0.7%). The same goes for manganese percentage that is below standard value (0.15%) and titanium which is below 0.15%.

AA 6061 is one of the most common aluminum alloys for general purpose use. It is a precipitation hardening aluminum alloy with good mechanical properties, widely used in automotive, aero and marine industry for various applications. Previous research [5] reveals that aluminum AA 6061 is not susceptible to stress corrosion cracking, so this effect was neglected in further research.

Mechanical properties of the mentioned material are given in Tab. 3 ($\sigma_y$ - yield strength, $\sigma_u$ - ultimate tensile strength, $E$ - elastic modulus).

<table>
<thead>
<tr>
<th>Material</th>
<th>$\sigma_y$ (MPa)</th>
<th>$\sigma_u$ (MPa)</th>
<th>$E$ (GPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AA 6061</td>
<td>55.1</td>
<td>124</td>
<td>68.9</td>
</tr>
</tbody>
</table>

In order to reveal surface defects that could serve as crack initiation points on other fastener holes, dye penetrant test was performed. Results of the test can be seen in Fig. 3.
Results of the performed dye penetrant test reveal defects on the back of the steering wheel surface on the edge of the fastener holes. Defects originate from poor machining of the holes and can act as crack initiation points.

**FRACTURE THEORY**

For materials that exhibit small-scale yielding at the crack tip, linear elastic fracture mechanics parameters can be used to assess the mentioned fracture behavior. Stress intensity factor (SIF) is one of such parameters that completely defines crack tip conditions, i.e. stress, strain and displacement values and can be connected with energy release rate \( G \) [6]. Three types of SIFs – Mode I (opening), II (in-plane shear) and III (out-of-plane shear), can be recognized depending on the crack opening mode. In order to determine SIF, several methods were developed, including singular integral equations [7], weight functions [8], finite element [9] or extended finite element methods [10]. Closed-form SIF solutions were developed for a number of different geometries and loads [11] and of particular interest in this work is the example of two cracks of length \( a \) emanating from a round hole of radius \( r \) subjected to internal pressure \( p \), Fig. 4.

Mode I SIF for example in Fig. 5. according to [11] can be determined using equation:

\[
K_i = p \sqrt{\pi a} \cdot F_i(s),
\]

where:

\[
s = \frac{a}{R+a},
\]

and:

\[
F_i(s) = (1 - \lambda) F_0(s) + \lambda F_1(s),
\]

with \( F_0 \) and \( F_1 \) being:

\[
F_0(s) = (1-s) \left[ 0.637 + 0.485(1-s)^2 + 0.4s^2(1-s) \right],
\]

\[
F_1(s) = 1 + (1-s) \left[ 0.5 + 0.743(1-s)^2 \right].
\]

Research of deriving solutions for SIFs is ongoing, especially for the applied solutions. Some earlier work includes FE alternating method for calculating SIF of cracks emanating from fastener holes [12]. Displacement extrapolation method was used in order to determine SIF for a crack emanating from a rivet hole in a plate [13]. An effort to model multiple curved cracks emanating from a circular hole using singular integral equation was made [14]. Hybrid displacement discontinuity method was used for describing interaction of two collinear circular hole cracks in infinite plate subjected to internal pressure [15]. Analytical conforming mapping method can be engaged to resolve the problem of cracks emanating from a hole in a linear elastic plate subject to in-plane tension [16]. Conforming mapping was also used for anti-plane problem of two asymmetrical edge cracks emanating from an elliptical hole [17]. Combining fracture and fatigue criterions, a mathematical model was developed to investigate the mixed mode crack growth process of a plate with two cracks emanating from drilled holes [18].

Previous research on steering wheels mainly concentrate on mechanisms of force transmission [19], impact in case of collision [20] and ergonomics [21]. Nevertheless, no previous work had been found that deals with crack occurrence or fracture of steering wheels of any kind. This paper represents a step in that direction using numerical analysis to investigate fracture behavior of cracked steering wheel and helping to design against failure in the presence of flaws.

**FINITE ELEMENT ANALYSIS**

In order to build a numerical model of cracked steering wheel, finite element (FE) method was used. Material behavior was considered linear elastic isotropic with properties set according to Tab. 3. Steering wheel FE model was defined according to geometry in Fig. 2, but due to symmetry only half of the wheel is modelled, Fig. 5. Symmetric boundary
conditions are applied on the cross-section surface and only rotation round z-axis is allowed.

As for load, fastener holes are loaded according to torque loads measured in Tab. 1. The fastener surface is assumed to be smooth. Thus, it transmits only radial loads to the hole. Additionally, pressure of drivers arm resting on steering wheel is added to juncture of rim and horizontal spoke.

On fastener hole no. 6 two full-through cracks are modelled according to Fig. 1. Crack walls opening and tip deformation are, in the real case, restricted by the friction with the steering column surface, but in FE analysis this was neglected. To accommodate the singular stress field in the vicinity of the crack tips, these areas were meshed with 20-node isoparametric brick elements collapsed to wedges. The rest of the model consists of 20-node isoparametric brick elements. Particular care was taken when meshing the area around crack tips ensuring that the mesh is fine enough to properly capture stresses and deformations. The model was meshed with about 66,000 elements with this number showing balance between accuracy of results and computer process time economy. Models with about 40,000 and 80,000 elements were tested, but failed in one of the mentioned categories.

For cracks nearly perpendicular to the loading axis, the mode II influence on the SIF is at the most 2% for the ratio suggested in [22]. Although Mode I SIFs are assumed to dominate over mode II effects, both are calculated in this work.

The FE method is employed in order to obtain SIFs because of its versatility when dealing with a vast array of different cracked structures, being standardized specimens [23] or real structures [24]. It was proved [25] that the theoretical strain/stress singularity at the corner of a 20-node isoparametric element can be achieved by moving the mid-side nodes of the elements to 1/4-point positions toward the crack tip. SIF can then be estimated for plane strain conditions from the crack-surface displacements at these 1/4-points which is what was done here.

RESULTS AND DISCUSSION

Using proposed FE model and procedure, SIFs are calculated for an array of different crack lengths \( a \) relative to hole radius \( R \) (\( a/R = 0.25, 0.5, 0.75, 1, \ldots 3 \)). Values of SIFs are normalized according to relation:

\[
K_{f} = p \sqrt{\pi a}. \quad (6)
\]

Results are given in Fig. 6. In order to validate the FE model, results for mode I SIF were calculated according to Eq. (1). Although they cannot be quite compared because of notable differences in geometry and load, they can give some sort of confidence in using FE model. Reference SIFs obtained by eq. (1)-(5) do not take into account finite width of the plate, while that obtained by FEA do. It can be seen that the SIF values obtained by equation and FE model correspond in some way, and the difference can probably be attributed to loading conditions. Main difference is that in the cracked steering wheel there is also a mode II SIF. It can be noted that mode I SIFs dominate over mode II values. Also, there is a considerable growth of SIF with the crack changing length between \( a/R = 0.25 \) and 0.5; after that SIF values grow steadily.

\( a) \)
CONCLUSION

Assessment of a fractured steering wheel has been performed in this paper. During regular use of steering wheel mounted on the boat, two cracks started emanating from one of the six fracture holes thru which the wheel was attached to column. After the final fracture, the wheel was examined experimentally and numerically. Analysis of chemical composition revealed that the wheel was made of AA 6061 aluminum alloy. Furthermore, the surface of wheel around holes was examined using dye penetrant test. Results showed surface defects caused by poor machining that could serve as crack initiation points. Visual inspection of the wheel revealed that two cracks emanated from a hole in which fastener was exposed to excessive torque. Using FE analysis mode I and II stress intensity factors were calculated according to different crack lengths. Obtained results give insight into the fracture behavior of a cracked steering wheel and can be found useful used in the design process, mounting or exploitation of such component improving their safety in transportation environment this way. Further study should include verification of numerically predicted crack growth with measured values obtained by monitoring the process.

ACKNOWLEDGMENT

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FAULT RISK ASSESSMENT OF UNDERWATER VEHICLE STEERING SYSTEM BASED ON VIRTUAL PROTOTYPING AND MONTE CARLO SIMULATION

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ABSTRACT

Assessing the risks of steering system faults in underwater vehicles is a human-machine-environment (HME) systematic safety field that studies faults in the steering system itself, the driver’s human reliability (HR) and various environmental conditions. This paper proposed a fault risk assessment method for an underwater vehicle steering system based on virtual prototyping and Monte Carlo simulation. A virtual steering system prototype was established and validated to rectify a lack of historic fault data. Fault injection and simulation were conducted to acquire fault simulation data. A Monte Carlo simulation was adopted that integrated randomness due to the human operator and environment. Randomness and uncertainty of the human, machine and environment were integrated in the method to obtain a probabilistic risk indicator. To verify the proposed method, a case of stuck rudder fault (SRF) risk assessment was studied. This method may provide a novel solution for fault risk assessment of a vehicle or other general HME system.

Keywords: fault risk assessment; underwater vehicle; virtual prototyping; Monte Carlo simulation; steering system; fault simulation

INTRODUCTION

The steering system is a vital subsystem that controls the track and attitude of space motion of underwater vehicles, which have a large scale and complicated constitution. Faults in the steering system may be located anywhere and are usually transient and sudden. Current industrial production has addressed condition monitoring and health management for steering systems; however, fault detection at an early stage remains difficult. Thus, a fault in the steering system is very likely to lead to a fatal accident. Collision is the most common form, which often leads to an inability to obtain fault data. The above items result in a lack of comprehensive and available historic data, which presents a substantial challenge to qualitative risk assessment [1].

However, a serious fault in the steering system may not always lead to a fatal accident. This is because the adverse effect caused by the fault can be recovered via effective emergency operations. For example, if a stuck rudder fault (SRF) of the stern rudder occurs in an underwater vehicle, collision with the seabed may be avoided if recovery operations are followed to maintain an acceptable depth or to emerge from the water safely. This involves a “manoeuvring limitation”: under the circumstances that are beyond the manoeuvring limitation, a fatal accident is inevitable; however, under the circumstances that are within the manoeuvring limitation, a fatal accident can be avoided via effective recovery operations. Manoeuvring limitations are generally dictated according to the current motion and environmental parameters when the fault occurs, such as speed, fault parameter and depth.

However, one possibility cannot be overlooked: under circumstances of potential recovery, a fatal accident finally occurs due to a failure to follow correct operation procedures (human error). Although the human error probability of a well-trained operator is very small, it should not be overlooked. Notably, many traffic accidents have been proven to be caused by human error. According to Ref [2-3], the human factor is the most important factor leading to highway transportation accidents, in which human error contributes to a large proportion. Additionally, human error probability could be considerably increased under emergency circumstances, which warrants detailed study to obtain a comparison for this change.
According to the above analysis, risk assessments of an underwater vehicle related to a steering system fault address a typical HME systematic safety problem. To obtain a solution to this problem, the interrelationships between the three elements (human, machine and environment) should be comprehensively considered in one framework instead of investigating them individually. In fault risk assessment, though a fault in the machine is the essential risk, environment factors as well as human error can substantially influence the safety of the HME system. However, the randomness of the three elements aggravates the complexity of the assessment process.

Above all, to assess the risk posed to the vehicle caused by a steering system fault, two difficulties are confronted: the scarcity of fault data and the puzzle of randomness.

**LITERATURE REVIEW**

Simulation technology has been rapidly developing in recent decades following advances computer science; simulations are now being applied to nearly all domains of modern society. High efficiency, low cost and infinite flexibility account for the huge popularity of simulation technology [4]. Simulation-based safety/risk assessment is an important research trend, and researchers have been developing it for application in many issues such as those related to fires, energy, structures, environment, health, foods, and the economy [5-11]; this diversity also demonstrates the strong adaptability of simulations. Monte Carlo simulation is the most widely used method [12-15], primarily due to its capability to address problems of randomness and uncertainty.

Many trials and positive simulation-based results in assessing risks have been used to vehicles in recent years. Ref [16] proposed an accident risk assessment in marine transportation based on a Markov Chain Monte Carlo simulation that can consider any accident or marine system and that does not need large-scale data collection. Ref [17] proposed a risk assessment method of aero engine failure based on Monte Carlo simulation that has been claimed to be very flexible. Ref [18] proposed an assessment approach for safety performance of road locations based on a combination of statistical analysis, numerical modelling, and micro simulation models, which can be used to estimate the number of conflicts and injury severity of a particular road location. Ref [19] proposed a probabilistic assessment method for vehicle safety under various driving conditions utilizing dynamic simulation models and the response surface method.

Most methodologies proposed in the above studies can be classified into two categories. One category is to utilize a simulation method (usually Monte Carlo) to estimate the frequency of accidents in a particular region and period. This methodology can only provide a global estimation of accident occurrence but is unable to consider fault details (or other reasons) that lead to accidents. Therefore, this approach cannot provide guidance for the design or use of the vehicle to reduce risk. The other category uses a simulation method to estimate the probability of a particular fault. The primary disadvantage of these methodologies is that they disregard environmental and human factors that may affect system safety. Thus, they need to be improved for application in an HME systemic safety problem.

In this paper, to address the first and second difficulties stated in section 1, respectively, virtual prototyping and Monte Carlo simulation are adopted, both of which are combined to form a novel simulation-based method for fault risk assessment in an underwater vehicle steering system. To verify the proposed method, a case study of SRF risk assessment of the underwater vehicle is presented afterwards in section 3. A conclusion is made in the last section.

**CASE STUDY PROCESS**

The steering system is an important subsystem that controls the track and attitude of space motion of underwater vehicles, which is a typical double closed-loop mechanic-electronic-hydraulic hybrid control system. It is a pump-cylinder servo system consisting of an operating device, pump control unit, cylinder and feedback unit, etc. Steering system has a large scale and complicated constitution. The actual rudder angle is controlled to track the order angle given by the operator via two-stage feedback control. The detailed principle of the steering system is illustrated in Fig. 1 [20], in which measurement points 1#-7# are going to be used in the validation process of virtual prototyping.

**VIRTUAL PROTOTYPING AND ITS VALIDATION**

Virtual prototyping is a digital model of the actual system, which is essentially a modelling and simulation (M&S) technique. Common platforms of virtual prototyping are AMESim, ADAMS [21], Pro/E, etc. Amesim is chosen as the modelling platform for virtual prototyping, and the established model is shown in Fig. 2, where measurement points 1#-7# correspond to Fig. 1, respectively. In the modelling process, a good balance should be achieved between the accuracy of the components and the complexity of the system [22].

For use in fault risk assessment, the established virtual prototyping needs to be verified and validated to ensure credibility and acceptability. A series of validation experiments was designed and implemented. Parts of the actual system and both the experiment plant and sensors are illustrated in Fig. 3.

![Fig. 1. Principle of steering system of underwater vehicle](image-url)
A detailed introduction and further discussion on verification, validation and accreditation (VV&A) can be found in Ref [23][24][25]. This paper, however, only focuses on a comparison between the actual signal and a simulation signal of the measurement points. Suppose that $x_t$ and $y_t$ are the observation sequences of the actual system and virtual prototyping, respectively; the error sequence between them is

$$e_t = x_t - y_t, \quad t = 1, 2, ..., N$$

where $N$ is the length of the sequence.

The Root-Mean-Square (RMS) and Correlation Coefficient (CC) of the error sequence are chosen as the similarity indicators in V&V, which can reflect the similarity of the amplitude and the trend between the two sequences, respectively. Their calculation formulas are shown in (2) and (3).

$$\text{RMS}_e = \frac{1}{\sqrt{N}} \sqrt{\sum_{t=1}^{N} e_t^2}$$

$$\rho_{X,Y} = \frac{\sum_{t=1}^{N} (x_t - \bar{x})(y_t - \bar{y})}{\sqrt{\sum_{t=1}^{N} (x_t - \bar{x})^2 \sum_{t=1}^{N} (y_t - \bar{y})^2}}$$

The calculation results are shown in Tab. 1, and the comparison between the signals of the actual system and virtual prototyping of 5# and 7# are illustrated in Fig. 4. From Tab.1 and Fig. 5, it can be inferred that the actual signal and simulation signal have a high similarity, which indicates that the virtual prototyping has a high credibility and acceptability for the use of fault risk assessment.
FAULT INJECTION AND FAULT SIMULATION

The steering system is a complicated mechanic-electronic-hydraulic hybrid control system. A fault in any component therein may lead to a failure in the entire system. There are nearly one hundred fault modes in the steering system according to its FMEA information, 25 of which are more typical and common. In this paper, two typical mechanical and electronic part failures, namely fault A and fault B, respectively, indicate (A) the displacement sensor of the flushing plate failing to obtain a constant gain of 1.5 and (B) the proportional valve amplifier with a constant output signal of zero. These two faults are included in the virtual prototyping. According to the characteristics of the faults, appropriate library components are selected to imitate the faults, as illustrated in Fig. 2.

Fault simulation is then carried out on the basis of fault injection. The step signal is chosen as the order signal, the initial value of which is 30°. The value stepped to 0° at 0 s. The simulation period is 10 s, and the sample interval is 0.1 s. The simulation results of these two faults and their comparisons with normal virtual prototyping are illustrated in Fig. 5.

It can be inferred from the figure that fault B could lead to a stuck rudder and fault A could not, though it also appears to be a very serious fault. This is the reason and purpose for modelling virtual prototyping and conducting fault simulation: to determine which of the faults can lead to a stuck rudder. In other words, the effects of each fault mode are assessed regarding their effect on the rudder angle output in an attempt to determine whether a stuck rudder could precipitate. Similarly, the 23 other typical faults were integrated and simulated. This work may be very time-consuming. Based on the above works, all of the fault modes that can lead to a stuck rudder could be screened out in Tab. 2, the failure rates (FR) of which are acquired from the design information, as listed in Tab. 2.

On this basis, a Reliability Block Diagram (RBD) is utilised to model the SRF probability of the steering system, as illustrated in Fig. 6.

From the RBD, SRF probability occurring at a particular time without considering maintenance can be acquired. If the vehicle is expected to constantly work for up to three months (2160 hours), the probability of SRF occurring at 2160 h is calculated at 3.46%. Under this circumstance, the event of SRF occurring at 2160 h could be treated as a random event X, which obeys a 0-1 distribution. Suppose X = 1 when SRF occurs, then

\[
P(X = k) = 0.0346^k \cdot 0.9654^{1-k}, \quad k=0,1
\]

MANOEUVRING LIMITATION DIAGRAM (MLD)

MLD regulates the maximum submerging or emerging induced by the rudder angle as determined using speed, depth and trim. The MLD under a specified trim and sea depth consists of (1) a horizontal axis (speed U), (2) an ordinate axis (depth h), (3) a limitation curve of a stuck emerging induced by the stern rudder, and (4) a limitation curve of a stuck submerging induced by the stern rudder. A typical MLD of an underwater vehicle is illustrated in Fig. 7 (sea depth is 300 m, trim is 0°). The region of “speed U(t)-depth h(t)” could be divided into the following subregions [26]:

<table>
<thead>
<tr>
<th>Component</th>
<th>Fault mode</th>
<th>Failure rates (FR)%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operating unit</td>
<td>Transducer, rotation, gain</td>
<td>5</td>
</tr>
<tr>
<td>Demodulation 1</td>
<td>Current signal</td>
<td>2</td>
</tr>
<tr>
<td>Pump control unit</td>
<td>Proportional or binary working</td>
<td>25</td>
</tr>
<tr>
<td>Pump control power</td>
<td>Pump action failure</td>
<td>5</td>
</tr>
<tr>
<td>Cylinder</td>
<td>Cylinder failure</td>
<td>0.5</td>
</tr>
<tr>
<td>Feedback unit</td>
<td>Gage, signal</td>
<td>4</td>
</tr>
<tr>
<td>Demodulation 2</td>
<td>Current, signal</td>
<td>2</td>
</tr>
<tr>
<td>Power unit</td>
<td>AC/DC 24V power, PCA</td>
<td>2</td>
</tr>
<tr>
<td>AC/DC 12V power, PCA</td>
<td>1.5</td>
<td></td>
</tr>
</tbody>
</table>
Safe region. In this subregion (with any speed or depth), although an SRF of submerging or emerging occurs when the stern rudder is at the maximum permitted angle, the vehicle could be recovered from colliding with the seabed or emerging from underwater via other available rudders (bow rudder and direction rudder) and reasonable emergency operations. The trim of the vehicle could be controlled within the permitted maximum trim $\theta_{max} = \pm 30^\circ$.

(2) Emerging limitation region. In this subregion with a speed of $U_0$, if an SRF of emerging occurs at the maximum permitted angle, the vehicle will inevitably rise to the surface or emerge from underwater.

(3) Submerging limitation region. In this subregion with a speed of $U_0$, if an SRF of submerging occurs at the maximum permitted angle, the vehicle will inevitably descend toward or even collide with the seabed.

To obtain the MLD of an underwater vehicle, thousands of combinations of factors, such as the depth, speed, stuck rudder angle, trim and recovering operation, must be simulated and calculated. Though it is difficult to obtain the MLD, its detailed process is not a focus of this paper. However, the SRF of emerging is considered to be safe in this paper. That is to say, only the SRF of submerging could lead to risk (colliding with the seabed), which will be assessed in the following part.

HUMAN RELIABILITY MODELLING

When an SRF of submerging occurs in the steering system, various circumstances, such as depth and speed, may variously influence the operator. Obviously, if the current circumstance is very dangerous (close to the limitation), the operator may be so nervous that they may fail to conduct recovering operations to prevent a fatal accident. Thus, it is very necessary to model human error, which is another random event.

Firstly, to describe the severe level of the current circumstance when the fault occurs, a “fault threaten indicator” $\lambda \in [0,1]$ is proposed in this paper. The larger the $\lambda$, the more dangerous the circumstance, which means that it is more adverse for carrying out recovering operations. Notably, $\lambda$ is determined by the current condition parameters when a fault occurs. An experience-based calculation formula of $\lambda$ is given as follows:

\[
\lambda = \lambda_x + \left( \lambda_y \cdot \lambda_z \cdot \lambda_u \cdot \lambda_d \right)^{1/4} = \lambda_x + \left[ \frac{1}{2} \frac{\delta_{max} \cdot h}{\theta \cdot U_{max}} \cdot \left( \frac{1 - \theta_{max}}{\theta_{max}} \right)^{1/4} \right]^{1/4}
\]

Secondly, to describe the influence of $\lambda$ on human error, a “mental influence indicator” $\eta \in [0,1]$ is proposed in this paper. A larger indicates increased psychological pressure affecting the operator, thereby indicating an increased probability of human error. $\eta=1$ stands for a circumstance in which the psychological condition of the operator begins going against the recovering operations, which means the “mental stress” grade of the performance-shaping factors (PSFs) that influences the human reliability is only qualified to be 0.6 (Considering the strict selection and thorough training of the operator).

Suppose that $\lambda$ and $\eta$ have a relationship of an exponential positive correlation,

\[
\eta = \frac{e^{\lambda} - 1}{e - 1}
\]

Many techniques have been proposed for estimating human error probabilities (HEPs), including HEART (human error assessment and reduction technique), THERP (technique for human error rate prediction), and SLIM (success likelihood index methodology) [27]. Analytic hierarchy process (AHP) [28] is referred to in this paper. To simplify the research, detailed process of the method is not focused in this paper and HR is taken as a unary function of the mental stress grade. Other PSFs are assumed to be constant in the recovery process. Based on the analysis result of AHP and expert’s experience, the weight of mental stress is supposed to be 0.15 and the human reliability in a normal condition is supposed to be 0.999. Thus, the HR of a single operation considering the mental stress influence of fault threat is

\[
HR = 0.999 - 0.15 \times (0.6) \times \eta = 0.999 - 0.06 \eta
\]

If SRF of submerging occurs, different recovering operation sets should be conducted according to the fault threat of the current circumstance, as listed in Tab. 3. Operation steps S1 to S5 are combined to form anti-sink manoeuvres with an increase in $\lambda$, each step of which obeys the human reliability formula (7).

RESULT AND DISCUSSION

When the underwater vehicle is cruising in the sea, an SRF of submerging may occur under circumstances involving many random factors (PDFs of these are listed in Tab. 4). The PDFs of the first five factors are determined based on experience. Thus, these PDFs may vary according to the vehicle and working place.
Based on the PDFs in Tab. 4 and MLD obtained in 3.3, a Monte Carlo simulation could be conducted. The volume of the sample is 100k. In every simulation trial, a set of random values is used. In one simulation experiment, there are 479 events that are beyond the MLD; the number of events in which a fatal accident occurs due to human error within the MLD is 79, as illustrated in Fig. 8. Thus, the fatal risk (collision) probability of an SRF of the steering system is \((479+79)/100000=0.558\%\).

The distributions of all samples (left list) and fatal accidents (central list) as a function of stuck rudder angle, sea depth, diving depth, trim and speed are illustrated in Fig. 9. The right list of figures is the risk probability as a function of the five random factors, which is obtained based on the left and central list. The correlation coefficients of risk probability and each factor are also calculated.

The sensitivity of SRF risk to each factor can be inferred from Fig. 9. The distributions of all samples at each interval point are nearly the same for the stuck rudder angle and speed. However, the distributions of fatal accidents present an obvious rising trend, which means a remarkably positive correlation between the risk and the two factors. However, the situations are the opposite for the sea depth and trim. The decreasing trends, however, are not as notable as the above two. It is difficult to determine a distinct relationship between the diving depth and the risk. Therefore, the stuck rudder angle and the speed are the most vital factors in determining the risk of an SRF of the steering system of the underwater vehicle. It is recommended that the vehicle maintains a limited stern rudder angle and cruising speed, which is beneficial for successful recovery when SRF occurs.

**CONCLUSION**

In this paper, a fault risk assessment method for an underwater vehicle steering system based on virtual prototyping and Monte Carlo simulation is proposed; the method is then verified in a case study. Brief conclusions could be drawn as follows.

<table>
<thead>
<tr>
<th>Random factors</th>
<th>Distribution type</th>
<th>Distribution parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sea depth (H) (m)</td>
<td>Discrete uniform distribution</td>
<td>(H \sim U(50,300),) interval=10</td>
</tr>
<tr>
<td>Diving depth (h) (m)</td>
<td>Continuous uniform distribution</td>
<td>(h \sim U(30,H))</td>
</tr>
<tr>
<td>Trim (\theta) (°)</td>
<td>Discrete uniform distribution</td>
<td>(\theta \sim N(0,2),) interval=1</td>
</tr>
<tr>
<td>Speed (U) (kn)</td>
<td>Discrete uniform distribution</td>
<td>(U \sim U(3.24),) interval=1</td>
</tr>
<tr>
<td>Stuck rudder angle (\delta) (°)</td>
<td>Discrete uniform distribution</td>
<td>(\delta \sim U(-30,30),) interval=5</td>
</tr>
<tr>
<td>SRF</td>
<td>(0-1) distribution</td>
<td>(P = 0.0346^\delta \cdot 0.9654^k), (k=0,1)</td>
</tr>
<tr>
<td>HR</td>
<td>(0-1) distribution</td>
<td>(HR_c = (0.999 - 0.06\eta)^n), (n = 2,3,4,5)</td>
</tr>
</tbody>
</table>
(1) The establishment and V&V of virtual prototyping is an important basis of the proposed method. The credibility of fault simulation data is directly determined using the similarity between the virtual prototyping and the actual system. Notably, instead of being demanded for all aspects, this similarity could be qualified for one particular use.

(2) Obtaining MLD is another important basis of the proposed method, which requires substantial calculation work. Different vehicles should have their own MLDs (except for extreme circumstances in which a fault leads to inevitable accidents); however, the nature of the MLDs may substantially differ.

(3) The distribution information of some important parameters of the vehicle is very important for fault risk assessment. In the case study of this paper, the PDFs of the five random factors are determined based on experience, which may not exactly fit reality. This may affect the accuracy of the risk assessment result. Therefore, the distribution information for random factors should be catalogued over time.

Overall, the method for fault risk assessment of an underwater vehicle steering system based on the combination of virtual prototyping and Monte Carlo simulation proposed in this paper fully incorporates the advantages of the simulation: high efficiency, low cost and infinite flexibility. The relationships between the three elements of the HME system and their influence on risk are comprehensively considered. This method provides a novel solution for fault risk assessment of vehicles and other general HMS systems. The primary shortcoming of this method is its large computation complexity. Further work should focus on efficiently obtaining MLDs for different vehicles and more accurate modelling of steering system faults.

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