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This journal is devoted to designing of diesel engines, gas turbines and ships’ power transmission systems containing these engines and also machines and other appliances necessary to keep these engines in movement with special regard to their energetic and pro-ecological properties and also their durability, reliability, diagnostics and safety of their work and operation of diesel engines, gas turbines and also machines and other appliances necessary to keep these engines in movement with special regard to their energetic and pro-ecological properties, their durability, reliability, diagnostics and safety of their work, and, above all, rational (and optimal) control of the processes of their operation and specially rational service works (including control and diagnosing systems), analysing of properties and treatment of liquid fuels and lubricating oils, etc.

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ASSESSMENT OF UNDERPOWERED PROPULSION MACHINERY IN ELECTRICALLY DRIVEN SMALL INLAND WATERWAY PASSENGER SHIPS FROM CLASSIFICATION SOCIETY POINT OF VIEW

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Abstract
Paper presents short operational and engineering analysis of underpowered propulsion in small electrically propelled small inland passenger ships. There is evidence that in certain weather conditions the phenomena of added aerodynamic resistance of small water crafts may have serious influence on their speed and manoeuvrability. Existing regulations like class societies rules for ship classification and construction or EU Directive 2006/87/EC do not provide any requirements or guidelines on prediction of air or hydrodynamic resistance or propulsion power computations to be assessed by third party in design process. In the opinion of authors, the case is particularly important when electrical or hybrid propulsion is considered as prime mover. Existing knowledge allows for engineering analysis to be conducted to provide better knowledge on the selection and construction of innovative propulsion machinery for ships where passengers safety is major factor of concern by waterways administration, class societies and insurance institutions.

Key words: inland waterways, passenger ship, electrical, hybrid, propulsion power, aerodynamic resistance

Introduction
Design of inland waterways crafts is considered easy even for students and freshly graduated engineers. The use of existing knowledge and relatively simple computation leads to fast design and selection of certain equipment quite easily as it is often available from the shelf. The use of more advanced analytical tools in case of novel ship hull and propulsion designs is often neglected. Operational experience shows that in some cases the fact of underpowered designs will happen. In worse cases, there is no possibility for design improvement without investment in more powerful propulsion system. Alternatively, calculations similar that are performed for bigger ships should be performed using knowledge accumulated by naval architects. In this case, the main obstacle is budget limit.

Decisions to use propulsion machinery selected on principle of lowest computed power installed may lead to operational problems in reality like inability to sail forward in windy conditions or poor maneuverability. Consequently, small or light crafts scheduled for
operation in restricted waters or rivers could experience navigation problems in case of combined wind action and waves. A practical example in PRS experience could be fact of crafts repowering and change of initially installed electrical propulsion to diesel drives. The paper is an attempt to collect some recent class society and engineering experience on operational consequences of underpowered waterborne crafts. More engineering effort and use some more advanced design tools could lead to partial solution to this problem. Authors suggest possible solution to predict and take right measures to improve ship design either by aerodynamic drag reduction or by increase of power and/or propulsion efficiency.

**Short outline of class society requirements and other legal regulations**

Classification rules do not impose requirement on ship hydrodynamic performance. The speed of small recreational or commercial boats with electric propulsion is within range of 6-8 km/h [3]. In case of commercial ships built to comply with class rules general requirement is that minimal speed of inland waterway ship or watercraft is not to be less than 13 km/h. [5]. Such requirement is also present in EU Directive laying down minimal technical requirements for inland waterway vessels [1].

The current state of the class requirements is that the applicable class rules (LR, BV, GL, PRS or RRR) do not contain any formula that allow to define ship power requirements. On the other hand there is given requirement for of 70% of nominal thrust to be available during minimum of 30 min in case of astern movement maneuvering. In general all new ships are to be subjected to maneuverability testing where ship speed is to be measured. Dedicated requirements are pending for all ships that are navigating on EU inland waterways according to the requirements of the 2006/87/EU Directive [1].

**Small ship design and engineering in practice**

In Poland, during last 12 years number of design projects of small passenger crafts with electrical propulsion has been conducted and some resulted in construction of prototype vessels. It is common in some circles to claim that design and construction of small ship, particularly for use on restricted waters is easy and unproblematic. That is not true and there is real evidence that numerous problems may appear. Some originates from improper design input data or wrong assumptions and omitting of engineering computations. Lack of design experience is also a potential risk for design process quality. Selected vessels are listed in the table underneath. As demand for such crafts is limited there are only few example of ships for more detailed analysis. For this study only polish designs were taken into account.

**Class requirements for operational area and wind loading**

A summary of environmental requirements present in class rules for ships navigating in different operational areas is presented in Table 1. Compliance with that regulations is a part of ship class notation.

<table>
<thead>
<tr>
<th>Area Notation</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>PRS Rules Part 1 [5]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wind [ºB]</td>
<td>5 - 6</td>
<td>Up to 6</td>
<td>3 - 4</td>
<td>-</td>
<td>3.6.3.4</td>
</tr>
<tr>
<td>Wind speed [m/s] - max</td>
<td>10</td>
<td>10 - 12</td>
<td>7</td>
<td>-</td>
<td>3.6.3.4</td>
</tr>
<tr>
<td>Wave height [m]</td>
<td>2</td>
<td>1.2</td>
<td>0.6</td>
<td>-</td>
<td>3.6.3.2</td>
</tr>
<tr>
<td>Minimal ship speed [kn(m/s)]</td>
<td>6 (3)</td>
<td>7</td>
<td>-</td>
<td>-</td>
<td>3.6.3.5</td>
</tr>
<tr>
<td>Distance from shore [miles]</td>
<td>Up to 6</td>
<td>3ºB/6 4ºB/1.5</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

Wave height ($h_{1/10}$) is measured from through to the crest representing value of 10% of highest waves during the particular – not to long measuring period. This corresponds 5%
probability of exceeding that height.

**Ship resistance and propulsion system design**

In majority cases, designers conduct in practice very simple calculations, not taking all engineering parameters into account. During ship design process for assessment of ship performance, it is typical to assess water resistance. Usually omitted, determination of resistance resultant from aerodynamic forces is to be done to be able to select prime mover power to comply with area. The importance of wind knowledge about resistance is in often critical and could result in serious financial penalties for shipyard and costs for ship owner. Recently done combined shipping industry project resulted in elaboration of new guidelines for bigger ships. With the assistance of the Sea Trial Analysis-Joint Industry Project (STA-JIP) and ITTC, the new IMO EEDI rules to reduce CO2 emissions. As a result, there is validated and approved method to consider the effects of wind on the resistance given in the ISO Standard 15016 (2002) and the ITTC Procedure 7.5-04-01-1.2.

For preliminary design of small ships, majority of calculations done are based on simple formulas. Performing full resistance assessment with model testing for hull and propeller require substantial budget and this is performed very rarely in case of small inland crafts. The analyzed ships discussed here possess relatively small electrical motor power in comparison to similar size crafts powered by diesel engines.

**Assessment of ship aerodynamic resistance**

The computations of the ship aerodynamic resistance is usually performed when necessary – for mooring analysis or on big fast ferries – HSC (High Speed Crafts), container ships or on naval vessels. There is no any information that aerodynamic resistance calculations were performed in the discussed inland waterways passenger ship examples. In case of windless weather conditions, the wind resistance acting on a ship sailing forward can be determined using the formula:

\[
F_d = \frac{1}{2} \cdot \rho \cdot V_s^2 \cdot A \cdot C_d
\]

where:
- \( F_d \) - the drag force,
- \( \rho \) - the density of the air,
- \( V_s \) - the speed of the ship relative to the air,
- \( A \) - the cross-sectional area of the ship,
- \( C_d \) - the dimensionless coefficient of resistance, dependent basically on physical parameters, ship hull and superstructure shape and spatial dimensions and on ship speed relative to the air.

Majority of the wind resistance is due to eddie-making type, and therefore it varies roughly with \( V_r^2 \) (\( V_r \) is the relative velocity of air to a ship) what is presented in Fig. 1.
$A_L$ – longitudinal projected area,
$A_T$ – transverse projected area,
$V_w$ - wind speed,
$V_s$ – ship speed,
$V_R$ – wind velocity relative to ship,
$\theta$ – angle of wind direction relative to the longitudinal centre line of a ship measure from the bow,
$\alpha$ – direction of the resultant force relative to the centre line

According to Hughes formula

$$F_d = k \cdot \rho \cdot V_R^2 \cdot \frac{A_L \cdot \sin^2 \theta + A_T \cdot \cos^2 \theta}{\cos(\alpha - \theta)}$$

Where: $k$ – an empirical constant $k = (0,5 - 0,65)$

The additional propulsion power required to overcome the aerodynamic drag is given by:

$$P_d = F_d \cdot V_R = k \cdot \rho \cdot V_R^3 \cdot \frac{A_L \cdot \sin^2 \theta + A_T \cdot \cos^2 \theta}{\cos(\alpha - \theta)}$$

**Design and construction problems in small ships design**

Requirements and procedure for testing of the new build inland waterway ships is described in Rules [7] and is based on EU Directive requirements. As a result of analysis of published information about projects and existing inland ships and ferries [6], specifics of electrically propelled vessels could be summarized as:

1. Relatively small size - 6 to 24m,
2. Lightweight design with hull made of steel, GRP composites or aluminum alloys,
3. Number of passengers - 6 to 100 (150 max),
4. Mono-hull or quite often twin hull design,
5. Energy storage (battery) weight is substantial comparing to hull weight

As an example of design issues that are to be analyzed, enclosed are diagrams with analysis of power requirements and weight estimations:
Fig. 2 Operational time and range as function of solar craft speed – design analysis example. [3]

Fig. 3 Weight distribution in relation to construction material in small water craft design [3]

Fig. 4 An example of weight relation of catamaran crafts in a function of construction materials and ship length [8]
The above graphs point out that the hull and equipment weight of the small craft shall be kept at lowest possible value due to important influence on energy required for propulsion. Additionally, prediction of operational characteristics in case of battery powered should be conducted to analyze energy usage for planned usage timeframe. Excessive use of energy may lead to lack of propulsion power in case of change of weather conditions when additional aerodynamic drag require more energy to be used to keep speed or operation of the vessel systems. The safety measures are present in class society rules – usually spread in various parts of rules. Synergy of safety requirements is not always properly understood by designers, fresh ship owners and water administration.

Maneuverability testing is described in EU Directive Amendments [1] and the all requirements are included into PRS Rules [7]. The requirement for minimal speed at least 13 km/h is enclosed in Part III p. 2.2.1 of PRS Rules [5]. The requirement for astern thrust 70% of forward thrust by 30 min. is in part VI p.1.10.2. In certain cases, there could be request to increase the astern thrust due to safety or maneuverability reasons. Full maneuverability requirements are enclosed in PRS publication no 27/P-2010 [7]. The maneuverability requirements are not easily implemented by ship-owners due to budget reasons and sometimes due to lack of test area. Having in mind that, ship operational parameters should be constant during all day navigational scheme, it is very risky not to install additional energy power in case of battery powered ships. A solution requested by class society rules is to have additional source of energy i.e. diesel generating set.

Selected ship examples

Ship A.

Mono-hull ship – The first bigger passenger ship in Poland equipped with electrical propulsion. The ship was used for short trips on lakes and river estuaries. According to the available information the design appeared to create some technical and operational problems – windy weather prevented to be used for tourist trips. Very soon operational use has been suspended. Ship owner decided to make major modification of the propulsion system and removed electrical drive into diesel drive. The vessel was prototype with limited founds and probably without detailed design that should include detailed calculations for battery powered ship propulsion system. We can assume that in case of this passenger ship having steel design with construction same as for typical inland waterway ship with diesel propulsion was too heavy. Battery powered vessel appeared to be uneconomical, slow speed, difficult to make maneuvers at slow speed so ship owner decided to conduct propulsion machinery conversion to diesel drive.

Ship B.

Twin hull small passenger ship used as river ferry or water tram used in restricted water areas like rivers and channels. Advanced design with solar cells appeared economical solution maximal speed during testing within range defined by Rules. Ship power system consists of two battery banks and diesel and solar generator. Electrical motors place in machinery compartment above waterline and power transmission to the propeller is made using “L” drive. The ship is in operation starting from Sept 2008.

Ship C.

Small solar powered boat of 6m length. Experimental design constructed by a team from University of Technology. Boat performed several trips on polish rivers and channels as well as some trips at sea in Gdańsk Bay. The electrical motors propulsion power is very low (2 kW) and in windy conditions some problems of not sufficient power has been observed.
Ship D

Ship E
Hybrid, electrically and solar energy powered passenger ship for inland waterways. Completed research project and selected engineering analysis. Model built and testing has been done. The craft designed to be operated in estuary of Vistula river inland waterway channels.

Assessment of wind resistance for selected ships with electrical propulsion
Evaluation of wind generated aerodynamic resistance has been done for sample ships using technical data available in literature and PRS Register of Inland Waterways Ships. Areas of wind action has been assessed using ship general plans and officially available dimensional data. Formulas used are those presented above. Results of computations are presented in Fig. 5.

Tab 2. Basic data of analyzed passenger ships

<table>
<thead>
<tr>
<th></th>
<th>SARA</th>
<th>Ship A</th>
<th>Ship B</th>
<th>Ship C</th>
<th>Ship D</th>
<th>Ship E</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hull material/shape</td>
<td>Cat/Steel</td>
<td>Monohull/Steel</td>
<td>Cat/GRP</td>
<td>Cat/GRP</td>
<td>Monohull/GRP</td>
<td>Monohull/Al</td>
</tr>
<tr>
<td>Passengers/crew</td>
<td>122</td>
<td>78+2</td>
<td>28+2</td>
<td>10+2</td>
<td>28+2</td>
<td>55+3</td>
</tr>
<tr>
<td>Length [m]</td>
<td>18.0</td>
<td>20.63</td>
<td>13.46/11.40</td>
<td>6</td>
<td>13.9/12.71</td>
<td>20/19.5</td>
</tr>
<tr>
<td>Beam max [m]</td>
<td>4.5</td>
<td>4.02</td>
<td>3.02/3.22</td>
<td>2.5</td>
<td>3.25</td>
<td>5.7</td>
</tr>
<tr>
<td>Height over waterline [m]</td>
<td>-</td>
<td>3.94</td>
<td>2.97</td>
<td>2.7-2.8</td>
<td>2.7</td>
<td>4.4</td>
</tr>
<tr>
<td>Hull height [m]</td>
<td>-</td>
<td>1.41</td>
<td>1.32</td>
<td>0.4</td>
<td>0.4</td>
<td>0.47</td>
</tr>
<tr>
<td>Depth [m]</td>
<td>-</td>
<td>0.64</td>
<td>0.36</td>
<td>0.4</td>
<td>0.4</td>
<td>0.47</td>
</tr>
<tr>
<td>Displacement [dm³]</td>
<td>33.3</td>
<td>7500</td>
<td>28000</td>
<td>7500</td>
<td>28000</td>
<td>7500</td>
</tr>
<tr>
<td>Weight/Hull height</td>
<td>32.4</td>
<td>2500</td>
<td>2500</td>
<td>2500</td>
<td>2500</td>
<td>2500</td>
</tr>
<tr>
<td>Cross-section area over-water [m²]</td>
<td>15.84</td>
<td>9.5</td>
<td>5.6</td>
<td>8.78</td>
<td>25.08</td>
<td></td>
</tr>
<tr>
<td>Cross-section area longitudinal [m²]</td>
<td>36.35</td>
<td>26.4</td>
<td>9.1</td>
<td>31.65</td>
<td>56.63</td>
<td></td>
</tr>
<tr>
<td>Prop power [kW]</td>
<td>11</td>
<td>2 x 8</td>
<td>2 x 8</td>
<td>2 x 1</td>
<td>2 x 9.8</td>
<td>2 x 12 kW</td>
</tr>
<tr>
<td>Speed [km/h]</td>
<td>9.9</td>
<td>10</td>
<td>12</td>
<td>8-10</td>
<td>12</td>
<td>12-15</td>
</tr>
<tr>
<td>Battery</td>
<td>Yes</td>
<td>Yes</td>
<td>Yes</td>
<td>220 kWh</td>
<td>20,48V/1000Ah</td>
<td>Yes</td>
</tr>
<tr>
<td>Autonomy min.</td>
<td>8-9</td>
<td>8-10</td>
<td>8h</td>
<td>8h</td>
<td>8h</td>
<td>8h</td>
</tr>
<tr>
<td>Curve no</td>
<td>Serie 2</td>
<td>Serie 3</td>
<td>Serie 4</td>
<td>Serie 5</td>
<td>Serie 5</td>
<td>Project</td>
</tr>
</tbody>
</table>
Risk of under-estimation of ship power requirements

Design calculations should be done bearing in mind all aspects related to the propulsion power requirements. In practice there are only draft calculations performed and analysis of hydrodynamic resistance assuming certain – roughly estimated efficiency or not taking into account all factors that have influence on resistance and propulsion system efficiency. An example of Solar Boat designed on Gdańsk University of Technology is rare example of complex design analysis that included also testing in model tank.
Solution to the problem of under-powered propulsion of small inland waterway crafts.

Class society acceptable solution to the problem of under estimation of power requirements could be proposed based on:

- Introduction of requirement to provide classification society with computation results for propulsion system to verify correctness of selection of prime mover. Data needed for verification could be minimal power for still water, minimal power for maximal environmental loads that are used by class as figures places in all ship safety documents – Class certificate. Class society can verify if input data used for selection of propulsion power are satisfactory from rules point of view.

- Introduction of certain formulas into the rules for ship classification and construction. In this case the formulas should be based on naval architecture and experimental experience i.e. using agreed ITTC guidelines and formulas.

Additionally, we can summarize practical aspects of underpowered vessel to the following conclusions:

1. The ship smaller speed than value set by ship owner could have negative effect (i.e financial) on project compliance with figures set as design data.

2. Achieved speed lower than 13 km/h could cause a problem with obtaining of EU inland waterways class certificate. Minimal speed required by Council Directive 82/714/EEC is forward speed. It is obvious that astern speed and in fact effective thrust will be smaller as well.

3. Worse manoeuvring characteristics measured during trials (normally to be done on still water) in fact will be worse when combined action of wind and waves will occur.

As the minimum speed for inland ships is a part of class requirements the non-compliance with that regulation is usually a reason for derogations. Real consequence of underpowered vessel may result in the necessity of replacement of specific machinery of components.

Propulsion upgrade may include:

- installation of stronger motor as well as modification of propeller,
- installation of more efficient batteries.

In case of small craft with limited space and buoyancy, some of such activities may be a tough task. Change of whole drive may be very expensive and could pose number of additional modifications and will cost extra money, time and also loss of reputation.

Summary

1. Innovative ship design with electrical propulsion and new energy sources require more knowledge to perform design due to the fact that detailed computations or simulations ought to be conducted. There is no such requirements in the existing ship classification rules for inland waterway crafts. EU Directive 82/714/EEC do not have requirements for propulsion calculations and only minimal vessel speed is required to be checked by class or inspection bodies to allow Navigation Documents to be issued. It is recommended, to provide such data to the ship classification society for verification purposes on voluntary basis for all ships. Using the results of our study, such approach ought to be mandatory for passengers ships and ferries, especially of innovative design.

2. Taking into account the trend towards propulsion efficiency, the available knowledge and engineering tools approach should be formally included into Rules. Action to implement new requirements for checking vessels propulsion efficiency and energy conservation measures to obtain efficient propulsion shall be promoted. Having
requirements for manoeuvrability prediction or testing included into the requirements of EU Directive, the extra effort and related cost to perform such calculations for new ships can be neglected as minor. Now-days it is not complicated to conduct such quite complex calculations due to availability of software design tools and quite large amount of data for design analysis.

3. The proposed new requirements should be mandatory for all ferries or passenger ships with hybrid or pure electric electrical propulsion where batteries are used for storing energy for propulsion. Propulsion calculations (power, thrust requirements, energy consumption or conservation …..) with additional aerodynamic resistance calculations should be required for estimation of the ships manoeuvrability performance in extreme and possibly all operational conditions. Maximal values should be calculated and included into booklet for the use of ship master.

4. Elaboration of digital model of electrically propelled ship aerodynamic resistance during navigation could be further used for development of complex vessel energy conservation and management system for electrical battery powered surface craft or ship. The energy management system will be an essential part of ship propulsion control system allowing to compute and process data necessary for optimal navigation, operational planning and final decision taking either by ship crew or by automatic energy management system.

5. The presented findings of short study of underpowered propulsion system in small electric powered crafts is an example of identification of potential problems that may occur in case of omitting some environmental phenomena in design phase in case of innovative ship design. The problem is worth of further exploration using better selected resistance coefficients as well as real information describing ship structure and resistance/power data used for computations. The elaborated solution should be implemented into engineering guidelines.

References


THERMODYNAMIC ANALYSIS OF A GENERATION IV SCWR
NUCLEAR POWER PLANT CYCLE

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Abstract

This article presents a flow and thermodynamic analysis of a Generation IV nuclear cycle. An SCWR (Supercritical water reactor) is a high temperature and high pressure reactor that uses water at a temperature above its thermodynamic critical point as the working fluid. The cycle used for the calculations consists of one interstage superheater and 7 regenerative heat exchangers. Division pressure was optimized in view of the cycle efficiency, and the possibility of using another pressure value that would be more beneficial due to the structure of the grid of blades was mentioned.

Keywords: nuclear reactors, steam turbines, thermodynamic cycles, SCWR reactors

1. Introduction

Nuclear power engineering is currently on the rise; there are 66 nuclear power units under construction[1]. Because of the increasing interest in energy generation through nuclear fission, research is being conducted on the development of a new generation of nuclear reactors (Figure 1). Generation III reactors are currently being constructed, while works are pending on Generation IV that is to be characterised with:

- increased safety of reactors operation;
- increased proliferative immunity;
- minimization of radioactive waste generation;
- reduction of reactors construction and operation costs.
Evolution of Nuclear Power

Generation IV reactors are expected to be put into operation following 2030. In this article a supercritical water reactor (SCWR) is the subject of a thermodynamic analysis. An SCWR reactor operation is based on two proven technologies: pressure reactors and conventional supercritical power units. The aim of SCWR reactors is to generate relatively inexpensive electric energy. A pictorial diagram of an SCWR reactor is given in Figure 2.
2. Thermodynamic cycle used for calculations

A thermodynamic cycle consists of a reactor in which water steam with the following parameters: $t_1 = 570^\circ\text{C}$ and $p_1 = 260$ MPa is generated. Temperature at the reactor inlet is $250^\circ\text{C}$. Steam enters a high pressure (HP) turbine where it expands to $p_3$ pressure that has been optimized in view of maximum efficiency of the cycle. Following expansion in the HP section steam enters the superheater. The cycle consists of 7 regenerative heat exchangers, 3 of which are supplied from the intermediate pressure (IP) section, while the next 4 are supplied from the low pressure (LP) section. A schematic diagram is given in Figure 3. The operation of a circulating pump was assumed for the calculations.

![Diagram used for the SCWR reactor calculations](image)

Table 1 shows the values used for the calculations. The calculations were based on the generator shaft power, number of regenerative heat exchangers, efficiency of individual elements of the cycle.

**Table 1. Values used for the calculations**

<table>
<thead>
<tr>
<th>No.</th>
<th>Parameters</th>
<th>Designation</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Power</td>
<td>$N_e$</td>
<td>MW</td>
<td>182</td>
</tr>
<tr>
<td>2.</td>
<td>Pressure downstream of the reactor</td>
<td>$p_1$</td>
<td>bar</td>
<td>260</td>
</tr>
<tr>
<td>3.</td>
<td>Temperature downstream of the reactor</td>
<td>$t_1$</td>
<td>$^\circ\text{C}$</td>
<td>570</td>
</tr>
<tr>
<td>4.</td>
<td>Temperature downstream of the superheater</td>
<td>$t_4$</td>
<td>$^\circ\text{C}$</td>
<td>570</td>
</tr>
<tr>
<td>5.</td>
<td>Pipelines efficiency</td>
<td>$\eta_k$</td>
<td>-</td>
<td>0.95</td>
</tr>
<tr>
<td>6.</td>
<td>HP section efficiency</td>
<td>$\eta_{HP}$</td>
<td>-</td>
<td>0.92</td>
</tr>
<tr>
<td>7.</td>
<td>IP section efficiency</td>
<td>$\eta_{IP}$</td>
<td>-</td>
<td>0.94</td>
</tr>
<tr>
<td>8.</td>
<td>LP section efficiency</td>
<td>$\eta_{LP}$</td>
<td>-</td>
<td>0.9</td>
</tr>
<tr>
<td>9.</td>
<td>Pump efficiency</td>
<td>$\eta_P$</td>
<td>-</td>
<td>0.8</td>
</tr>
<tr>
<td>10.</td>
<td>Pressure in the condenser</td>
<td>$p_k$</td>
<td>bar</td>
<td>0.05</td>
</tr>
<tr>
<td>11.</td>
<td>Rotational speed</td>
<td>$n$</td>
<td>RPM</td>
<td>3000</td>
</tr>
<tr>
<td>12.</td>
<td>Number of regenerative heat exchangers</td>
<td>$z$</td>
<td>-</td>
<td>7</td>
</tr>
</tbody>
</table>
3. **Optimization of division pressure downstream of the HP turbine**

First of all, $p_3$ pressure was optimized in view of the cycle maximum efficiency. Maximum efficiency was obtained for $p_3 = 130.91$ bar, i.e. 0.53 of $p_2$ pressure. In references it may be found that the optimum value of $p_3$ pressure is within 0.2 and 0.25 of $p_2$ for conventional systems [5]. The efficiency for such a pressure value amounts to 48.2%. The curve of steam cycle division pressure in the function of efficiency is given in Figure 4. Further calculations were performed for various ratios of $p_2/p_3$ (Table 2) because of a minor difference in efficiency. Another value of $p_3$ pressure may be recommended due to structural reasons. Figure 5 shows enthalpy reduction at individual sections of the turbine. With the increase of $p_3$ pressure the HP section share drops, and the IP section share increases.

**Table 2. Pressure and cycle efficiency**

<table>
<thead>
<tr>
<th>No.</th>
<th>$p_3/p_2$</th>
<th>$p_3$ pressure [bar]</th>
<th>Cycle efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>0.3</td>
<td>74.1</td>
<td>0.476224</td>
</tr>
<tr>
<td>2.</td>
<td>0.4</td>
<td>98.8</td>
<td>0.479906</td>
</tr>
<tr>
<td>3.</td>
<td>0.53</td>
<td>130.91</td>
<td>0.481577</td>
</tr>
<tr>
<td>4.</td>
<td>0.6</td>
<td>148.2</td>
<td>0.481372</td>
</tr>
</tbody>
</table>

*Fig. 4. $p_3$ pressure optimization curve*
Figures 6, 7, 8, 9 show a percentage share of enthalpy reduction at individual cylinders of the turbine. Similarly to Figure 5, a decreased share of enthalpy reduction of the high pressure section and an increase in the case of the intermediate pressure section may be observed, while the share of enthalpy reduction of the low pressure section remains practically unchanged for all the analysed division pressures.

Fig. 5. Diagram of enthalpy reduction for various values of $p_3/p_2$ pressure

Fig. 6. Percentage distribution of enthalpy reduction at individual parts of cylinders for a division pressure ratio of 0.3

Fig. 7. Percentage distribution of enthalpy reduction at individual parts of cylinders for a division pressure ratio of 0.4

Fig. 8. Percentage distribution of enthalpy reduction at individual parts of cylinders for a division pressure ratio of 0.53

Fig. 9. Percentage distribution of enthalpy reduction at individual parts of cylinders for a division pressure ratio of 0.6
4. Flow channel design

The flow channel was designed for four division pressure variants. Because of the division pressure changes, the number of stages in the high pressure and intermediate pressure cylinders changed. The changes did not affect the low pressure cylinder.

Diagrams 10, 11, 12 and 13 show the shape of the high pressure channel for varying values of $p_3$ pressure; the number of stages drops with the increase of the division pressure. With the lowest division pressure there are eight stages. With the optimum division pressure value there are only four stages.

![Flow channel shape for a division pressure ratio of 0.3](image1)

![Flow channel shape for a division pressure ratio of 0.4](image2)

![Flow channel shape for a division pressure ratio of 0.53](image3)

![Flow channel shape for a division pressure ratio of 0.6](image4)

The number of stages in the IP channel rises from 12 to 16. Total number of stages of the HP and IP section for efficiency optimum pressure is 19, one less than in the case of division pressure lower than optimum.

![Flow channel shape for a division pressure ratio of 0.3](image5)

![Flow channel shape for a division pressure ratio of 0.4](image6)
The number of cylinders was calculated for the low pressure section in accordance with the algorithm presented below. The steam mass stream flowing through the last stage of the LP turbine:

\[ \dot{m} = 117,198 \frac{kg}{s} \]  \hspace{1cm} (1)

Specific volume of steam downstream of the turbine:

\[ v = 23,36 \frac{m^3}{kg} \]  \hspace{1cm} (2)

Turbine outlet velocity:

\[ c_2 = 250 \frac{m}{s} \]  \hspace{1cm} (3)

Maximum permissible bending stress affecting the blade:

\[ \sigma_{gr} = 350[MPa] \]  \hspace{1cm} (4)

Assumed ratio:

\[ k = 0,383[-] \]  \hspace{1cm} (5)

Blade material density:

\[ \rho = 8000 \frac{kg}{m^3} \]  \hspace{1cm} (6)

Rotational speed:

\[ \omega = 314[RPm] \]  \hspace{1cm} (7)

Boundary surface:
Effective area:

\[ \Omega_{gr} = \frac{\sigma_{gr} \cdot 10^6 \cdot 2\pi}{k \cdot \rho \cdot \omega^2} = 7,279[m^2] \]  

(8)

Number of cylinders:

\[ a = \frac{\Omega}{\Omega_{gr}} = 1.5 \approx 2 \]  

(10)

The low pressure turbine was divided into two cylinders. The steam mass stream flowing through the turbine and bleeders mass streams were divided into two parts.

![Flow channel shape](image)

*Fig. 18. LP flow channel shape*

5. Summary

Analysing the obtained results we may conclude that the idea of SCWR reactors may constitute an inexpensive and effective source of electric energy generation. The calculations also prove that cycles with the above-mentioned reactors have a high efficiency. It should also be highlighted that the optimum division pressure is higher than in the case of conventional cycles. The analysis of flow channels shows that in some cases the adoption of a pressure value other than the optimum one may be more beneficial, due to the structure of the grid of blades.

References


OPERATING PROBLEMS OF TURBOCHARGING SYSTEMS IN COMPRESSION-IGNITION ENGINES

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Abstract

Supercharging of diesel engines is one of the most popular methods of improvement of their operating indexes by influencing their general efficiency, volumetric and mass power coefficient (downsizing), unit exhaust emissions etc. Turbocharging systems are commonly used in compression-ignition engines and spark ignition engines. The most common type of supercharging is turbocharging based on a turbocharger as the main element.

It is a flow machine, in which a turbine rotor and compressor rotor are found on the common shaft. The turbine rotor is propelled with the exhaust gas from the engine while the intake air flows through the compressor rotor. The use of a turbocharger allows utilization of waste energy of the exhaust gas for improvement of parameters of the air supplied to the cylinders from the point of view of fuel combustion (increase of density and turbulence degree).

Due to unfavorable operating conditions such as high temperature of the exhaust gas (for compression-ignition engines – up to approx. 800 ºC and for spark ignition engines – up to approx. 1100 ºC), very high rotational speed (up to approx. 250 000 rpm), precise structure and complex encapsulation, a turbocharger is particularly exposed to damage or failures.

The operating problems of contemporary diesel engine turbocharging systems as discussed in this paper, relating to turbochargers and deriving from research and expert practice of the authors, constitute conclusions of the activities to some extent, which may contribute to further works on the improvement of diesel engine charging systems. This paper presents four types of damage occurring in turbochargers as divided into their causes. According to the division assumed in this paper, the damage is caused by:

- inappropriate lubrication,
- presence of foreign objects,
- exceeding the admissible rotational speed of the turbocharger shaft or excessive pressure and exhaust gas temperature,
- corrosion.

This paper addresses each of the above-mentioned cases and describes causes and effects of the same.

Keywords: compression-ignition engines, damage, turbocharger, turbocharging

1. Introduction

Engine charging involves feeding fresh charge of increased density into the engine cylinders, which enables feeding greater fuel mass (without changing the \( \lambda \) air excess coefficient). Charging aims at increasing of the unit power, efficiency and decreasing the unit exhaust emissions [3]. The increase of the charge density is obtained with an external machine (compressor) or the use of dynamic properties of the engine itself. The improvement of charging (which considerably
improves the engine efficiency), consistent with the improvement of the power energy balance, is based on the increase of the turbocharger output in a wide range of revolutions and taking into account of the engine properties and a reduction of the turbine inertia.

Thus, the evolution of charging systems leads to the improvement of the turbocharger efficiency and opportunities to control the charging pressure regardless of the engine operating conditions. Making a compressor drive independent of the engine parameters makes it possible to precisely control the charging pressure.

Initially, mainly turbochargers fitted with a wastegate were used in direct injected compression-ignition engines, which were later replaced by variable geometry turbochargers. Turbocharging in high-volume vehicle production was first applied in the early 60s in a GM Chevrolet Corvair. Unfortunately, the vehicle was characterized by poor operating indexes during operation at low speed ranges—its turbocharger was highly inertial, which generated a ‘turbolag’, which made smooth driving impossible.

A “turbolag” was the biggest problem in early turbocharged engines and discouraged drivers from using such solutions. Despite the fact that turbocharged engines were used successfully during car races (BMW 2002 F1 formula), passenger cars always required a more evenly distributed supply of power. Turbochargers of that period were large and heavy and began to rotate only at the shaft revolutions of 3500 rpm and their power for low speeds was rather low.

In 1975 Porsche was the first company to manufacture the first turbocharged vehicle operating in a different way. It was Porsche 911 Turbo 3.0 with a mechanism allowing a turbocharger to gain speed before the supercharging actually began. The essence of this mechanism was a recirculation pipe and a valve that started exhaust gas recirculation before the exhaust gas generated adequate pressure for the operation of a turbine. Therefore, the ‘turbolag’ was much shorter and its power increment was more linear.

Three years later, Porsche manufactured a 911 Turbo with 3.3 l engine, which followed the 911 Turbo of 1975. The vehicle was equipped with a intercooler.

Development of charging systems aims at the improvement of the efficiency of turbochargers and the opportunities to control the charging pressure regardless of the engine operating conditions. Making a compressor drive independent of the engine parameters makes it possible to precisely control the charging pressure. Electrically driven compressors are a possibility in the future. Apart from precise control, their additional strong point involves an opportunity to increase the efficiency of the exhaust gas aftertreatment systems by eliminating of a turbine causing a decrease in the temperature of the exhaust gas directed to the aftertreatment systems (NOX and PM in particular). A technological and structural improvement (including the improvement of the inertia parameters) could be achieved by a decrease of the weight of the turbine and compressor rotors. An overview of modern charging system solutions is widely described in the subject literature.

A turbocharger is propelled with exhaust gas and its task is to pump the largest possible volume of air into the cylinders. It is placed in the exhaust manifold as close as possible to the exhaust gas inlet to the manifold so as not to lose the kinetic energy of the exhaust gas for the passage in the exhaust system. Exhaust gas passing with high speed drives the turbine rotor connected with the compressor rotor by a common shaft. Air is sucked mostly by a cone filter and gets straight into the inlet manifold after compression. The turbine rotor rotates with the speed reaching 290 000 rpm. The exhaust gas of the temperature of 900°C and the high rotating speed cause the turbocharger to heat up to very high temperatures. Such extreme operating conditions are very demanding.

A turbocharger has to be precisely manufactured from materials of the highest quality. It consists of an exhaust gas driven turbine and a compressor mounted on the opposite ends of a common shaft and closed in a cast shroud.
An exhaust gas turbine has a turbine wheel and a shroud with an inlet and outlet cast of high-cadmium alloys or heat-resistant alloys based on cobalt and cast with the use of the lost cast waxing technology. An air compressor is composed of a cast compressor wheel and the shroud. Both sections are mounted on the opposite ends of a common shaft and closed in a cast central body of the turbocharger.

A shaft connecting two sections is embedded with the use of a carefully designed system of bearings for obtaining high rotational speeds and, unlike the crankshaft bearings, for small loads. They have to maintain the turbine and compressor wheels in the smallest possible distance from the shroud. The main tool in keeping their position is the space filled with oil found between the shroud, the bearings and the shaft. They have a decisive influence on the efficiency and durability of the turbocharger.

Sealing systems separate the shroud from the turbine and compressor sections. They prevent oil from passing to the operating area of the turbine and compressor and from passing of the exhaust gas to the central area of the turbocharger.

The applied engine oil has to be of top quality. Following the start of the engine and before driving, the engine should reach the operating temperature of approx. 90°C in order to ensure minimum appropriate conditions for lubrication of the turbocharger bearings. The same rule applies when the engine is stopped. Modern vehicles are equipped with a turbo timer preventing immediate engine stop after high-speed driving and showing the time left for the turbine to cool.

One of the basic signals of a failure of the turbocharger is a decrease in the engine power and increased noise during engine operation. Another syndrome of turbine wear is tailpipe smoke. Blue exhaust gas may indicate that engine oil is burnt, which may be caused by a loss of airtightness of the turbine, whereas black exhaust gas may indicate that some fuel remains unburnt and the turbine pumps too little air. There are many reasons for smoke appearance (failure of the injection pump or injectors). Therefore, not every smoke appearance signifies a failure of the turbocharger. A turbocharger failure may also lead to engine overload, which is at best indicated by a light on the instrument cluster and, in extreme cases, may lead to engine damage [1].

Despite the fact that turbochargers are precision devices, their mechanism is relatively simple, strong and effective. High rotational speeds of the rotors and high temperature of the exhaust gas render the turbochargers very sensitive to inappropriate operation. Assuming appropriate operation, a turbocharger can operate without failures for a long period (no less than 200 000 km of the vehicle mileage).

2. Types of turbocharger damage

2.1. Division

Types of turbocharger damage were divided in terms of damage causes. These are:
- inappropriate lubrication,
- presence of foreign objects,
- exceeding the admissible turbocharger rotational speed or excess pressure and exhaust gas temperature,
- corrosion.

The further part of this paper addresses each of the above-mentioned cases and describes causes and effects of the same.

2.2. Inappropriate lubrication

There are several reasons for inappropriate lubrication. First of them is related directly to oil and its filter – a turbocharger is lubricated with oil from the engine oil system. Because of operation in severe conditions and, for example, partial combustion of oil, the oil parameters may
be deteriorated, which results in a change of the oil density, viscosity and resistance to foaming. The oil may be contaminated, which may cause micro-machining, scratching or grooving of the top layers of the assemblies. Contaminated oil causes accelerated wear of the bearings and destruction of the mid part where the bearings are fitted. The oil may have low original quality or be inappropriately selected for the engine. It is also significant to select an appropriate oil filter of adequate parameters. A damaged or congested filter may considerably deteriorate the turbocharger lubrication.

To sum up, a damage of turbochargers may in this case be a result of:
- poor engine oil quality,
- deterioration of the oil quality as a result of partial combustion of oil,
- failure to ensure periodical replacement of oil and its filter,
- congested, damaged or poor quality oil filter,
- failure of the oil filter bypass valve,
- engine wear.

The effects of the damage are in the first stage visible in the form of scratches on the race of the transverse (Fig. 2.1) and longitudinal bearings (Fig. 2.2).

Damage of bearings leads to a seizure of the turbocharger shaft journal (Fig. 2.3). The resulting gaps may cause scratching of the turbine or compressor rotor vanes against the body surface, which results in damage of the same (figs. 2.4).
The compressor (Fig. 2.5) or turbine rotor vanes are also damaged.

As can be observed, a single damage is followed by another. In order to prevent such type of damage, oil and filters should be of good quality and need to be replaced with each repair of the turbocharger. Additionally, oil and filters should be replaced at regular intervals in accordance with the vehicle manufacturer’s recommendations or even more frequently when used in arduous dusting conditions.

Examples of irregularities in the lubricating process also include oil starvation. Precisely matched parts of the turbocharger have to operate in permanent oil film. In case of even small lubrication disorders, rotating and stationary parts may contact, which will lead to serious damage immediately. Aside from lubricating of the cooperating parts, oil also works as a cooling medium.

Reasons for momentary oil starvation:
- inappropriate turbocharger startup after reassembly or repair,
- renewal of oil and its filter,
- a long dwell time in vehicle operation,
- low pressure of oil due to poor functioning of the lubrication system,
- contamination of oil, for example, with coolant or fuel,
- inappropriate engine start in low temperatures,
- operation of the engine in overload conditions (high load for low engine speeds).

Because of oil starvation, especially a prolonged one, the surface of the shaft and the bearings will, get polished and will seize (Fig. 2.6).
Lubrication problems may also be caused by lack of oil or considerable decrease in the oil pressure. Oil pressure deficit is the most serious form of oil starvation incidents and may be attributed to:
- bent, broken or congested turbocharger oil duct,
- faulty oil pump,
- low pressure of oil in the engine oil pan,
- penetration of air into the lubrication system,
- lack of lubrication due to excessively long operation of the vehicle in high terrain inclination conditions.

Considerable shortage of oil pressure for an extended period (exceeding 8 - 10 seconds) causes damage to the turbocharger bearings. This results in polishing and burning of the bearings surface (Fig. 2.7) as well as discolorations of the shaft resulting from significantly increased temperature (Fig. 2.7).

The lack of oil is more dangerous than its instantaneous shortage and more frequently leads to seizures of the surface of the bearings and the shaft, including those of the longitudinal shaft, which consequently leads to a destruction of the turbocharger.

Another lubrication problem is inhibited outflow of oil from the turbocharger. Reasons:
- congested drain duct of the turbocharger (Fig. 2.11),
- blowby in the piston and cylinder system (especially for excessive pressure above 50 hPa),
- congested venting system,
- excessively high oil level.

Disorders in the oil outflow may cause a decrease in the oil pressure and deterioration of the lubrication, which may lead to effects similar to those connected with decreased lubrication pressure or oil starvation.
2.3. The presence of foreign objects

The damage is caused because of penetration of foreign objects (elements of the air filter box or the filter itself, fragments of piston rings or spark plugs) into the exhaust or intake system. Foreign objects may also include tiny fragments of pistons or valves and carbon deposits from the combustion chamber. Damage caused by foreign objects is visible on the turbine and compressor vanes. The use of a compressor with damaged vanes (the turbine and the compressor vanes) leads to further damage, as the system is unbalanced. Examples of damage caused because of penetration of a foreign object into the compressor and turbine are presented in Fig. 2.8 and 2.9 respectively.

Causes:
- foreign objects left after reassembly or repair,
- ineffective air treatment system,
- loss of airtightness or damage to the engine air intake system,
- damage to the engine (damaged valve guides or heaters).

Damage resulting from the penetration of foreign objects into the exhaust or intake system result from negligence of the staff and may cause widespread damage, which can render any turbocharger irreparable. The mechanism of damage is of erosive character and depends on the size of a foreign object getting onto the rotor vanes. Even the smallest element carried with the air or exhaust gas flow has a strong influence on the vanes as a result of considerable gas flow energy.

![Fig. 2.8. Damage following penetration of a foreign object into the compressor](image1)

![Fig. 2.9. Damage following penetration of a foreign object into the turbine](image2)

2.4. Exceeding the admissible rotational speed of the turbocharger shaft or excessive pressure and temperature of the exhaust gas

Excessive increase of the turbocharger shaft speed in connection with an increase in the exhaust gas temperature can be observed in cases of inappropriate changes in the injection map (unauthorized software update). Excessive speed of the rotor may also be caused by unlimited charging pressure for higher engine speeds. This may be caused by blocking (carbon deposits) of the geometry in a setting for low engine speeds. In addition, the absence of charging pressure control favors uncontrolled increase in the rotor speeds. Such situation occurs in the case of damage to an electrical adjuster, the electrical system of a vehicle or loss of airtightness of the
turbocharger pneumatic control system. On the other hand, loss of airtightness of the intake system of a vehicle on the compression side cause the turbocharger to operate with smaller loads, which also leads to an increase in the turbocharger shaft speed. Increased pressure of the exhaust gas generates axial forces influencing the turbine rotor. This results in an increased wear of the thrust bearing and turbocharger seal rings, which may be caused by inappropriate engine operation.

On the other hand, overheating of the turbocharger is caused, above all, by deterioration of its cooling conditions, which is mainly caused by excessively quick engine stop after operation for a prolonged period, particularly under high loads. This type of overheating constitutes one of the most dangerous factors causing heavy damage of the compressor rendering it irreparable.

Overheating leads to the accumulation of carbon deposits (from burnt oil), which results from excessive temperature of the exhaust gas or excessively quick stop of the engine after operation. This may be prevented by letting the engine idle for a period of approx. 1 minute (the period of time necessary to cool the turbocharger). The transfer of high temperature from the turbine shroud to the mid part of the turbocharger causes oil burning and corrosion of the turbocharger bearings. The main damage is observed on the grooves and seal rings of the turbine shaft (Fig. 2.10) as well as on the bearing and the bearing pins.

![Fig. 2.10. Damage of the groove, the seal ring and the longitudinal bearing pins](image1)

Damage observed in the areas of the turbocharger seals lead to oil losses. Additionally, it leads to unintended outflows of air and exhaust gas (Fig. 2.11).

![Fig. 2.11. Sources of oil leakage and blowby of air and exhaust gas in the turbocharger](image2)

Burnt oil also causes congestion of the oil outflow in the midpart of the shroud (Fig. 2.12). This usually results in a destruction of the shaft and the midpart of the shroud in places where it cooperates with the seal ring and the bearings.
Fig. 2.12. The result of overheating – congested (by oil originating carbon deposits) turbocharger drain duct

Causes:
- congested or worn air filter element,
- excessively quick engine stop after operation,
- inefficient engine power system as connected with inappropriate operation of injectors,
- excessively long period of operation without oil change,
- blowby of air and exhaust gas,
- inefficient injectors,
- inappropriate assembly of the turbine,
- inefficient turbocharger lubrication system,
- inefficient oil return system (defects in engine crankshaft venting),
- excessive temperature of the exhaust gas (Fig. 2.13).

Fig. 2.13. Chipped and melted turbine rotor vanes because of high exhaust gas temperature

2.5. Corrosion

Corrosion of a pressure adjuster is a relatively rare damage in the area of the turbocharger (Fig. 2.14). Such a failure results in the absence of reaction of the turbocharger to the signal from the acceleration pedal. Such situations are often the case in Volkswagen Group vehicles.
3. Conclusions

The development of diesel engines always aims at decreasing the fuel consumption, increasing the power to capacity ratio, reducing the engine weight, ensuring reliability of operation under various operating conditions and increasing its durability. One of the long known methods of engine improvement is engine supercharging. Structural solutions applied in the supercharging equipment have greatly evolved. Despite the fact that a turbocharger is a high precision device, its mechanism is relatively simple, strong and effective. If properly operated, a turbocharger can operate reliably for years despite the fact that its operating conditions are very arduous. Turbochargers do not require any special maintenance. A periodical inspection of the turbocharger fitting in the engine is usually sufficient, which ensures appropriate operation of the turbocharger. The discussed operating problems of modern diesel engine supercharging systems widely investigated during the author’s research and expert practice are to some extent the consolidation of the author’s activity, which may contribute to further works on the improvement of diesel engine supercharging systems.

References

ENERGY STORAGE IN COMPRESSED AIR – SOLUTION SUPPORTING RENEWABLE ENERGY SOURCES

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Abstract

This article presents a brief description of a power system, the current national power system daily load, the use of wind power as a renewable energy source and its share in the national load. It also discusses the methods for storing energy, their characteristics and possible solutions. The power storage and generation solution proposed in the article is based on the collaboration between a gas turbine and an air storage system. The last chapter contains a visualization of a conceptual compressed air tank that could be used for offshore wind turbines.

Keywords: wind power plants, energy storage, CAES, compressed air, renewable energy sources, power system

1. Introduction

The provision of a sufficient amount of energy is one of the fundamental challenges that mostly highly developed countries are currently facing. The conducted analyses show that the increase in power demand is close to 1% per year, while in recent years in Poland it amounted to 2% [1]. That is why the continuous development of power generating capacity, modernization of the existing high power units, support of distributed power generation and renewable energy sources are of crucial importance.

Some alternative energy sources, though inexhaustible and commonly available, occur periodically and are usually not correlated with the period in which they are in demand [2]. Energy storage is closely linked to renewable energy sources, as it is essential in order to guarantee the stable operation of a power system.

Energy storage is a well known problem. Its history reaches as far as the beginning of generation, transmission, distribution and usage of electric energy. Many prototype systems are currently being launched that enable a large-scale transfer of energy generated by renewable sources to peak demand periods.

The demand for energy supplied to the NPS\(^1\) varies in time. It is related to short-term fluctuations of energy consumption each day and seasonal changes. When highly unpredictable sources are introduced into the system, electric energy consumption is even more variable. Wind and solar power plants are both examples of such sources [3]. The rate of increase in wind power

\(^1\) National Power System
plants installed capacity made it necessary for many countries to search for new technical solutions providing for the stabilization of the NPS. Energy storage in compressed air is one of such solutions.

2. Impact of wind turbines on the power system

The conversion of mechanical energy into electric energy is the last stage of energy conversion in most power plants. Electric energy generated in a power plant is transferred to the power system, where it is consumed by consumers. The system load varies in time. Certain peak and off-peak periods of power demand may be differentiated during the day, which is reflected in Figures 1 and 2.

For comparison purposes, characteristics of the Polish and Irish power systems were prepared, and the power generated by wind power plants was marked.

In January 2013 the installed capacity of wind power plants in Poland amounted to 2700 MW, while in December 2014 it is expected to reach 4040 MW. It clearly shows that the share of wind power plants in the national power system is increasing significantly (and will soon reach 20%). Unfortunately, as can be observed in Figures 1 and 4, the average power reached by wind turbines (installed capacity in Q3 2014 – 3680 MW) constitutes only 30% of installed capacity.

![Figure 1. Daily load of the National Power System as of 25 March 2014 with energy generated by wind power plants – Poland (compiled on the basis of PSE Operator data)](image)

The daily load is similar in both countries. Conventional baseload power plants are the main source covering the energy demand.
Generation of energy from wind sources is very unstable, variable and unreliable. The percentage likelihood that the load of wind energy sources in Poland will amount to between 0 and 20% (installed capacity) is 58.8%, between 0 and 30% (installed capacity) – 73.01%, and between 0 and 40% – 84.02%. Therefore, the wind power sources load is less than 40% for approximately 7360 h per year[4]. Wind power plants, as compared to non-renewable electric energy sources, have certain specificities. These are, among all [4]:

- frequent lack of correlation between the amount of generated power, dependant upon the current wind speed, and the end users demand for power (as may be observed in Figure 4, where maximum wind sources generation correlates with a non-peak period in the power system),
- sudden (often unpredictable) changes in power introduced into the power system, related with instantaneous changes of the wind speed (sudden maximum blasts and average wind speeds shown in Figure 3),
- poor predictability of wind power sources operation in the long term, as part of planning their collaboration with the power system.

*Figure 2. Daily load of the National Power System as of 25 March 2014 with energy generated by wind power plants (compiled on the basis of eirgrid.com data)*
Generally speaking, the greater the wind power sources installed capacity share, the greater the consequences for the power system [1].

Figure 3. Variability of wind speed in March, in the south regions of Poland (compiled on the basis of pogodaradlin.pl data)

Figure 4. Weekly load of the National Power System (compiled on the basis of PSE Operator data)
In order to mitigate the negative consequences of wind power plants operation for the NPS, systems operators apply remedies that can be divided into three basic groups [5]:

- regulatory activities involving the generation sources that are already present in the power system,
- application of an energy storage technology,
- introduction of demand control systems.

3. Energy storage

The electric energy storage technology that is currently used may be divided into indirect (with the conversion of electric energy into a different kind of energy, such as kinetic or chemical) and direct energy storage (in an electric or magnetic field).

Activities related with the utilization of energy storage technology enable:

- mitigation of the variability of wind power sources generation introduced into the power system in shorter periods of time,
- restriction of the use of conventional peak load sources when changes in wind power sources generation take place,
- transfer of electric energy generation from non-peak to peak hours.

A proper energy storage system should be characterised with [6]:

- high energy density,
- simple charging and discharging, and a large number of cycles,
- high energy efficiency of the cycles,
- possibility of a simple conversion of energy into a different form of energy,
- should reach the required economic efficiency and not pose a threat to the environment,
- required storage time and time of distribution to the user.
The basic parameters for the methods of energy storage, recommended by the American Electric Power Research Institute, for the support of wind power engineering integration into the power system are shown in Table 1.

<table>
<thead>
<tr>
<th>No.</th>
<th>Technology</th>
<th>Cycle efficiency %</th>
<th>Nominal power [MW]</th>
<th>Discharge time [h]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Pumped-storage Hydroelectricity</td>
<td>80</td>
<td>100-1000</td>
<td>&gt; 1 hour</td>
</tr>
<tr>
<td>2.</td>
<td>Compressed Air Energy Storage</td>
<td>60-75</td>
<td>0.1-1000</td>
<td>few hours</td>
</tr>
<tr>
<td>3.</td>
<td>Flywheel Energy Storage</td>
<td>90</td>
<td>0.1-10</td>
<td>0.1</td>
</tr>
<tr>
<td>4.</td>
<td>Conventional Batteries</td>
<td>60-80</td>
<td>0.1-10</td>
<td>0.1 ... &gt; 1</td>
</tr>
<tr>
<td>5.</td>
<td>Rechargeable electrochemical</td>
<td>70</td>
<td>0.1-20</td>
<td>&gt; 1</td>
</tr>
<tr>
<td>6.</td>
<td>Fuel cell</td>
<td>50</td>
<td>0.1-1</td>
<td>&gt; 1</td>
</tr>
</tbody>
</table>

A pumped-storage power station is a common method for energy storage (Germany is planning on building such systems in old coal mines). As the obtained capacity may affect the entire power system, it is a large scale technology. Compressed air systems are an alternative storage type, as they provide the same capacity but are more profitable.

Activities related with the end users demand control are an important aspect of energy storage. The expansion of demand control involves the introduction of financial incentives by the transmission and distribution systems operators in the form of special tariffs (hourly, dynamic tariffs).

4. Energy storage in compressed air

Old mine deposits, salt caverns, salt mine excavations, hard rocks excavations, areas following aquiferous layers [7] are used for energy storage in the form of compressed air, where air is compressed to the level of approximately 70-80 bars. In this type of power plants the generator is fuelled with liquid or gas fuel (a non-renewable source). There is no inlet air compressor, however, that under normal conditions consumes approx. 60% of mechanical energy of a conventional generator. This enables the reduction of CO₂ emissions and the increase in efficiency as compared to a conventional gas power plant [8].

In the analysed type of power stations, low-cost energy is used that is available outside of the power system peak load hours – during the nights and on weekends. When wind conditions are favourable, the excess energy is used for supplying the compressor that compresses air. It is shown in Figure 6. The consumed energy is used for compressing air in large tanks [3]. The collected energy is used during peak hours or when wind conditions are unfavourable and power generation by power plants does not meet the planned amount (e.g. an amount commissioned by a consumer). The control system starts a fuelled gas turbine and feeds compressed air to the turbine [8] (Figure 6).
Environmental friendliness of gas turbines, relatively low investment costs and high capacity reached by power units, simplicity and light weight of their structure, flexible movement, independence from water sources and possible automation led to the popularity of power stations with gas turbines, and their use is recommended in low capacity power plants, peak load power stations and in special circumstances. The use of this sort of solutions supports distributed generation. Listed below are possible technical solutions to the collaboration between gas turbines and compressed air tanks.

Figure 7 shows a “simple” air compression system; the heat generated during compression is lost. Total efficiency of the compression and expansion process in relation to electric energy reaches approx. 40% [9]. The use of natural gas in the combustion chamber enables the regulation of generated power and the increase in the power plant capacity.
Figure 8 shows a system utilizing exhaust heat due to the use of a recuperator. Such a system reaches the energy efficiency of a power plant of more than 50%. Reliability of such systems amounts to approximately 99%.

Figure 9 shows a CAES² system with the storage of exhaust heat generated during air compression, and its subsequent use for heating the air fed to the turbine. As a result, thermodynamic changes are similar to adiabatic changes. Such systems reach the efficiency of 60% without a combustion chamber.

Figure 9 CAES system with exhaust heat storage [compiled on the basis of 9, 10, 11]

² Compressed air energy storage
Figure 10 shows a system of the greatest efficiency, over 70%. This is a system containing both a combustion chamber and a recuperator. Such solutions are currently being designed and tested by those countries in which wind power engineering progress is greatest. The main objectives of the tests are air tanks and heat exchangers and storages that recover exhaust heat generated during compression.

A significant advantage to combined wind and CAES power plants is the increase in local energy security, because it is possible to start up a combined system without external supply and reach the desired capacity in a short period of time. In contrast to gas collection, its advantage is the possible autonomous operation of a gas turbine during expansion.

The start up of a pumped-storage power station takes anywhere between 1 and 15 minutes, while the start up of CAES system to full capacity is two or even three times shorter than an average start up time of a unit with a gas turbine, and it takes approximately 10 minutes. Many countries, especially the USA and Germany, being the European leader in wind power engineering, are involved in researching compressed air energy storage.

5. Air tanks

The maintenance of a constant pressure in an underground tank, e.g. through the hydrostatic pressure of water, would make it possible to increase the power plant efficiency to well over 70% due to the use of an isobarically isolated thermodynamic cycle. The concept of such a compressed air tank that could be used for offshore wind turbines is shown in Figures 11 and 12. The volume of this tank is approximately 30 m³.
Location-wise, both in Poland and in Europe there is a potential for onshore and offshore compressed air storage power plants. The construction of such power stations seems especially interesting in view of the current and planned major investments in wind power engineering.
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Summary

Measurements of nitrogen oxides concentrations in exhaust gases of marine engines under various cruising are difficult or impossible to perform due to their cost and technical difficulties. Such measurements are generally performed on engines under static load conditions, which are determined by appropriate tests, and this is the basis for computing the values of indicators and characteristics of NO\textsubscript{x} emissions. The values of the following parameters are determined: specific emissions, emissions intensity, cruise emissions, as well as the propeller, regulatory, and adjustment characteristics of – most frequently – specific emissions. These indicators and characteristics are of limited use for the determining of the NO\textsubscript{x} emissions level in the exhaust gas of engines belonging to a set of different vessels operating in a given sea area which e.g. due to stricter environmental requirements requires reliable information about the current level of pollution.

The authors have made a successful attempt at developing mathematical models in order to determine the resistance characteristics of the hulls of various ships and the power of their engines under different cruising conditions, and to estimate indicators and characteristics of NO\textsubscript{x} emissions on the basis of numerical simulation studies. The computer program developed allows for the examination of various vessel types, and their various sizes, cruising speed distributions, and changing external conditions (e.g. water density and depth, sea state etc.).

The present paper will presented the course of the model’s construction, the assumptions, and methods for determining the inter-relationships, and the results of numerical studies for a number of ships under various cruising conditions.

Key words: emissions of toxic nitrogen oxides, marine engines, emission models.

1. Introduction

The problem of air pollution in harbors and harbor approach areas is all the more important because of the fact that harbors are typically located near or in large cities, and their limited area causes a large concentration of vessels in a small area.

The currently effective MARPOL 73/78 convention on the protection of the marine environment is limited to determining the allowable unit value of emissions of certain toxic exhaust gas components (nitrogen oxides and sulfur oxides), and to conducting qualification research as a condition of obtaining the certificate confirming the meeting of the international legal requirements by marine engines, and vessels in which they are installed. These studies should be
conducted in accordance with the theoretical tests contained in the ISO 8178 standard, during the measurements taken at the manufacturing plant. The exhaust gases toxicity characteristics determined in this manner for an engine tested is then valid for the whole series (type, and even family) of similar engines. These studies are typically of a qualifying nature, and are not relevant to the actual operating conditions of the engines in the marine propulsion systems.

In reality operating conditions and emissions from a marine main propulsion engine are largely influenced by the so-called external conditions, of which, – due to their significant influence on the resistance of the hull – the following conditions may not be overlooked: navigation in shallow water, in channels with strong currents of water, and in storms, as well as an increase of draft (e.g. as a result of the increased cargo load, filling empty ballast tanks and the cargo space, or a decrease in the water density), as well as an increase in the number of appendages or the hull roughness as a consequence of the fouling and corrosion. It can thus be assumed that the vessel's actual operating conditions, regardless of the preset settings of the control and the propulsion systems, are characterized by a certain degree of dynamic changes in the propulsion system parameters, and hence emissions from propulsion engines. Engine operating conditions and emissions are also affected by changes of the vessel motion set by the crew (moving off, accelerating, braking). Therefore, operating conditions of ships indicate the necessity to develop a comprehensive method for determining the exhaust gases toxicity characteristics, with static and dynamic components, of the ship's main propulsion engine.

Research conducted currently throughout the world, concerning atmosphere pollution caused by emission of toxic compounds from ship engines, conceived both globally [1,2], regionally [3,4], and also locally, e.g. in the vicinity of large sea ports [5] is based on simplified input data [2, 6, 7]. The existing data bases of harmful compounds emission from exhausts of ships operating in various regions of the world [7] cannot be used, however, for the estimation of emission in meso- and microscale, e.g. the Baltic Sea or the Gulf of Gdansk, as they lead to a considerable underestimation of emission indexes, which is mainly due to insufficient detail of vessel movement characteristics [8].

There is thus a need need for methods that allow a much more precise determination of the vessel movement characteristics, as well as emission factors.

2. Propulsion characteristics of vessels with displacement hulls under static and dynamic operating conditions

The preliminary statistical calculations, and the analysis of the data emitted primarily by AIS shows that in the established research concept the crucial role is played by the general identification of the actual operating conditions of typical vessels, and on its basis the development of relative (general) characteristics of marine propulsion system, described by a set of functional interrelationships of selected propulsion system parameters [10]. It should be stressed that this type of functional relationships, which by their nature describe phenomena related to the operating process of the ship's propulsion, are empirical. It was assumed that the basic parameters of the vessel movement under consideration will concern the waters of the Baltic Sea, and that the parameters of the routes and vessels will be determined on the basis of pilot books and information issued by AIS. It was also assumed that a typical marine propulsion system consists of a low-speed reciprocating internal combustion engine and a fixed pitch screw propeller.

In order to determine the parameters of the ship's propulsion system, necessary in the calculation of toxic compounds emissions intensity, it is necessary to know the broad set of parametric mathematical models. Because of the issue under consideration, and the availability of the necessary data, it was concluded that changes in the intensity of toxic compounds emissions resultant from changes in the external operating conditions of the vessel's propulsion system are in each case the consequence of a useful demand required change, due to the changed resistance of
the hull, which means that they can be identified in an approximate manner based on the following set of predicted approximative equations and formulas:

- hull resistance characteristics, taking into account:
  a) influence of the fairway depth on the hull resistance;
  b) additional wind (air) and wave resistance during the storm;
  c) draft changes;
  d) hull roughness changes:
- power characteristics depending on the vessel speed;
- load characteristics;
- propeller characteristics of the main engine power;
- propulsion system parameters under nominal and transient load conditions during moving off, accelerating and braking – time \( t \) and distance \( S \);
- toxic compounds emissions intensity.

In the presented method, the first to be determined is the analytical dependence approximating the resistance and its components according to the characteristics of the hull and vessel motion under the current external operating conditions. In particular, it is necessary to have the following data:

- the relationship between the wave height, the speed of its motion, and the wind speed;
- the share of the wind and wave resistance components;
- relative resistance of the emersed part of the ship (superstructures);
- deep- and shallow-water wave characteristics;
- data of the parameters of hulls and propulsion systems of vessels, and the current data concerning motion and the changes of vessel parameters;
- current weather data.

The aforementioned relationships are also the output data and the algorithm used for the development of computerized procedures for the determination of emissions of toxic compounds by the ship's main propulsion engines in real (steady and transient) operating conditions of the propulsion system. They are also the basis for updating and enriching the database, and for the development of exhaust gases toxicity tests to be increasingly accurate and adequate to the reality.

In generally applied methods of determining emissions of toxic compounds by internal combustion engines, the basis are the toxicity tests developed on the basis of empirical studies of the actual operating conditions. Therefore, the accuracy of the emissions estimation depends on the accuracy with which the developed tests reflect the actual engine operating conditions, usually described in terms of load parameters.

The analysis conducted showed that for the purposes of the considered problems of toxic emissions into the atmosphere from marine main engines, the dynamic nature of the impact of the propulsion system operating conditions, as well as external navigation conditions may in many cases be omitted. This is because the duration of dynamic operating conditions is negligibly short in comparison to the duration of operation with the load close to being steady. Operating conditions of that kind occur e.g. during the acceleration of the vessel. Presented mathematical description and performed calculations for transient (dynamic) operating conditions of a marine propulsion system were used to estimate the impact of the changes in the hull motion parameters on the load parameters of the main engine, and the characteristics of the vessel exhaust gases toxicity under a variety of external conditions. The detailed description of an algorithm used for modeling ecological characteristics of marine main propulsion internal combustion engines can be found in [11].

The main characteristics describing the operating conditions of the main engine being a component of the marine propulsion system is the vessel's resistance characteristics, and the characteristics of the propellers. It is also known that these two types of characteristics are
determined experimentally during modeling tests. The degree of compatibility with the reality of the so-created characteristics is dependent on the accuracy of representation of the real object in the model, and the conditions in which the measurements and calculations were performed. Characteristics prepared in this manner should be included the technical and operational documentation of the vessel. In practice, however, such characteristics tend to be very difficult to obtain. Therefore, the most commonly used methods for computing the resistance characteristics are those developed on the basis of the available general data of a specific vessel [10].

Following the analysis of empirical approximate methods for calculating the resistance of vessels operating in deep (unlimited) water, methods [11] and [12] were selected for further consideration.

In the first of these relationships, the total resistance of the hull in deep water is described with the formula:

\[
R = 0.0132 \cdot 9.81 \cdot \rho \cdot L \cdot B \cdot T \cdot \delta (48 \cdot F_{nV}^2 - 29 \cdot F_{nV} + 5.9) \frac{\delta}{L^2 B^2 T^2} ,
\]

where:

- \( R \) [kN] – resistance;
- \( \rho \) [kg/m\(^3\)] – seawater density
- \( F_{nV} \) – Froude number relative to the volume of underwater hull \( F_{nV} = \frac{v}{\sqrt{g \sqrt{V}}} = \frac{v}{\sqrt{g \sqrt{L \cdot B \cdot T \cdot \delta}}} \)
- \( V \) [m\(^3\)] – underwater hull volume;
- \( L, B, T \) [m] – hull dimensions,
- \( \delta \) – hull’s block coefficient \( \delta = \frac{V}{L \cdot B \cdot T} \) the volume ratio of the underwater hull to the volume of the cuboid circumscribed around the underwater hull,
- \( g \) [m/s\(^2\)] – gravitational acceleration.

The application of the formula, however, was limited to the following ranges of the hull parameter changes. \( F_{nV} = 0.37 \div 0.77; \delta = 0.70 \div 0.86; L/B = 5.0 \div 9.5; B/T = 3.3 \div 13. \)

Because of the above limitations in the application of the method described above, the calculation of the vessel’s total resistance may also be performed using the method described in [12]:

\[
R = g \left\{ 0.17 \cdot \Omega \cdot v^{1.825} + 1.45 \left( 24 - \frac{L}{B} \right) \delta^5 \frac{D}{L^2} v^4 \right\} ,
\]

where:

- \( v \) [m/s] – vessel speed;
- \( B, L \) [m] – hull breadth and design waterline length
- \( D \) [m] – vessel draft;
- \( \Omega \) [m\(^2\)] – wetted area;
- \( g \) [m/s\(^2\)] – gravitational acceleration.

On the basis of equation (1) or (2), it is thus possible to determine the resistance characteristics of a particular vessel when cruising in deep (unlimited) water. Resistance characteristics determined in this manner allow to determine the effective towing capacity characteristics of the vessel, and the useful demand required by the propulsion engine from the general forms of equations.
\[ P_h = R \cdot v \]  
\[ P_e = \frac{R \cdot v}{\xi_o \cdot \eta_{LW} \cdot \eta_r} \]

where:

\( P_h \) [kW] – effective towing capacity,

\( R \) [kN] – total resistance of the vessel in deep-water conditions,

\( \xi_o \) [-] – propulsive efficiency \( \xi_o = \eta_p \cdot \xi_r \cdot \xi_k \);

\( \eta_p \) [-] – free propulsor efficiency (free propeller) – at undisturbed water inflow;

\( \xi_r \) [-] – rotative efficiency, taking into account changes in speed and direction of water inflow,

\( \xi_k \) [-] – hull efficiency, and to be more specific, the impact factor of the hull on propeller operation

\[ \xi_k = \frac{(1 - i)/(1 - w)} \]

\( \eta_{LW} \) [-] – shafting efficiency,

\( \eta_r \) [-] – reduction gear efficiency.

Assuming, based on reference literature [13, 14, 15], the average values of the coefficients for propulsion systems equipped with low-speed reciprocating internal combustion engines (\( \xi_o = 0.65 \), \( \eta_{LW} = 0.985 \), \( \xi_k = 1 \)), the formula to calculate the approximate useful demand required will be simplified to the form

\[ P_{e_w} \approx 1.56 \cdot R \cdot v, \]  

On the basis of data obtained from the AIS, relative basic (general) standard characteristics of marine propulsion system were determined, describing the static properties of marine propulsion system components (the ship's hull and propulsion engine) under normal operating conditions, which are shown in Fig. 1 [10].

\[ R^* = 0.3885 + 1.5721 \cdot v^2 + 2.1807 \cdot v^3 \]

\[ P^* = -0.1242 + 0.9771 \cdot v - 2.463 \cdot v^2 + 2.6267 \cdot v^3 \]

**Fig. 1. Standard averaged a) resistance characteristics of the displacement hull vessel, b) dependence of the main propulsion engine power on the individual cruising speed of the displacement hull vessel [10]**
Fig. 2 shows the characteristics of engine power (bundles of curves) in real operating conditions of the vessel in a limited water depth, and Fig. 3 – relative increase of the additional wave resistance $\Delta R_F$ depending on wind speed and storm condition operations.

![Graph](image1)

**Fig. 2. Averaged relative dependence of the main propulsion engine power of a vessel operating in waters of different depths [10]**

![Graph](image2)

**Fig. 3. Relative increase of the additional wave resistance depending on wind speed and storm condition operations [10]**

3. Simulation model of nitrogen oxide emissions in the exhaust gas from the main engines in real operating conditions of the vessel

Conditions of toxic compound formation in a marine main propulsion engine are determined by load parameter values (effective towing capacity $P_e$, vessel speed $v$), thermal state parameters (exhaust temperature $T_{sp}$, water temperature in the cooling system $T_w$, oil temperature in the lubrication system $T_o$), and the parameters of the air surrounding the engine (temperature $T_a$, pressure $p_a$, relative air humidity $\varphi$) $\bar{p}_j = (P_e, v, T_{sp}, T_w, T_o, T_a, p_a, \varphi)$. Load and thermal state parameters also depend on the external conditions described in the set $\bar{p}_j$ [16].

The following indicators characterizing the level of toxicity of marine engines were adopted on the basis of the existing legislation concerning the issues under consideration:
emissions of the toxic compound $e_{ZT}$ [kg/(kW-h)];
- emissions intensity $E_{ZT} = P_t \cdot e_{ZT}$ [kg/h];
- total emissions; $m_{ZT} = E_{ZT} \cdot t$ [kg];
- cruise emissions $b_{ZT} = \frac{m_{ZT}}{L_t}$ [kg/mile].

In the construction and testing of a stochastic model of traffic and emissions from vessels [11, 18-20] in a specific water area, it is necessary to know the number of vessels navigating in the analyzed area, their distribution in terms of their function, size, speed, power, and type of main engines, etc.

The routes analyzed for the purposes of statistical processing of marine vessel traffic streams were the approach fairways to the harbors of Gdynia and Gdańsk, and the fairway splitting into both these water lanes.

Based on the data obtained from the AIS, it was possible to create i.a. distributions of vessel incidence in each category (Fig. 4), and their speed (Fig. 5) [16].

![Fig. 4. Vessel incidence in particular categories for the Port of Gdańsk [16]](image-url)
The mathematical model of toxic compounds in exhaust gases of the main propulsion engines, which was developed by the authors, served as a basis for the determination of the power output characteristics in the function of velocity $P_e = f(v)$ of a selected tanker ($L = 60$ m, $B = 6$ m, $T = 4.1$ m, $v_{\text{max}} = 12$ kt), (Fig. 6).

Because the amount of toxic compounds emitted into the atmosphere depends primarily on the engine load, it was subsequently possible to determine, on the basis of the propeller characteristics of specific toxic compound emissions, the value of, among others, the intensity of nitrogen oxides emissions depending on the cruising speed of the tanker $E_{\text{NOx}} = f(v)$ (Fig. 7).

The obtained statistical data on vessel traffic [19, 22] also served as a basis for the development of typical characteristics of the vessel speed changes as a function of time for line service vessels. Based on the motion characteristics of passenger-car ferries, it was possible to determine the intensity of the emissions of toxic compounds in the exhaust of their main engines [17].

Fig. 6. Characteristics of power output as a function of velocity $P_e = f(v)$ of a selected tanker navigating to the Liquid Fuel Terminal ($L = 60$m, $B = 6$m, $T = 4.1$m, $v_{\text{max}} = 12$w) [16]

Fig. 7. Calculated dependence between the NOx emissions intensity and the velocity $E_{\text{NOx}} = f(v)$ of a selected tanker navigating to the Liquid Fuel Terminal ($L = 60$m, $B = 6$m, $T = 4.1$m, $v_{\text{max}} = 12$w)[17]
Fig. 8 shows the changes of the nitrogen oxides emissions intensity $E_{NOx}$ as a function of time during changes in speed (active braking) of ferries entering the port in Gdynia.

![Graph showing changes of nitrogen oxides emissions intensity $E_{NOx}$ as a function of time during changes in speed (active braking) of ferries entering the port in Gdynia.](image)

**Fig. 8. Changes of the nitrogen oxides emissions intensity $E_{NOx}$ as a function of time during changes in speed (active braking) of ferries entering the port of Gdynia [17]**

### 4. Conclusions

Modeling emissions of harmful compounds is a very important, and at the same time very complex issue. Many attempts are undertaken worldwide aimed at estimating the models of harmful compounds emissions in vessel exhausts. Unfortunately, due to the fact that the structure of the model depends not only on its purpose, but also, to a large extent, on the quantity and quality of input data, and that many studies are based on the insufficient amount and quality of data, frequently obtained from a variety of sources, and the need to use simplifications, all this significantly translates into the credibility of the model.

The possibility of obtaining data from the AIS, such as: the name, length, breadth, and type of ship, universal time related to the passing through a “gate”, course and velocity over the ground (COG and VTG), and the ship’s draft, permits the creation of innovative models describing vessel traffic in the area under consideration, and toxic compounds emissions from exhausts, both for a single vessel, and the whole area.

The presented a mathematical model for calculating the power of the vessel’s main propulsion system is universal and can subsequently be utilized, based on data obtained from the AIS, for the calculation of the toxic compounds emissions in the exhaust of seagoing vessels.

It should be added that apart from the problems with which automotive professionals struggle while modeling toxic emissions, in the case of vessels the parameters interfering with the accurate determination of the emissions levels of particular compounds may additionally include the technical condition of the engine, particularly its fuel system, and weather conditions (especially wind force and direction).

### References


PROGRAM FOR THE EMISSION SIMULATION OF TOXIC COMPOUNDS IN THE MAIN ENGINE EXHAUST OF VESSELS OPERATING IN A SPECIFIED WATER REGION

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Abstract

The current worldwide research on air pollution caused by the emission of harmful compounds from ships' engines is based on a simplified input. The existing databases of harmful compound emissions in the exhaust of vessels operating in various parts of the world cannot be used to estimate emissions in meso- and microscale, e.g. in the Baltic Sea or the Gulf of Gdansk, because this would lead to excessive generalization of emission factors, mainly due to the lack of the required detail concerning vessel motion characteristics.

The first issue tackled within the framework of the project, one of whose effects is to create a research tool in the form of a computer program which simulates toxic compound emissions in the exhaust of the main engines of vessels operating on a specific water body, was to create a vessel traffic database in a selected marine region (e.g. the Gulf of Gdansk). The data was obtained from the marine Automatic Identification System (AIS).

In addition to the parameters of vessel traffic in the analyzed region of the sea, the database created, is a collection of the available construction and operational data of those vessels, such as their size, displacement, power rating of the main propulsion engines, etc. Those data, after the appropriate processing, produced the so-called generalized resistance characteristics of the ships which became the basis for the determination of the maximum demand required by the screw propeller under given operating conditions. Information of the motive power (instantaneous) is necessary to determine the emission characteristics of harmful exhaust gases.

The extensive research resulted in the development of an original computer program, MEFSAS (Model of Emission From Ships At Sea), which allows to determine the power ratings of the screw propulsion engines (and on their basis the emissions characteristics) depending on the aforementioned variables of the operating conditions in the static and dynamic. states.

Key words: emissions of toxic compounds, marine engines, emission models, simulation

1. Introduction

The problem of air pollution in harbors and harbor approach areas is all the more important because of the fact that harbors are typically located near or in large cities, and their limited area causes a large concentration of vessels in a small area. Not without significance are also the broadly understood operating conditions. The latter include the manner in which engines are operated, the incidence and nature of steady and transient load states, as well as the external
conditions affecting the engine operation. Toxicity of the exhaust gases is also influenced by the types of fuel and lubricating oil used.

Factors determining the global emissions of substances contained in exhaust gases of marine engines are classified and described in detail in [1, 2].

The process of modeling the emission of toxic compounds in the exhaust of a marine engine is very complex, and it requires the input data that can be divided into four fundamental groups [9]:
- vessel parameters – length, breadth, draft, technical condition of the propulsion system, propulsion type (including the type and number of engines), type and number of screw propellers, etc.;
- vessel motion parameters – velocity and heading;
- external conditions – wind force and direction, air and water temperature, atmospheric pressure, humidity, sea state;
- number of vessels, taking into account their categories.

Models of emissions from land transport, which are made in Europe, cannot be used to assess emissions from ships, due to the difference of both hydro-meteorological conditions and the specifics of the ship operation.

The model of toxic emissions in the exhaust gases of marine engines, STEAM (Ship Traffic Emission Assessment Model), which is presented in [3], is based on data transmitted by AIS, which is a basis of the calculations for determining emissions factors related to the emissions of harmful compounds in exhaust gases. In this model, however, simplifying assumptions were not avoided either, which may result in the fact that the determined emissions factors do not reflect the real emissions value.

2. Theoretical foundations for determining the ship's resistance and the power of the main propulsion system

In order to give a vessel a certain velocity, the main propulsion engine has to provide adequate power to the propeller, which is necessary to overcome the resistance to the motion of the ship, and the energy loss of the propeller, shafting, gears, and couplings. A general motion equation of a ship may be presented as follows [4,5]:

\[-(m + m_{11}) \frac{dv}{dt} - R - \Delta T + T = 0\]  (1)

where:

- m – weight of the ship, propellers and rudder,
- m_{11} - weight of accompanying water,
- R – ship's total resistance,
- T – propeller thrust
- \Delta T – thrust deduction.

The total resistance R of the vessel depends on the size of the vessel, its velocity, and the shape of its hull. The resistance is also affected by external factors such as waves of the sea, hull fouling, draft variations, etc.

It can therefore be concluded that the value of the maximum demand required depends primarily on the dimensions of the ship and its instantaneous velocity.

The value of the propeller thrust depends on the diameter of the propeller, its geometrical shape, speed, and the velocity of the vessel. The propeller thrust created must equalize the total
resistance of the ship \(R\) and the thrust deduction \(\Delta T\), acting on the hull in the direction opposite to its motion.

For steady movement \((dv/dt=0)\) equation (1) becomes:

\[
R + \Delta T = T
\]

The total resistance of the ship is the sum of the resistance components

\[
R = \sum R_{(i)} = R_F + R_{VP} + R_W + R_D = S \frac{\rho \cdot v^2}{2} (c_F + c_{VP} + c_W) + R_D = k \cdot c_T \frac{\rho \cdot v^2}{2}
\]

where:
- \(R_F\) – frictional resistance,
- \(R_{VP}\) – viscous pressure resistance (form resistance)
- \(R_W\) – wave resistance,
- \(R_D\) – additional resistance,
- \(S\) – wetted surface \([\text{m}^2]\),
- \(\rho\) – water density \([\text{kg} \cdot \text{m}^{-3}]\),
- \(v\) – flow velocity around the hull \([\text{m} \cdot \text{s}^{-1}]\),
- \(c_F\) – frictional resistance coefficient,
- \(c_{VP}\) – viscous pressure resistance (form resistance) coefficient,
- \(c_W\) – wave resistance coefficient,
- \(k\) – additional resistance coefficient (assumed \(k = 1.1 \div 1.2\)),
- \(c_T\) – total resistance coefficient.

Frictional resistance is related to the tangential stresses that are induced on the wetted surface of the hull due to the viscosity. Frictional resistance coefficients \(c_F\) thus depends primarily on the Reynolds number, expressed as

\[
Rn = \frac{vL}{v}
\]

where:
- \(V\) – velocity of the vessel,
- \(L\) – length of the vessel,
- \(v\) – kinematic viscosity coefficient of water.

Wave resistance is related to the wave pattern generated by a moving ship on the water surface without viscosity (ideal fluid), i.e. on the phenomena whose existence is conditioned by gravity. Therefore, wave resistance coefficient \(c_W\) depends primarily on the Froude number.

\[
Fn = \frac{v}{\sqrt{gL}}
\]

Viscous pressure resistance is related to the effects of viscosity on the pressure distribution and hence to form of the the waves pattern. Viscous pressure resistance coefficient thus depends both on the Reynolds and Froude numbers.

Additional; resistance \(R_D\) is primarily composed of appendage resistance \(R_{AP}\) and air resistance \(R_{AA}\).

Appendages affecting the total resistance of the ship are such components as bilge keels, shaft brackets, shafts, propeller screws, rudders and shaft bossings on the hull.

Air resistance around the emersed part of the ship is of viscous character. The components of air resistance are frictional resistance and viscous pressure resistance. Air resistance results from both the relative motion of the ship in still air, and from the absolute motion of the air, i.e. the wind. Air resistance is strongly dependent on the size and shape of the the emersed part of the ship.
(especially its superstructures), and the volume and direction of the relative speed of the air. The formula for air resistance takes the form:

\[ R_{AA} = c_{AA} \frac{\rho A}{2} V_{WR}^2 A_T \]  

(6)

where:
- \( \rho \) – air density,
- \( V_{WR} \) – relative air velocity,
- \( A_T \) – midship cross section area of the emersed part of the ship,
- \( c_{AA} \) – air resistance coefficient,

The total resistance of the ship \( R \) is the sum of the following resistance components: frictional \( R_F \), form \( R_{FV} \), wave \( R_W \) and additional \( R_D \)

\[ R = R_F + R_{VP} + R_W + R_D = \Psi (\nu, L, \rho, \nu, M, A, G, O) \]  

(7)

This force can be presented as a function of \( \Psi \): instantaneous speed of the ship \( \nu \), hull length \( L \), water density \( \rho \), kinematic viscosity coefficient of water \( \nu \), vector \( M \), characterizing the inertia of the ship, vector \( A \), containing information about the variable motion resistances of the ship associated with a given water region (water depth, width of the fairway (channels), etc.), vector \( G \), describing the ambient conditions (e.g. ambient pressure and temperature), and vector \( O \), describing the navigational conditions (wind force and direction, wave height and length, etc.)[2].

Since the resistance coefficients \( c_F, c_{VP} = f (Rn) \) and \( c_W, c_{VP} = f (Fn) \), it was assumed for modeling purposes that the values necessary to perform the calculations for a given category of a tramp vessel are the generated values of the vessel’s length \( L \) and its instantaneous speed \( \nu \).

The total hull resistance \( \tilde{R} \) is presented by the equation:

\[ \tilde{R} = \tilde{S} \frac{\rho \bar{v}^2}{2} (c_F + c_W) + \tilde{R}_{AA} \]  

(8)

where:
- \( \tilde{S} = \frac{\sum_{i=1}^{n} S_i}{n} \) for \( i=1,2, ..., n \) – average wetted area, computed using \( n \) relationships,
- \( \bar{v} \) – vessel speed generated on the basis of statistical data,
- \( \tilde{R}_{AA} \) – air resistance generated on the basis of statistical data,

Frictional resistance coefficient was computed using the ITTC formula [5].

\[ c_{F_{ITTC}} = \frac{0.075}{(logRn-2)^2} \]  

(9)

Since the outer surface of the hull (even with high-quality coatings), cannot be considered hydrodynamically smooth, the calculations took into consideration also the hull roughness expressed as an additive hull roughness coefficient \( \Delta c_F \) using the formula [5,6]:

\[ \Delta c_F = \left[ 105 \left( \frac{k_s}{L} \right)^{1/3} - 0.64 \right] \cdot 10^{-3} \]  

(10)

The coefficient of friction was calculated from

\[ c_F = c_{F_{ITTC}} + \Delta c_F \]  

(11)
Wave resistance cannot be determined analytically. Theoretical methods of determining the ship's wave resistance are based on the following assumptions [5,7]:

– water is considered to be non-viscous liquid,
– flow around the hull is considered to be potential, non-cyclic.

The above assumptions lead to a nonlinear boundary value problem for Laplace's equation in three dimensions. This problem can hardly be solved by the adoption of further simplification consisting in the linearization of the boundary conditions.

For the calculated vessel the towing capacity will be

$$\bar{p}_t = \bar{R} \cdot \bar{v} \quad (12)$$

Instantaneous effective power of the propulsion engine $\bar{p}_e$ will then have the value expressed by the equation:

$$\bar{p}_e = \frac{\bar{p}_t}{\eta_0 \eta_s \eta_g \eta_B} \quad (13)$$

For the calculation of coefficients, efficiencies were adopted on the basis of the literature [4,5].

3. Program for the emission simulation of toxic compounds in the main engine exhaust of vessels operating in a specified water region

In order to perform simulations of vessel traffic in the analyzed area, and the estimation of emissions of harmful substances in the main engine exhaust gases at specified intervals of time, a computer calculation program was developed, known as MEFSAS (Model of Emission From Ships At Sea) [2].

Fig. 1 shows a sample window of a model input parameters (hydrometeorological conditions), and Fig. 2 a window of toxic emissions in the exhaust on particular days of the week for the first category of vessels (bulk carriers) [2].

**Fig.1. Sample window of input parameters for a mathematical model used to estimate the emissions of harmful substances in the exhaust gases of the marine main engines – MEFSAS (hydrometeorological conditions) [2]**
Apart from the presentation of simulation results in a tabular form, the MEFSAS program may be used to visualize the results using bar graphs, which significantly simplifies their analysis.

The basic options for presenting the simulation results include:

- Bar graphs broken down by the vessel type (Fig. 2).
- Bar graphs of toxic emissions for different vessel types as a function of time, presented separately for CO, HC i NOx (Fig. 3).

All the simulation results obtained using the MEFSAS program can be saved to a text file, which can be subjected to statistical analysis (the scope of which was not defined at the time of writing the program) using practically any tool, such as: Statistica, Excel, etc. (rys.4).
The following data is saved to the file:
- day of the vessel's entry into the water region,
- minute of the event (in relation to day),
- tramp or liner vessel information,
- vessel type
- wind direction,
- wind speed,
- air temperature,
- water temperature,
- atmospheric pressure
- air humidity
- vessel's length,
- vessel's breadth,
- vessel's draft,
- immersed hull volume
- vessel's height
- vessel's velocity
- vessel's age,
- main propulsion type (two- or four-stroke engine)
- maximum demand required of the main engines
- wind speed relative to the vessel's hull,
- wind direction relative to the vessel's hull,
- specific NOx emissions for a given engine type,
- specific CO emissions for a given engine type,
- specific HC emissions for a given engine type,
- NOx emissions for a given engine type,
- CO emissions for a given engine type,
- HC emissions for a given engine type,

Figure 5 shows an algorithm which is the basis for calculations performed by the MEFSAS program.
Fig. 5 Algorithm being the basis for calculations performed by the MEFSAS program.

START

- Selecting the starting point of the program: day, hour, month – \( i_0 \)
- Entering the simulation time: \( i_k \)
- Generating the fairway for a vessel
- Generating the vessel type: \( K_v \)
- Generating the vessel length: \( L_v \)
- Generating the vessel speed: \( V_v \)

\[ \tau = \tau_0 + \tau_1 + \ldots + \tau_k \]

- If \( \eta \leq \tau \) then No
- \( i = i + 1 \)

- Calculation of the vessel's principal dimensions: \( B_v, H_v, l_v \)
- Calculation of the vessel's displacement: \( V_v \)
- Loading traffic tables for liner vessels
- Calculation of the total hull resistance: \( R_{Hv} = 2R_{Hv} \)
- Calculation of the towing capacity: \( P_{tw} \)
- Calculation of the instantaneous effective power: \( P_{ei} \)

Generating atmospheric conditions for the present month:
- wind direction;
- wind force;
- air temperature;
- water temperature.

- Calculation of the vessel motion time in transit: \( s \)
- Calculation of the wave resistance increase: \( R_{SWV} \)
- Calculation of the wind resistance increase: \( R_{SWW} \)
- Calculation of the total resistance increase caused by the impact of hydro-meteorological conditions: \( R_{SWH} = R_{SWV} + R_{SWW} + R_{SWSW} \)
- Calculation of the corrected effective power: \( P_{e} \)
- Generating the vessel age
- Generating the main propulsion engine type
- Calculation of the \( k \)-th compound emission intensity: \( E_k \)
- Calculation of the \( k \)-th toxic compound emission intensity

\[ i = \eta \]

- No
- Yes

STOP
4. Conclusions

Continuous development of maritime transport, with the ever increasing demands on environmental protection, high costs, and problems associated with the measurement of emissions of harmful compounds in the exhaust of ships in transit, as well as the lack of sufficiently accurate methods for the determination of indirect emissions, were the main reason for undertaking research studies on modeling the emissions processes of marine diesel main propulsion engines under the marine operating conditions [2].

The current worldwide research on atmospheric air pollution caused by the emission of harmful compounds from ships' engines, based on a simplified input data cannot be used to estimate emissions within e.g. the Baltic Sea or the Gulf of Gdansk, since they lead to a significant underestimation of emissions, mainly because of insufficient detail of the marine traffic characteristics. Besides, the known models of the harmful compound emissions in exhaust gases of marine engines, which are used primarily to support local and regional model studies concerning air quality, are deterministic models with varying degrees of accuracy which depends on the resolution of the spatial allocation of emissions at a particular location and time. Moreover, the accuracy of a given model depends to a large extent on the amount and quality of input data, determined by financial resources earmarked for the creation, implementation, and calibration of the model [2].

The mathematical model of toxic emissions proposed in [2], which is based on stochastic processes using Monte Carlo methods, allows for a rapid analysis of marine traffic in a particular area, and for calculation with considerable accuracy of the emission intensity of each harmful compound, and their weight in relation to both a single vessel, and to vessels remaining in the area for a specified period of time. Furthermore, the model developed is the first fully predictive model, and the accompanying computer simulation program allows the analysis of marine traffic, and emission intensity at the selected point of time, taking into account hydro-meteorological conditions corresponding to that point.

The mathematical model was the basis for the development of a computer program that allows to solve the model's equations. The results generated by the program can be saved to a file compatible with Microsoft Excel, which allows for their analysis independent of the software used by the model. Additionally, it is possible to visualize the simulation results in the form of easily readable charts showing: the number of vessels in the analyzed water area during the day, with the option of splitting the vessels by their type, the emissions of various toxic compounds by the day and vessel type, as well as the total emissions of individual compounds from all vessels on each of the days of the simulation. Another feature of the program is the is the possibility to visualize the motion of the simulated vessels in the analyzed water region on the basis of the simulation results. This feature is based on the animation showing the vessels plotted on the chart of the relevant area.

The simulation program developed is open to any modifications related to the specifics of the analyzed issue, and, what is more, because of its versatility, it may be implemented into any area of marine operations very quickly after the introduction of a new set of input data.

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Abstract

This paper presents the thermodynamic analysis of a combined cycle gas turbine units without and with the carbon capture installation (CCS). The characteristics of the combined cycle units and the carbon capture technology were discussed. The operation methods of the oxy - combustion and the post - combustion technology was presented. The three units: combined cycle gas turbine without CCS, the combined cycle gas turbine with oxy - combustion and the combined cycle gas turbine with post - combustion were analyzed. The emission of carbon dioxide from above mentioned units also was compared.

1. Introduction

Combined cycle gas turbines (CCGT) are a combination of a gas turbine cycle with a steam turbine cycle by a heat recovery steam generator (HRSG), what makes them one of the most efficient technologies of electricity generation from fossil fuels. These units are also one of the fastest developing solutions, with efficiency already reaching above 60%. CCGT units are also distinguished by a number of advantages, such as short construction time, low investment cost, high reliability and flexibility of operation, favorable ecological characteristics – with 60% efficiency CO₂ emission is about 330 kgCO₂/MWh. This value is around 2.5-times lower than CO₂ emission of modern coal-fired power plants, equal to over 800 kgCO₂/MWh. Currently, CCGT are not popular in Poland due to a high natural gas prices in Poland [1,2].

The energy sector is facing the new challenges to reduce the CO₂ emission level. Recently developed Carbon Capture and Storage technologies (CCS) are expected to allow near zero-emission production of electricity from fossil fuels. The aim of CCS technologies is to separate carbon dioxide followed by its transportation and storage. There are 3 basic technologies of CO₂ separation:

- post-combustion
- pre-combustion
oxy-combustion

The post-combustion technology is based on CO₂ separation from the flue gas, so it does not interfere with the combustion process and the basic structure of the power unit. That makes the post-combustion technology easy to implement, and even possible to install in existing power units, so-called “capture-ready”. Separation of CO₂ may be realized by various methods, but now the optimal solution is chemical absorption, which confirms the literature review, e.g. [3,4,5,6]. This method is also applied in one of the analyzed units, described in pt. 2.3.

Chemical absorption is based on the absorption of the gas molecules by the liquid sorbent. The process is conducted in the absorber-desorber system. A high level of recovery and high carbon dioxide purity proves the effectiveness of this method. Realization of the CO₂ capture process based on chemical absorption requires to provide a suitable amount of heat energy to the desorber column for regeneration of the sorbent. To provide required heat, a steam extraction is performed in the low pressure steam turbine. This procedure lowers the steam turbine power, and hence, decreases the efficiency of the CCGT unit. In order to reduce this power drop new sorbents are searched, which would be characterized by low energy consumption.

The oxy-combustion is an alternative CO₂ separation technology, which concept is based on the combustion of fuel in an oxidant with increased oxygen content. The elimination of nitrogen from the combustion process makes the flue gas consist mainly of carbon dioxide and water vapor, allowing for the CO₂ separation with a relatively low energy demand. In spite of the simplified CO₂ separation in relation to other CCS technologies, in this case the oxygen separation process is associated with a significant demand for electric energy. Due to the requirement of a high performance and sufficient oxygen purity, currently the use of cryogenic air separation unit is considered. In the worldwide available literature there are few positions about oxy-combustion technology in CCGT plants, and those are often concerning customized units, e.g. [7,8].

2. The structures of analyzed units

2.1. The combined cycle gas turbine unit without carbon capture installation (CCGT)

The structure of the combined cycle gas turbine without carbon capture installation is presented in Fig. 1. This unit is further identified as the CCGT case and it is a reference unit for the compared systems presented in point 2.2 and 2.3, which are extended with the installations associated with the application of selected CCS technology. The class G gas turbine for the analyzed combined cycle power plants is proposed. The gas turbine has a net electrical power equal to 200 MW, a combustion chamber outlet temperature is \( t_{3a} = 1500^\circ\text{C} \) and the pressure ratio is \( \beta_K = 23 \). The open-air film cooling in the gas turbine is used. The gas turbine outlet flue gas is sent to the heat recovery steam generator producing the steam to supply the steam part. The triple-pressure heat recovery steam generator with the steam reheating is decided to use. The most important parameters for the gas turbine and the steam cycle of the CCGT unit in are presented in point 2.4.
2.2. The combined cycle gas turbine unit with oxy-combustion technology (OXY)

The structure of the CCGT unit with oxy-combustion is presented in Fig. 2. This unit is further identified as the OXY case. In the unit a higher pressure ratio than in the air-combustion units is applied, equal to $\beta_K = 50$. This high pressure ratio has for example the turbine Rolls Royce Trent 1000, the unit with the same pressure ratio value is also analyzed in [10]. The main assumptions are listed in pt. 2.4. In the oxy-combustion it is necessary to introduce the flue gas recirculation, so to the compressor is directed the flue gas from the cooler (FC) outlet. After compression its mixed with the oxidant and sent to the combustion chamber. The possible influence of the steam part on the gas turbine work parameters is eliminated by maintaining the constant temperature of the recirculated flue gas by the FC. This temperature is set so as to avoid the moisture condensation ($t_{5,1a} = 90^\circ\text{C}$). The oxidant consists of 99.5% $\text{O}_2$ and 0.5% $\text{N}_2$. Streams of the mixed recirculated gas and oxidant are set so as to provide the constant $\text{O}_2$ content equal to 2% in the flue gas at the combustion chamber outlet.
The not recirculated part of flue gas is cooled in the dryer (FD) to the temperature of $t_{2c} = 20^\circ$C, resulting in condensation of almost all the moisture content. Subsequently the flue gas (consisting about 90% of CO$_2$) is compressed to the pressure of 15 MPa in the CC installation and sent to the place of storage. The CO$_2$ capture effectiveness is assumed at the level of 98%, remaining 2% is emitted to the atmosphere. Design and working parameters of the cryogenic air separation unit and CC installation are not analyzed. There have been made assumptions of the unit energy consumption, for the oxygen production in ASU equal to $E_{N(ASU)} = 0.2$ kWh/kgO$_2$ [10], and for the carbon dioxide compression in CC installation equal to $E_{N(CC)} = 0.1$ kWh/kgCO$_2$ [1]. The own needs of the ASU installation $N_{ASU}$, and of the CC installation $N_{CC}$ are equal to:

$$N_{ASU} = \dot{m}_{2o} \cdot E_{N(ASU)}$$

$$N_{CC} = \dot{m}_{2c} \cdot E_{N(CC)}$$

Where $\dot{m}_{2o}$ and $\dot{m}_{2c}$ are the mass flows of the oxidant from ASU and compressed CO$_2$, respectively.

### 2.3. The combined cycle gas turbine unit with post-combustion technology (ABS)

The integration of the combined cycle power plant with the carbon capture installation in post-combustion technology is presented in Figure 3. This unit is further identified as the ABS case. The optimization process and analysis of this structure is presented in [11]. The main assumptions for the ABS unit are listed in pt. 2.4.
The flue gas from the HRSG is directed to the absorber column, where it is cooled to the temperature of 40 °C. The CO₂ recovery rate is $R = 90\%$, which means that 10% of the CO₂ is emitted into the atmosphere. MEA (monoethanolamine) is used as the sorbent, whose energy consumption equal to 4 MJ/kgCO₂ is assumed. The chemical absorption process takes place in the absorber column. The MEA and CO₂ mixture is directed to the stripper column. It is necessary to provide a suitable amount of the heat energy for the sorbent regeneration. For this purpose the steam extraction is performed. The medium in the stripper heat exchanger is heated to the temperature of 125 °C. The pressure of the steam extraction used for regeneration of the MEA equals $p_{11s} = 0.287$ MPa.

The steam extraction for the sorbent regeneration causes a significant decrease of the steam turbine power, and in consequence decrease in the unit’s power and efficiency. The CO₂ compression in CC installation causes further decrease in the efficiency and power of the ABS unit. Similarly to the OXY case, the energy consumption of the CO₂ compression equal to $E_{N(CCS)} = 0.1$ kWh/kgCO₂ is assumed. In this case the auxiliary power of the CC installation $N_{CC}$ is equal to:

$$N_{CC} = \dot{m}_{sc} \cdot E_{N(CC)}$$

(3)

Where $\dot{m}_{sc}$ is the mass flow of the separated carbon dioxide.
2.4. Operation parameters of the units

The main assumptions for the gas turbine part and steam part of the analyzed cases CCGT, OXY and ABS are presented in Tab. 1. To the units air is supplied at a temperature of 15 °C, pressure of 101,325 kPa, and a relative humidity of $\varphi = 60\%$. The combustion chambers are powered by the natural gas with the lower heating value $LHV = 50,18$ MJ/kg and volumetric composition: 98,21% CH$_4$, 1,27% N$_2$ and 0,52% CO$_2$. The models of the analyzed cases have been made in a GateCycle™ software [12]. Selection of the steam parameters was preceded by the optimization by means of genetic algorithm. Optimization process and detailed results are presented in other author’s papers, for the CCGT and ABS cases in [11], and for OXY case in [9].

*Table 1. The parameters of the gas turbine and steam part of the CCGT, OXY and ABS cases*

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>CCGT</th>
<th>OXY</th>
<th>ABS</th>
<th>unit</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>GAS TURBINE</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas turbine gross electric power</td>
<td>$N_{elGT}$</td>
<td>200</td>
<td></td>
<td></td>
<td>MW</td>
</tr>
<tr>
<td>Combustion chamber outlet temperature</td>
<td>$t_{3a}$</td>
<td>1500</td>
<td></td>
<td></td>
<td>°C</td>
</tr>
<tr>
<td>Compressor pressure ratio</td>
<td>$\beta$</td>
<td>23</td>
<td>50</td>
<td>23</td>
<td>-</td>
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<tr>
<td>Turbine cooling air flow</td>
<td>$m_{2,5a}$</td>
<td>0,20*m$_{1a}$</td>
<td></td>
<td></td>
<td>kg/s</td>
</tr>
<tr>
<td>Cooling air flow ratio of the expander stages:</td>
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<td></td>
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<td></td>
</tr>
<tr>
<td>I stage:</td>
<td>$\delta_I$</td>
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<td>II stage:</td>
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<td>III stage:</td>
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<td></td>
<td>-</td>
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<td>$\delta_{IV}$</td>
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<tr>
<td>Compressors isentropic efficiency</td>
<td>$\eta_{IC}$</td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{ICO}$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
<tr>
<td>Expander isentropic efficiency</td>
<td>$\eta_{IT}$</td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
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<td>Compressors and expander mechanical efficiency</td>
<td>$\eta_{mC}$</td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>$\eta_{mT}$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>-</td>
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<td>Mechanical efficiency of the generator</td>
<td>$\eta_{g}$</td>
<td></td>
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<td></td>
<td>-</td>
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<tr>
<td>Efficiency of the combustor chamber</td>
<td>$\eta_{CCH}$</td>
<td></td>
<td></td>
<td></td>
<td>-</td>
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<tr>
<td>Compressor inlet pressure loss rate</td>
<td>$\zeta_{1a}$</td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
<tr>
<td>Combustion chamber pressure loss rate</td>
<td>$\zeta_{2a-3a}$</td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
<tr>
<td>Lower heating value</td>
<td>$LHV$</td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
<tr>
<td>Expander outlet pressure</td>
<td>$p_{4a}$</td>
<td>103</td>
<td></td>
<td></td>
<td>kPa</td>
</tr>
<tr>
<td>Gas turbine and steam part own needs ratio</td>
<td>$\delta_{el}$</td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
<tr>
<td><strong>STEAM CYCLE</strong></td>
<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>High-pressure steam turbine inlet temperature</td>
<td>$t_{3s(h)}$</td>
<td>560</td>
<td>600</td>
<td>560</td>
<td>°C</td>
</tr>
<tr>
<td>High-pressure steam turbine inlet pressure</td>
<td>$p_{3s(h)}$</td>
<td></td>
<td></td>
<td></td>
<td>kPa</td>
</tr>
<tr>
<td>Intermediate-pressure steam turbine inlet temperature</td>
<td>$t_{3s(i)}$</td>
<td></td>
<td></td>
<td></td>
<td>°C</td>
</tr>
<tr>
<td>Intermediate -pressure steam turbine inlet pressure</td>
<td>$p_{3s(i)}$</td>
<td>3600</td>
<td>3360</td>
<td>4200</td>
<td>kPa</td>
</tr>
<tr>
<td>Condenser inlet pressure</td>
<td>$p_{CND}$</td>
<td></td>
<td></td>
<td></td>
<td>kPa</td>
</tr>
<tr>
<td>Steam turbine isentropic efficiency</td>
<td>$\eta_{IST}$</td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
<tr>
<td>Steam turbine and generator mechanical efficiency</td>
<td>$\eta_{mST}$</td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{g}$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>-</td>
</tr>
</tbody>
</table>
3. Thermodynamic efficiency evaluation

3.1. Evaluation methodology

Gas turbine gross electric power $N_{elGT}$ is determined by the relationship:

$$N_{elGT} = \left( N_{iT} \cdot \eta_{iT} - \frac{N_{iC} + N_{iCO}}{\eta_{mC}} \right) \cdot \eta_{G} \quad (4)$$

where: $N_{iT}$ – expander internal power, $\eta_{iT}$ – expander mechanical efficiency, $N_{iC}$ – compressor internal power, $N_{iCO}$ - oxidant compressor internal power (appears only in the OXY case, in the remaining cases $N_{iCO}$ is 0), $\eta_{mC}$ – compressors mechanical efficiency, $\eta_{G}$ – generator efficiency

Effectiveness of the CCGT unit is evaluated by the efficiency of electric energy production. Gross electric efficiency $\eta_{el, gross}$ is defined by the relation:

$$\eta_{el, gross} = \frac{N_{el,gross}}{\dot{m}_{f} \cdot LHV} = \frac{N_{elST} + N_{elGT}}{\dot{m}_{f} \cdot LHV} \quad (5)$$

gdzie: $N_{el,gross}$ – electric power of the CCGT unit, $N_{elST}$ – steam turbine electric power, $\dot{m}_{f}$ - fuel mass flow, $LHV$ – lower heating value of the fuel.

Electric efficiency of the gas turbine $\eta_{elGT}$ and steam part $\eta_{elST}$ are given by:

$$\eta_{elGT} = \frac{N_{elGT}}{\dot{m}_{f} \cdot LHV} \quad (6)$$

$$\eta_{elST} = \frac{N_{elST}}{\dot{Q}_{4a}} \quad (7)$$

Where $\dot{Q}_{4a}$ is the heat flow at the HRSG inlet.

By using the ratio of heat flow at the expander outlet $\dot{Q}_{4a}$ to the gas turbine electric power, indicated as $\alpha$, equation (5) can be written as:

$$\eta_{el, gross} = \eta_{elGT} \cdot (1 + \alpha \cdot \eta_{elST}) \quad (8)$$

$$\alpha = \frac{\dot{Q}_{4a}}{N_{elGT}} \quad (9)$$

The net electric efficiency of the combined cycle unit is defined with analogy to (5), taking into account the own needs of individual installations within the unit:

$$\eta_{el} = \frac{N_{elGT} + N_{elST} - \sum \Delta N_{i}}{\dot{m}_{f} \cdot LHV} \quad (10)$$

$$\sum \Delta N_{i} = \Delta N_{el} + \Delta N_{ASU} + \Delta N_{CC} \quad (11)$$

The applied additional installations vary depending on the analyzed case. In equation (11) all occurring installations are listed, but if the analyzed case is not equipped with selected installation, then its own needs are equal to zero. The total own needs rate of the unit $\delta$ is equal to:

$$\delta = \sum \delta_{i} = \delta_{el} + \delta_{ASU} + \delta_{CC} = \frac{\Delta N_{el} + \Delta N_{ASU} + \Delta N_{CC}}{N_{el, gross}} \quad (12)$$
where: \( \delta_{\text{el}} \) – gas turbine and steam part own needs ratio, \( \delta_{\text{ASU}} \) – air separation unit own needs ratio, \( \delta_{\text{CC}} \) – carbon dioxide compression installation own needs ratio.

### 3.2. Results

Characteristic parameters obtained for the analyzed combined cycle gas turbine units are presented in Tab. 2.

Table 2. Characteristic parameters of the CCGT, OXY and ABS cases.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>CCGT</th>
<th>OXY</th>
<th>ABS</th>
</tr>
</thead>
<tbody>
<tr>
<td>( N_{\text{IT}}, ) MW</td>
<td>428.1</td>
<td>520.6</td>
<td>428.1</td>
</tr>
<tr>
<td>( N_{\text{IC}}, ) MW</td>
<td>219.5</td>
<td>283.3</td>
<td>219.5</td>
</tr>
<tr>
<td>( N_{\text{ICO}}, ) MW</td>
<td>0.00</td>
<td>27.0</td>
<td>0.0</td>
</tr>
<tr>
<td>( N_{\text{elGT}}, ) MW</td>
<td>200.00</td>
<td>200.00</td>
<td>200.00</td>
</tr>
<tr>
<td>( m_{LHV}, ) MW</td>
<td>516.3</td>
<td>568.5</td>
<td>516.3</td>
</tr>
<tr>
<td>( \eta_{\text{elGT}}, ) -</td>
<td>0.3873</td>
<td>0.3518</td>
<td>0.3873</td>
</tr>
<tr>
<td>( \dot{Q}_{4a}, ) MW</td>
<td>301.8</td>
<td>386.0</td>
<td>301.8</td>
</tr>
<tr>
<td>( \alpha_{4a}, ) -</td>
<td>1.509</td>
<td>1.930</td>
<td>1.509</td>
</tr>
<tr>
<td>( t_{4a}, ) °C</td>
<td>595.0</td>
<td>642.2</td>
<td>595.0</td>
</tr>
<tr>
<td>( N_{\text{elST}}, ) MW</td>
<td>100.1</td>
<td>141.0</td>
<td>74.5</td>
</tr>
<tr>
<td>( \eta_{\text{elST}}, ) -</td>
<td>0.3282</td>
<td>0.3654</td>
<td>0.2469</td>
</tr>
<tr>
<td>( N_{\text{el.gross}}, ) MW</td>
<td>301.1</td>
<td>341.0</td>
<td>274.5</td>
</tr>
<tr>
<td>( \eta_{\text{el.gross}}, ) -</td>
<td>0.5813</td>
<td>0.5998</td>
<td>0.5316</td>
</tr>
<tr>
<td>( \Delta N_{\text{el}}, ) MW</td>
<td>6.0</td>
<td>6.8</td>
<td>5.5</td>
</tr>
<tr>
<td>( \Delta N_{\text{ASU}}, ) MW</td>
<td>0.0</td>
<td>32.5</td>
<td>0.0</td>
</tr>
<tr>
<td>( \Delta N_{\text{CC}}, ) MW</td>
<td>0.0</td>
<td>11.6</td>
<td>8.9</td>
</tr>
<tr>
<td>( \delta_{\text{el}}, ) -</td>
<td>0.0200</td>
<td>0.1493</td>
<td>0.0525</td>
</tr>
<tr>
<td>( N_{\text{el}}, ) MW</td>
<td>295.1</td>
<td>290.1</td>
<td>260.1</td>
</tr>
<tr>
<td>( \eta_{\text{el}}, ) -</td>
<td>0.5715</td>
<td>0.5103</td>
<td>0.5037</td>
</tr>
<tr>
<td>( u_{\text{CO}_2}, ) kg/MWh</td>
<td>330.1</td>
<td>374.0</td>
<td>382.1</td>
</tr>
<tr>
<td>( e_{\text{CO}_2}, ) kg/MWh</td>
<td>330.1</td>
<td>7.5</td>
<td>38.2</td>
</tr>
</tbody>
</table>

Where \( u_{\text{CO}_2} \) is the \( \text{CO}_2 \) production in combustion process per every 1 MWh of produced net electric energy, and \( e_{\text{CO}_2} \) is a \( \text{CO}_2 \) emission per every 1 MWh of produced net energy.

### 4. Conclusion

- The thermodynamic assessment of the combined cycle gas turbine (CCGT), the combined cycle power plant with the oxy – combustion (OXY) and the unit with the chemical absorption (ABS) indicated the lower gas turbine efficiency in the OXY case than in the other cases with the air - combustion, efficiency were equal to 0.3518 and 0.3873, respectively. However, the heat amount supplied to the heat recovery steam generator in the OXY case is higher than for the other units by about 28%, additionally the flue gas is characterized by the higher temperature, equal 642°C in OXY, related to 595°C in the remaining cases. As a consequence the steam cycle in the OXY case achieves electric power higher by 39.9 MW than the CCGT case, resulting in the gross efficiency of the OXY unit equal to 0.5998, which is the highest value among the analyzed cases. The CCGT unit reaches gross efficiency equal to 0.5813. The steam extraction used in the ABS case causes a drop in the steam cycle power by 25.6 MW in relation to the CCGT case and in consequence significant decrease in the gross efficiency to the level of 0.5316.
The CCGT case has no additional installations and is affected only by minor own needs of gas turbine and steam turbine installations, thus, it achieves the highest net efficiency, amounting 0.5715. In the cases with CCS technology, apart from mentioned own needs, there are additional installations decreasing the net electric efficiency of those cases. The air separation unit in OXY case is characterized by a high own needs, amounting 32.5 MW. Additionally, in the OXY and ABS cases, the compression of the captured CO₂ increases the own needs of this cases by 11.6 MW and 8.9 MW, respectively. In OXY and ABS cases, due to the application of CCS technology, achieved net efficiency are lower than in the base case (CCGT) by 6.12 percent. points in OXY case, and by 6.78 percent. points in ABS case.

The use of the less energy consuming air separation unit, e.g. membrane or hybrid (membrane - cryogenic) ASU in the oxy-combustion unit shall cause the net efficiency improvement for the unit. The less energy consumption sorbents and the new concepts to provide the heat to regenerate the sorbent are sought to increase the efficiency of the combined cycle power plant with the CCS installation in post-combustion technology.

Literature

ASSESSMENT OF THE 3-D COMBUSTION MODEL IN THE MARINE 4-STROKE ENGINE

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Abstract

The aim of this research is to formation of the model of combustion process in the marine, 4-stroke, diesel engine. The chosen object of research is a laboratory AL25/30 engine. For achievement of the aim, laboratory measurements are made and the results are used to determine the boundary and initial conditions. In addition, measurements of the fuel injection shape are made on the test bench and are made a faithful geometric model of structural components of the engine cylinder. The obtained data has been implemented in the three-dimensional model comprising the fuel injection phenomena, the brake-up and the evaporation of fuel, auto-ignition, flame propagation and heat exchange with the structural elements of the engine cylinder. As a result of activities succeeded in creating a model of the combustion process in the cylinder, which has been positively validated due to the maximum combustion pressure and the temperature of the exhaust gases. Obtained results of calculations also allowed verifying the assumption of negligible auto-ignition delay. The adopted ECFM-3Z model of the ignition and combustion, used for modeling of combustion in diesel engines, showed the calculated size of the auto-ignition delay of 7-8 ° CA.

Key words: marine engine, multidimensional model, Dukowicz evaporation model, TAB brake-up model, ECFM-3Z combustion model

1. Introduction

The development of the construction of the marine piston engines in the XXI century goes towards the growth of the combustion process efficiency while gaseous emission reduction. Both of mentioned objectives can be achieved through the development of engine design for optimal combustion process throughout all engine load conditions. Optimization of the combustion process should proceed towards the obtaining a certain fuel ignition in possible large ranges of thermodynamic parameters in the engine cylinder. The fuel ignition should be provided during the engine operation at malfunction conditions such as leakage of the engine cylinder or malfunction of the air exchange system. Moreover, combustion process in the engine cylinder should be complete. Such state of affairs provides, inter alia, the high temperature of the combustion process [1]. Unfortunately, an excessive of the combustion temperature increase results in an increase of the nitric oxides emission. For this reason, the combustion process parameters should be governed for the engine load, the engine rotational speed and the technical condition of the engine. Such correction provides electronic control of the valve timing and the fuel injection process [2].

To the optimal control of the combustion process requires knowledge about the phenomena occurring in the engine cylinder. Suitable for this purpose are the methods based on the direct measurement of engine operating parameters. It should be noted, that the measurement parameters
of fluids from the engine operation may be insufficient. The temperature of the cooling water, oil, and exhaust gas does not reflect the temperature of the engine cylinder and the measurement of cylinder pressure may be burdened with a significant error [3]. Due to the rapidity of phenomena occurring in the engine cylinder direct measurement of the combustion temperature is not possible. Applying optical methods [4] is possible only after modernization of the engine construction in the laboratory conditions and the cost of this type measurements may be relatively high [5].

In recent years, numerical methods are gaining in popularity. The reason for this is the increase in computing power and significant decrease in the modeling cost. Numerical methods can be applied to both qualitative and quantitative evaluation of the phenomena occurring in the engine cylinder [6]. It should be noted, that such models must be used with great care and only after a positive validation of the combustion process parameters concerned with the data obtained through the direct measurements.

The aim of the work is to create a multi-dimensional model of the combustion process in the cylinder of the marine 4-stroke engine. This model is intended to allow the appointment of thermodynamic parameters of the engine exhaust gases and analysis of phenomena occurring during the combustion process.

2. The model description

Model phenomena occurring in the engine cylinder was prepared as Euler description [8]. The base model of the combustion process in the engine cylinder is a geometric grid, including the shape of the cylinder with the air intake duct, the outlet duct and exhaust and inlet valves. The geometric grid was built based on the technical documentation of the research object. As a research object was selected laboratory 4-stroke engine, type Sulzer Al25/30. Analysis and selection of spatial grid parameters are presented in [9]. Prepared grid is presented in Fig.1 and parameters of the Al25/30 engine in Tab.1.

![Fig.1. The grid of the engine cylinder](image-url)
2.1 The fuel injection model

The fuel injection model is based on the geometrical dimensions of the injector nozzle, which are presented in Tab.1. According to [10], page 526, the injected fuel cone angle depends on the differential pressure in the nozzle and in the cylinder chamber and the ratio of the diameter to the length of the injector holes. Due to the low credibility of the measurement of the length of the injector holes, the injected fuel cone angle was determined empirically on a test stand. The tests were conducted at atmospheric pressure, assuming that the injection pressure is many times greater than the pressure prevailing in the cylinder chamber. Fuel injection was photographed at a frequency of 60 frames per second, and measurement was made on the collected photographic material.

Mass of fuel injected was calculated on the basis of fuel consumption. Fuel injection characteristic was determined based on the measured characteristics of injection pressure, according to the following relation:

\[
\dot{m} = f(\sqrt{p_w - p_c}),
\]

where:
- $m$ – mass of injected fuel in [g/s],
- $p_w$ – injection pressure in [Pa],
- $p_c$ – pressure in the cylinder chamber in [Pa].

Assumed injection timing equals to 25 degrees before top dead center of the piston position. Start of injection equals to 695 degrees angular position of the crankshaft. The basis for this decision was to analyze the characteristics of the pressure on the indication valve. I assumed that the pressure signal in the indicator valve is not delayed relatively to the pressure in the engine cylinder. In addition, it is assumed that self-ignition delay is negligibly small.

2.2 The brake-up and the evaporation model

The initial value of the droplet diameter of the fuel injection is taken as 0.325 mm, which corresponds to the diameter of the nozzle holes. A further break-up of fuel droplets has been
described by the Lagrange description [8] with TAB model application [11]. This model specifies the conditions for breaking-up of fuel droplets as a dimensionless factor that depends on the density of the fuel and the surrounded air, the viscosity of the fuel droplet, the relative velocity and diameter of droplets. If the value of the mentioned ratio is greater than 1, the drop breaks up. Distribution of mean droplet diameter, determined by the Sauter method [10], is assumed as Chi^2.

Simultaneously with the fuel atomization process begins the process of evaporation. This process results from the heating of fuel droplets. For modeling of heat flow from the air to fuel droplets and mass flow of fuel vapors from the droplets to air the Dukowicz’s model is adopted [12]. The spherical shape of fuel droplets (microgravity conditions) and a constant temperature and heat transfer conditions on the surface of the droplet is assumed.

### 2.3 The combustion process

Evaporated fuel is mixed with air in the engine cylinder. To modeling these phenomena the k-ε model [6] were using. The combustion process was described by the ECFM-3Z model. Mentioned model is prepared by Groupement Scientifique Moteurs consortium [13]. It is a model developed for modeling the combustion in diesel engines and it belongs to the CFM (Coherent Flame Model) class of models. The model can be used for both direct injected [14], and spark ignited engines [15]. This model assumes that the chemical reactions take place in the relatively narrow layer of the flame. The flame progresses to the direction of fresh mixture of air and fuel. Mentioned flame layer is defined a homogeneous mixture of fuel and air and its shape and size is diffusion phenomena. In the present model, the self-ignition delay is determined by air temperature, the density of the mixture and the molar concentration of oxygen and fuel. Chemical kinetic calculations are prepared for assumed substitute fuel composition in the form of hydrocarbon C_{13}H_{23}.

### 3. The laboratory stand

Boundary conditions and initial conditions, as well as data necessary for validation of this model were collected during laboratory tests. The laboratory researches are carried out on a laboratory engine Al25/30, which basic parameters are presented on Tab.1. During the laboratory research the engine operate at a constant speed and load equal 250kW. The engine was fueled by diesel oil with known specifications. The research covered the registration of parameters of turbocharger, fuel system, lubrication, cooling and air exchange systems. All parameters are measure with a sampling time equal to 1 second and used for modeling the data was the arithmetic average of 3 observations. Closer description of laboratory research can be found in [16].

### 4. The validation model and obtained results

Initial conditions of the model are measured during laboratory research pressure and temperature of the charge air in the air intake duct. These conditions were adopted for the entire volume of the intake manifold and the engine cylinder at the crank shaft angular position corresponding to beginning of the closing of the intake valve. Moreover, boundary conditions are determined, as the temperature of individual structural elements of the engine cylinder. The temperature of the intake valve with the inlet channel is assumed as measured temperature of the charge air. The boundary temperatures of the cylinder liner, piston and cylinder head are assumed as the lubricating oil temperature, measured behind the engine. For the temperature of the surface of exhaust valve with the exhaust duct is assumed temperature of the exhaust gas, measured behind the modeled cylinder of the engine.
The calculations are carried out from the beginning of the intake valve closing (575 degrees of the contractual angle of crankshaft position - CA), by compression and combustion processes until the end of the opening of the exhaust valve (contractual 850° CA). The top dead center of the piston position during the power stroke engine has the contractual CA equal 720°. Each step of the calculation corresponds to the CA required from 17 to 100 iterative calculations of the balance equations of momentum, energy and continuity for each grid cells. The convergence criterion for momentum, pressure and energy was assumed as the change in values of mentioned parameters is not larger than 0.01 after each iteration. The variable step of calculation was assumed from 0.01 to 1 degree of CA also. The results of calculations of each step are the input to the calculations for the next step. Calculations were performed using Fire software in 2013.1 version from AVL manufacturer.

On the 2 drawing the results of model validation are presented. Criteria of the model validation are assumed temperature of the exhaust gas after cylinder at the opening of the outlet valve (contractual 840° CA) and maximum combustion pressure in the engine cylinder.

![Fig.2. The calculated exhaust gas temperature and maximum pressure (p max)](image)

According to the presented results, calculated values are smaller from measured by 2.8% in the case of the exhaust gas temperature and by 4.5% in the case of the maximum in cylinder pressure.
On the left side of the Fig.3 the average values of temperature and pressure calculated for the whole volume of the engine cylinder are shown. It should be noted that obtained results are qualitatively consistent with results available in the literature [10]. On the right side of the Fig.3 the characteristics of the injection, vaporization and combustion of the fuel in the engine cylinder are shown. As previously mentioned, the fuel injection characteristic is a function of the (1) equation.

Fig.4. The temperature of the combustion process

In the initial period of fuel injection at a relatively low pressure and temperature in the engine cylinder, the amount of the evaporating fuel is less than the amount of injected fuel. Ignition of the fuel, referred to the sudden increase in the fraction of fuel burned, causes an increase in
temperature and pressure in the cylinder. The result of this is an increase of the amount of evaporating fuel to a value equal to the amount of injected fuel. The residual value of the fuel burned fraction in the initial stage of the combustion process is the result of chemical dissociation of fuel before auto-ignition. It should be noted that illustrated in Fig.3 temperature values are the average values of temperatures for the entire volume of the cylinder. Local temperatures differ significantly from the average values.

Figure 4 shows temperature changes in the engine cylinder with the rotation of the crankshaft on the example of the axial cross-section of the cylinder. Cross-section of the cylinder passes through the axis of the right side of the nozzle hole. The lower profile of the cross section corresponds to the shape of the piston bottom. According to the presented results, auto-ignition of the fuel occurs 3-4° before TDC of the piston position and causes a sudden increase of temperature. Since then, the development of combustion occurs towards a cooler piston bottom and the cylinder walls. The initial combustion period is very fast, because the combustion rate is determined by kinetic phenomena that are taking place in a mixture of fuel and air. At the first, the turbulent combustion is slowed down at the moment of lack of combustible mixture in the cylinder. According to the result from Fig.3, starting from the TDC of the piston position (contractual 720° CA), the combustion rate is determined by the fuel injection characteristics and diffusion phenomena, occurring on the surface of the liquid fuel droplets. A symptom of this state of affairs is to cover the characteristics of fuel injection and fuel evaporation.

It should be noted that calculations show a significant delay of auto-ignition. Assumed models and boundary and initial conditions have resulted in auto-ignition delay equal to 7-8° CA. For this reason, it must be assumed that the auto-ignition delay is not negligible, as assumed. Furthermore, according to the cited measurement results carried out on the same research facility [3] there is a significant delay between the measured pressure signal on the indicator valve and the signal measured in the engine cylinder. Therefore, the conclusion must accept that the determination of injection based on the characteristics of the pressure signal on the indicator valve gives a result considerably delayed due to two discussed phenomena.

According to results, presented in Fig.4, the maximum combustion temperature comes to 2500K. It should also be noted that at large part of the cylinder volume the temperature of the combustion process is high enough to allow oxidation of the nitrogen in accordance with the thermal Zeldovich mechanism [10].

5. Conclusions

The aim of this research was to formation of the model of combustion process in the marine, 4-stroke, diesel engine. The chosen object of research was a laboratory AL25/30 engine. For achievement of the aim, laboratory measurements were made and the results were used to determine the boundary and initial conditions. In addition, measurements of the fuel injection shape were made on the test bench and made a faithful geometric model of structural components of the engine cylinder. The obtained data has been implemented in the three-dimensional model comprising the fuel injection phenomena, the brake-up and the evaporation of fuel, auto-ignition, flame propagation and heat exchange with the structural elements of the engine cylinder. As a result of activities succeeded in creating a model of the combustion process in the cylinder, which has been positively validated due to the maximum combustion pressure and the temperature of the exhaust gases. This model allowed for a qualitative assessment of the processes occurring in the engine cylinder. The analysis of the temperature distribution in the cylinder showed significant differences of temperature in the volume of the cylinder. As a result of these differences in the maximum temperature reached in the cylinder combustion to 2500K, and the average for the entire volume of the cylinder is lower than 600K.
Obtained results of calculations also allowed verifying the assumption of negligible auto-ignition delay. The adopted ECFM-3Z model of the ignition and combustion, used for modeling of combustion in diesel engines, showed the calculated size of the auto-ignition delay of 7-8 ° CA. Therefore, further work is needed on the quantitative validation of the obtained model.

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References

3D MESH MODEL FOR RANS NUMERICAL RESEARCH ON MARINE 4-STROKE ENGINE

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Abstract

The article consist the 3d mesh analysis prepared for simulation of the processes in combustion chamber of marine compression ignition engine. The three moving meshes models where prepared: A – mesh for engine cycle work simulation; B – mesh of combustion chamber volume for work stroke simulations, no valves included; C - mesh of combustion chamber including mountings screw whole for work stroke simulations, no valves including. Prepared mesh where used for numerical simulations of injection and combustion processes in engine combustion chamber. Type C model, even if the total number of cells is lower in comparison to B model, result in calculation time increase. B and C models are solution for fast and robust validation of injection and auto ignition model parameters. Type A model is only one suitable for full cycle simulation. Only with accurate initial and boundary conditions the qualitative results of the injection, mixing and combustion process can be obtain on mesh type B and C.

Key words: CFD, RANS, marine engine, moving mesh, 3d mesh

1. Introduction

Nowadays and future regulations regarding environment protection and combustion emission products limitations are source of a need to design engine constructions with higher efficiency. Solution to decrease the research and development costs is to use computer fluid dynamics (CFD) technics, which is an effective tool for analyse and verification of the fluid flow and combustion processes in piston engine. According to Drake et al. [1], dynamic development of CFD methods in modelling of the piston engine processes started with the development of port fuel injection piston engines. The processes of the energy conversion to mechanical work from combustion in piston engine are complicated to describe numerically. In simulation there is a need for injection, mixing, ignition and combustion models verification. Numerical representation of the fluid flow, pressure temperature field, heat exchange in flow and on boundary wall need to be described also precisely. Assuming the number of thermodynamic processes which need to be defined in CFD and continues hardware development, simulation methods and models are also continuously...
developed. The newest injection and combustion models for compression ignition diesel driven engine are fully validated [2]. According to Collin et al. [3] today every new construction of the compression ignition engines is optimised by use of CFD methods.

Proper numerical analyse of engine processes needs preparation of the accurate representation of engine geometry, by mean of the finite volume elements (mesh) model. Also piston and valve movement need to be included. Mesh should be prepared together with the knowledge about simulated process. Still the final resulting mesh is an optimal solution for robust calculation in time. In presented paper the methodology of mesh preparation will be shown for marine compression ignition engine together with comparison of different solution for simulation purpose.

2. Methods of mesh preparation

Geometrical data about piston chamber and valves dimension together with data about piston and valves movement were prepared. The chosen compression ignition engine is from Maritime University laboratory, Sulzer A125/30, construction typical for marine purpose. Basic geometrical engine parameters are presented in Tab. 1.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. electric power</td>
<td>250</td>
<td>kW</td>
</tr>
<tr>
<td>Rotational speed</td>
<td>750</td>
<td>rpm</td>
</tr>
<tr>
<td>Cylinder number</td>
<td>3</td>
<td>–</td>
</tr>
<tr>
<td>Cylinder bore</td>
<td>250</td>
<td>mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>300</td>
<td>mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>12.7</td>
<td>–</td>
</tr>
<tr>
<td>Injector nozzle</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Holes number</td>
<td>9</td>
<td>–</td>
</tr>
<tr>
<td>Holes diameter</td>
<td>0.325</td>
<td>mm</td>
</tr>
<tr>
<td>Holes position diameter</td>
<td>7</td>
<td>mm</td>
</tr>
<tr>
<td>Holes position angle</td>
<td>150</td>
<td>deg.</td>
</tr>
<tr>
<td>Spray cone angle</td>
<td>6</td>
<td>deg.</td>
</tr>
<tr>
<td>Opening pressure</td>
<td>25</td>
<td>MPa</td>
</tr>
</tbody>
</table>

Three moving mesh models where proposed for the purpose of numerical simulations:

A – mesh for full engine cycle simulation presented on Fig. 1,
B – assuming axis symmetry of combustion chamber, moving mesh representing combustion chamber volume for one injector hole was prepared Fig. 2a,
C – assuming axis symmetry of combustion chamber, moving mesh representing combustion chamber with injector and mounting screw elements volume for one injector hole was prepared Fig. 2b.

Mesh A, presented on Fig. 1 where prepared for whole engine cycle, 720°. It was done with use of AVL Fire software, “Fame Engine Plus” (FEM+) module. Geometrical models were drawn for every engine stroke on fluid side. As the results four geometrical fluid models, representing exhaust, scavenging, intake, compression and combustion stroke were imported in to the Fire workflow manager. The reason for different models preparation is the result of assumption that after the valves are closed the simulation of the fluid flow in intake or exhaust channels can be omitted. It also decreases significantly calculation time.
On Fig. 3 the edges (green) representation for type A geometrical model is shown. Mesh calculated in FEM+ module is based on geometrical and edge input data. Edges are elements for proper definitions of surface cross section. Piston movement where conducted with use of crankshaft geometrical data by methodology presented by Heywood [5]. Valves movement where prepared by construction analysis of laboratory engine. For proper movement representation also selections on movement elements are defined. On Fig. 4, the piston and valves selection for non-moving, buffer and moving elements is presented. Proper choice of movement definitions is key element for robust and fine mesh generation.
Calculated mesh model consist of 500 000 cells at work stroke and 1 500 000 cells at scavenging stroke. Base cells size is 2 [mm] minimum and 8 [mm] maximum. Cells size should be defined by geometrical and simulated process complicity.

Flow velocity, mixture preparation and combustion processes scale related to combustion chamber complicity are the key parameters to decrease the cells characteristic size. Simultaneously, smaller cells lead to increase of the total number of cells and calculation time. To fulfil those statements the maximum cell size is limited to 2 [mm] for injection and combustion period and 0.125 [mm] for valves ports during the opening and closing crank angle degree. On Fig. 5 it is shown the mesh cross section view for scavenging stroke. Piston and liner includes also cut-off for valve opening. On Fig. 6 it is shown that the distance between open valve and liner cut-off surface is around 1 [mm] and distance between open valve and piston cut-off surface and top dead centre position is 2 [mm]. Mesh resolution for such area is also refined to avoid convergence problem because of the velocity increase.
Valves and piston surface engine elements which distance between the liner surface are from 1 to 2 [mm] and the surface move is parallel to each other can cause cells skewness. To avoid such result, which can cause negative cells preparation, not only cell size need to be refined (0.25 [mm]), but also mesh movement resolution need to be decrease to 0.5°.

Models type B and C were prepared with use of ESE Diesel tool form AVL Fire software. Models are based on assumption that geometrical and fluid flow characteristic is axis symmetric. Also it is possible to divide the combustion chamber volume according to angle symmetry of the injector nozzle holes. It results with model covering 40° of the combustion volume chamber for one nozzle injector hole. Methodology to prepare such mesh is simpler and it is based on the surface regular mesh rotate by 40° and 25 divisions. Models type B and C were divided in to two areas, injection area with 0.5 [mm] size and the rest volume with 1 [mm] size. For those models the valve movement is not described, also it is assumed that piston/liner cut-off and clearance between piston and liner is neglected.

For B type model the injector nozzle and mounting screw whole is also neglected.

### 3. Results

Calculations with all considered grids require the input of initial and boundary conditions data as well as the fuel injection parameters. Mentioned data were collected during laboratory tests, which the course and the results are presented in [5]. In addition, it is necessary to implement models of fuel injection, fuel spray brake-up, evaporation and combustion. The analysis of these problems is presented in the work [6]. It should be noted that the spatial geometry of the B and C meshes does not allow to model the full cycle of the engine operation. Omission of the valve geometry does not allow the modeling of the cylinder scavenging.

The computation time depends largely on the computing power of the computer, the number and complexity of solved equations, consisting of the combustion process and the size and complexity of the spatial grids. It is important problem because the calculation of one the engine cycle of 720° crankshaft angle (CA) with the single processor computer (4-core processor with the 3.4 GHz clock frequency, 16Gb of RAM memory and 64-bit operating system) take about 250 hours.

The Tab.2 presents a summary of the built grids parameters and the results of calculation time for the same input data. The calculations were carried out for the compression and combustion processes in the engine cylinder, starting from 140° CA before top dead center of the piston (TDC) to 130° CA after TDC. The presented calculations were performed on the same computer set and using the same equations and input data. Calculation time was measured using the internal software module of the Fire AVL package.
According to the presented results, average time for one calculation iteration is the longest for the C grid and the shortest for the B grid. It means that calculation time is largely dependent on the shape and structure of the individual cells of the mesh. It should be noted that the overall computation time is also influenced by the number of calculated iterations. According to the presented results, the best convergence of numerical calculations is obtained for grid A.

**Tab.3. The iteration number for characteristic steps of calculations**

<table>
<thead>
<tr>
<th>Grid</th>
<th>A</th>
<th>B</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Start of calculations – first 1°CA</td>
<td>Steps</td>
<td>100</td>
<td>10</td>
</tr>
<tr>
<td></td>
<td>iterations/step</td>
<td>19,9</td>
<td>74,3</td>
</tr>
<tr>
<td>Compression – next 119°CA</td>
<td>Steps</td>
<td>151</td>
<td>119</td>
</tr>
<tr>
<td></td>
<td>iterations/step</td>
<td>73,6</td>
<td>94,5</td>
</tr>
<tr>
<td>Start of fuel injection – next 28°CA</td>
<td>Steps</td>
<td>240</td>
<td>240</td>
</tr>
<tr>
<td></td>
<td>iterations/step</td>
<td>22,5</td>
<td>98,0</td>
</tr>
<tr>
<td>Combustion – next 106,5°CA</td>
<td>Steps</td>
<td>213</td>
<td>221</td>
</tr>
<tr>
<td></td>
<td>iterations/step</td>
<td>16,3</td>
<td>64,5</td>
</tr>
<tr>
<td>Exhaust valve opening – next 20,5°CA</td>
<td>Steps</td>
<td>62</td>
<td>41</td>
</tr>
<tr>
<td></td>
<td>iterations/step</td>
<td>50,7</td>
<td>55,9</td>
</tr>
</tbody>
</table>

As mentioned earlier, presented the average computation time is also apparent from the amount of implemented equations in the model. The average computation time for each model, which make up the model of the combustion process in the engine, for a single iteration is shown in the Tab.2. The presented results show that considered grids are optimal solutions for the selected computational models. Grid A is the best solution for the calculation of the fuel injection and grid B for the calculation of emissions, energy and turbulence phenomena. Increasing the accuracy of mapping the shape of the cylinder about the shape of the fuel injector in a C grid resulted in a significant increase in computation time for all considered model’s equations of the combustion process. An additional effect was an increase in the number of iterations.

The size of the calculation steps for the characteristic engine operation phases expressed in the angle of CA are presented in the Tab.3. The average number of iterations which are needed to obtain a result of the assumed accuracy, per one step calculation, is also presented.

According to the presented results, increase of the calculation step leads to an increase in the number of iterations. It must be remembered that the use of the excessive computational step size, decreases the convergence of the calculation, leading to an increase in the number of iterations.
This conduct may result in a significant increase in computation time and thus counterproductive. An example of this is the average number of iterations for the opening of the engine exhaust valve. Not implemented valves geometry in the B and C grids, allowed reducing the amount of calculation steps. Despite this, there has been an increase in the average number of iterations for one step. As a result, the computation time has not reduced significantly.

It should be remembered that the analysis of the optimal choice of the grid should also be conducted based on the validation of computational results obtained with the measured data. Calculations with B and C grids cannot be prepared for subsequent cycles of the combustion process. The advantage of these grids is the ability to quickly tune the boundary conditions and the parameters of the models of the combustion process. We can conclude that the B and C moving grid, based on the axial symmetry assumption are not consistent with the real model, but they can give a quick solution, but only in the case of a regular structure. Grids of this type may be a prelude to the calculation of the A grid.

It should also be noted that the advantage of the A grid is fully possible to estimate the velocity field, pressure and temperature at the start of the compression stroke with consideration of the fluid flow in a perpendicular direction to the cylinder axis.

4. Conclusions

The article consist the 3d mesh analysis prepared for simulation of the processes in combustion chamber of marine compression ignition engine. The three moving meshes models where prepared: A – mesh for engine cycle work simulation; B – mesh of combustion chamber volume for work stroke simulations, no valves included; C- mesh of combustion chamber including mountings screw whole for work stroke simulations, no valves including. Prepared mesh where used for numerical simulations of injection and combustion processes in engine combustion chamber. Type C model, even if the total number of cells is lower in comparison to B model, result in calculation time increase. B and C models are solution for fast and robust validation of injection and auto ignition model parameters. Type A model is only one suitable for full cycle simulation. Only with accurate initial and boundary conditions the qualitative results of the injection, mixing and combustion process can be obtain on mesh type B and C.

Acknowledgments

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References


THE ANALYSIS OF THE ECFM-3Z COMBUSTION MODEL IN THE MARINE 4-STROKE ENGINE FOR THE EXHAUST GAS COMPOSITION

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Abstract

The paper presents ECFM-3Z combustion model analysis in the marine, 4-stroke diesel engine. The purpose of the modeling was to determine the composition of the exhaust gas. This composition depends on the composition of the combustible mixture, combustion time and thermodynamic conditions prevailing in the engine cylinder during the working process. Mentioned parameters are variable in time and space, and therefore require the use of 3-dimensional model based on the finite volume method, taking into account the fuel injection, brake-up and evaporation, mixing with air, auto-ignition and combustion. All models presented in the literature are adapted to the parameters of relatively small engines. Different marine engine parameters require significant modifications taking into account the heat exchange with the structural elements of the engine, leakage through piston rings and energy losses by friction. It should also be noted that dimensions of the marine engine require careful optimization of spatial moving meshes according to computation time and quality of results. Paper presents influence of mixing time, start of injection and autoignition delay on modeling results of the exhaust gas composition.

Key words: marine engine, multidimensional model, mixing time, autoignition delay, ECFM-3Z combustion model

1. Introduction

The aim of the design development of marine piston engines is the reduction of fuel consumption and the reduction of toxic compounds emissions into the atmosphere. These objectives require the knowledge of parameters of phenomena occurring in engine cylinders during its operation. Parameters of the combustion process in the engine cylinder determine the composition of the exhaust gas. The assessment of phenomena, resulting in the emissions of toxic compounds into the atmosphere, requires a multi-dimensional modeling of the propagation of flame in the engine cylinder. This process is made up of many co-existing physical phenomena. Mentioned phenomena’s are the injection of fuel into the engine cylinder, fuel brake-up and evaporation, mixing with air, autoignition and flame propagation in a heterogeneous combustible mixture. It should be noted, that the result of the combustion process occurring are a number of chemical reactions. Chemical reactions are determined by temperature and pressure in the engine cylinder but also by the composition of the combustible mixture, the combustion chamber geometry, the phenomena of heat transfer, gas movement, leaks by the piston-rings-cylinder liner system etc.

Many models of combustion processes are developed. In the last 10 years increasingly popular in the modeling of combustion processes are Coherent Flame Models (CFM) [1]. CFM models describe the combustion process on the assumption that the scale of chemical reactions is many
times smaller than the scale of the turbulent flow. This enables the separation of both phenomena models. It’s further assumed that chemical reactions take place only in a very thin surface layer of the flame, which the shape and the location depend on of the turbulent flame propagation phenomena. CFM model was modified by Colin and Benkenida [2] in 2004. Mentioned model, named Tree Zone Extended Coherent Flame Model (3Z-ECFM), allows obtaining correct results of modeling for diesel engines. Mobasheri et al. [3], [4] apply 3Z-ECFM model to develop a strategy for fuel injection into the engine cylinder to reduce the NOx and the soot emissions. Moreover, authors tested and optimized the split fuel injection into the engine cylinder with capacity of 2.5dm³. Authors present Homogeneity Factor, a new parameter for supporting the air–fuel mixing and the combustion process in diesel engines. Taghavifar et al. [5] use 3Z-ECFM model to modification the structure of the combustion chamber of a small engine to changing the mixture formation, the combustion initiation and emissions. The 3Z-ECFM model is useful to predict of exergy parameters of a small dual fuel, high speed diesel engine [6] and real time predict of the NOx emission [7]. The 3Z-ECFM model with Eulerian–Lagrangian Spray Atomization model were used to simulation of primary break-up and atomization processes also [8].

Presented works show wide possibilities of modeling of the combustion processes using the model 3Z-ECFM. It should be noted that engines used in shipbuilding are significantly different from small, on-road engines and engines used in the automotive industry. Main differences are i.e. relatively low speed of marine diesel engines, the large stroke in relation to the cylinder bore, the ignition of fuel before TDC (top dead center), the boost pressure greater than the exhaust gas pressure for all loads of the engine, higher compression ratio and a large heat exchange surface in relation to the cylinder volume. These differences cause that default settings of the 3Z-ECFM model allow the calculate parameters of the combustion process only on a small scale. The model of the combustion process in the marine engine cylinder was presented in [9]. Default settings of the 3Z-ECFM model allow to correct modeling of thermodynamic parameters of the combustion process. Modeling of the combustion process in the marine engine cylinder to assess the composition of the exhaust gas requires a modification of mentioned model parameters.

The main purpose of the study is the analysis of selected 3Z-ECFM model parameters to assess the composition of the exhaust gas from the marine diesel engine. Following parameters of the 3Z-ECFM model of the combustion process are taken into account in the presented analysis: start of injection angle (SOI), the autoignition delay and the intensity of fuel and air mixing (mixing time).

2. The 3Z-ECFM model description

Phenomena occurring in the engine cylinder were modeled as the Euler description [10]. The model base of the combustion process in the engine cylinder is a geometric grid, including the shape of the cylinder with the air intake duct, the outlet duct and exhaust and inlet valves. Analysis and selection of spatial grid parameters are presented in [11]. Parameters of the laboratory engine are presented in Tab.1. Evaporated fuel is mixed with air in the engine cylinder. To modeling these phenomena the k-ε model [12] was used. The combustion process was described by the ECFM-3Z model. In the present model, the autoignition delay (τ) is determined by air temperature (T), the density of the mixture (ρ) and the molar concentration of oxygen ([O₂]) and fuel ([Fuel]) for all cells of grid by the following equation [13]:

$$\tau = 4,804 \cdot 10^{-8} \cdot [O_2]^{-0.53} \cdot [Fuel]^{0.05} \cdot [\rho]^{0.13} \cdot e^{\frac{5914}{T}}$$, \hspace{1cm} [1]

The autoignition delay, calculated according to equation 1 was delayed from 0 to 43% during the analysis. The composition of the exhaust gas depends on mixing time of fresh fuel and fresh air before combustion. The phenomenon of mixing was defined by standard 3Z-ECFM model equations [13]. The mixing time was changed from 0 to 420% in comparison to the standard
parameters of the mixing time during presented analysis. The Dukowicz fuel evaporation model [14] and TAB fuel brake-up model [15] are used. The dipper description of the combustion model was presented in [9].

**Tab. 1. The laboratory engine parameters**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. electric power</td>
<td>250</td>
<td>kW</td>
</tr>
<tr>
<td>Rotational speed</td>
<td>750</td>
<td>rpm</td>
</tr>
<tr>
<td>Cylinder number</td>
<td>3</td>
<td>–</td>
</tr>
<tr>
<td>Cylinder bore</td>
<td>250</td>
<td>mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>300</td>
<td>mm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>12.7</td>
<td>–</td>
</tr>
</tbody>
</table>

### 3. Results

An application of standard values of the autoignition delay and the mixing time in the 3Z-ECFM model allows obtaining correct results of pressure distribution in the cylinder compared to measured values. Please note that measured values essentially concern pressure on the indicator cock. Mentioned pressure value differs from the pressure values in the engine cylinder [16]. The left side of Fig.1 shows pressure characteristics in the cylinder for the default 3Z-ECFM model values and direct measurements.

![Fig.1 Results of in cylinder pressure measurement and ECFM-3Z model calculation results with default values](image)

According to presented results, the maximum measured pressure is greater about 8.3% than the calculated value. It should be noted that results of modeling require the correct selection of parameters such as SOI, characteristics of fuel injection into the cylinder and fuel consumption, determine geometric dimensions of the fuel injector and injected fuel stream and the pressure and temperature of charge air. The method of selection of these parameters is presented in [9]. This model is valid only in appearance. Analysis of results of toxic compounds fractions in the exhaust gas show significant discrepancies with the measured values. The right side of Fig.1 presents results of temperature distribution as a function of CA (crankshaft angle). Presented results are average values for the entire cylinder volume. Calculated fractions of nitric oxide (NO), carbon monoxide (CO) and carbon dioxide (CO₂) in the exhaust gas are presented also. **Tab.2** presents the comparison of measured fractions of listed chemical species in the exhaust gas with the calculated values. For comparison, the average values of the weight fractions for the entire volume of the
exhaust gas duct and the crank angle between 110° after TDC and 140° after TDC. According to presented results calculated NOx fraction is many times lower than the measured fraction. Calculated CO fraction is higher than the measured CO fraction. The reason for this is too low modeling combustion temperature, in the comparison to real conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>NOx [ppm]</th>
<th>CO [ppm]</th>
<th>CO₂ [%]</th>
<th>p_{max} [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement</td>
<td>788</td>
<td>715</td>
<td>6.3</td>
<td>8.5</td>
</tr>
<tr>
<td>Model with default values (SOI -21°)</td>
<td>78</td>
<td>5782</td>
<td>5.44</td>
<td>7.79</td>
</tr>
</tbody>
</table>

Presented results support the conclusion, that obtaining values of calculated average pressure for the entire cylinder volume, similar to measured values, do not allow for proper modeling of the composition of the exhaust gas. The use of default 3Z-ECFM model parameters results in a lower combustion temperature than expected. As a result, incomplete fuel combustion occurs (increase of CO fraction in the exhaust gas) and the NOx fraction in the exhaust gas decreases.

The way to increase calculated temperature of the combustion process is the earlier SOI and the autoignition delay. Such a set of model parameters can result in approaching the calculated value of NOx and CO fractions to measured values. The autoignition delay causes the greater part of fuel evaporates before autoignition. The effect of this is the increase of the effect of combustion controlled by chemical kinetics and the reduction of the influence of the diffusion in the combustion process on calculation results. The autoignition delay in the 3Z-ECFM model can be carried out by limiting the flow of energy from the in-cylinder air into vaporized fuel (increase of
the mixing time). The physical interpretation of this parameter is the extension of the heat flow from air to evaporated fuel, caused by improper selection of parameters of fuel injection process into the cylinder. Presented Eq.1 is an empirical equation, adapted for modeling of the autoignition delay in a relatively small piston engines. Extension of mixing time parameter causes fuel autoignition delay and thus increases of heat release rate in the initial stage of the combustion.

Fig.2 presents the influence of autoignition delay changes on the value of mass fractions of NOx, CO and CO2 in the exhaust gas. Characteristics of average temperature and average pressure for the entire cylinder volume and the intensity of the heat release from the combustion process are presented also. Data comes from calculations for the engine load equal 220kW at 750rpm, the SOI -21° and the mixing time equal 400% of default model values.

According to presented results, the increase of the autoignition delay shifts the combustion process on the expansion stroke. The result of this is the intensification of the combustion process, despite the slight decrease of pressure in the cylinder. The 43% increase of the autoignition delay results the increase of the maximum heat release by 26%. Effect of this is the increase of average temperature of the combustion process by 60K. The increase of combustion process temperature causes the increase of the NOx mass fraction in the exhaust gas. The increase of temperature promotes the oxidation of CO to CO2. For this reason, Fig.2 presents increase of the CO2 fraction in the exhaust gas with a decrease of the CO fraction.

Fig.3 presents the influence of the mixing time parameter on modeling results. These results correspond to the engine load equal 220kW at 750rpm, the SOI -18° and the autoignition delay equal 143% of standard values of the model. According to presented results changing the mixing time does not change the characteristic of pressure in the engine cylinder. This means that the change of this parameter is not visible during the in-cylinder pressure analyze. Therefore it can be
concluded that it is possible to adjust parameters of the model of the combustion process, which result is changes in the calculated composition of the exhaust gas without modifying the characteristic of in-cylinder pressure. Increasing the mixing time causes a slight increase in the intensity of the combustion process. The increase of the mixing time by 80% (from 340% to 420% of the default value) causes the increase of the maximum heat release by about 16%. The effect of this is the increase of combustion process temperature. Mentioned temperature increase is not as large as in the case of increasing the autoignition delay. It should be noted that the increasing the mixing time will slightly slow down the combustion process. Increasing the mixing time of more than 400% moves the combustion process in the direction of the expansion stroke, and causes slight deterioration of the combustion process. The result of this is the increase of CO fraction in the exhaust gas with the constant value of the NOx fraction in the exhaust gas. The conclusion is that the mixing time increasing over 400% will not affect the growth of the NOx fraction despite the increase of the intensity and temperature of the combustion process. It should be noted that the increase of the SOI angle does not cause a qualitative change dependances presented on Fig.3. Therefore, it can be assumed that 400% of the default value of the mixing time is the adjustment limit for this type of the engine.

An important parameter for the composition modeling of the exhaust gas is SOI (start of injection angle). This parameter can be measured directly by static experiment or estimated by the analysis of the in-cylinder pressure characteristic. Both methods are not accurate therefore, the influence of the SOI value on the modeled exhaust gas composition was considered also.

Fig.4 presents influence of the SOI angle on modeling results. Presented results correspond to the engine load equal 220kW at 750rpm, the mixing time equal 420% and the autoignition delay equal 143% of default model values.
The increase in the distance between the SOI and the TDC causes early autoignition of fuel. The result of this is the intensity change of the combustion process. The analysis of heat release characteristics shows that at the SOI equal to 18°CA, mentioned combustion intensity is the lowest. It should be noted that this is the value of the SOI measured on a real object and a significant delay of the combustion is caused by maximum considered values of the mixing time and the autoignition delay. As expected, the increase of the SOI angle reduces the NOx fraction in the exhaust gas. This is due to the slowdown in the combustion process and the reduction of combustion temperature. The result is a deterioration of the combustion process, resulting in a rapid increase of the CO fraction in the exhaust gas. Analysis of obtained results of the CO2 and the NOx fractions shows that there is a certain interval of the SOI angle, that the contents of both chemical species are the largest. The conclusion is that a significant delay of SOI does not result in the expected limitations of the NOx fraction in the exhaust gas.

4. Conclusions

The paper analyzed the effect of the SOI angle, the autoignition delay and the mixing time on modeling results of the combustion process. The purpose of the modeling was the validation of the 3Z-ECFM model in terms of the composition of the exhaust gas of marine diesel piston engine. The analysis allows drawing following conclusions:

- Default settings of 3Z-ECFM model parameters do not provide correct results of modeling the combustion process in the marine piston engine. Required modification resulting in the increase of temperature of the combustion process. This can be achieved by delaying of the autoignition of the combustible mixture.
- The greatest impact on changing of modeling results has changes in the autoignition delay and the SOI angle. Earlier fuel injection and/or the autoignition delay increase the intensity of the combustion process.
- The validation of the combustion process model to assess the composition of the exhaust gas by analyzing the characteristic of combustion pressure is insufficient. According to obtained results it is possible to change the calculated composition of the exhaust gases without modifying the characteristic of the engine cylinder pressure. This can be done by adjusting the mixing time in the 3Z-ECFM model.
- The change of the SOI angle causes a rapid change of the CO content in the exhaust gas. For this reason, this value should be chosen with as high accuracy.
- Significant delay of the SOI does not result in the expected limitation of the NOx fraction in the exhaust gas. The certain interval of the SOI angle, that the contents of CO2 and NOx in the exhaust gas are the largest was observed.

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References


ISSUES OF ECONOMIC ANALYSIS OF ELECTRIC ENERGY GENERATION IN A FLOATING POWER PLANT

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Abstract

This article presents basic dependences for conducting a technical and economic analysis related with investment projects. It describes a combined system of a compression ignition engine and a steam turbine, whose aim, when placed on a floating platform, is to generate electric energy in the so-called distributed generation system. Such a system involves a reciprocating internal combustion engine and a connected steam turbine system that uses the energy contained in the exhaust gas of the combustion engine. Assumptions and restrictions, as well as results of calculations are given in the article. Calculations were conducted only in view of investment and operational costs. The parameters of Wartsila and MAN Diesel & Turbo low-speed combustion engines of the capacity of approximately 54 MW were used for the analysis, and on the basis of them the cost of generating 1 MWh of electric energy was estimated.

Keywords: electric energy, compression ignition engine, steam turbine, floating power plant

1. Introduction

Business entities to a large degree determine the economic development of a country, as well as the welfare of its society. Each of them produces a part of gross domestic product. The new social and economic system that has been developing in Poland since the beginning of the 90s led to the evaluation of free market economy. Processes taking place in the global economy had a dramatic impact on the changes in the conditions under which Polish enterprises operate. The changes were mostly caused by factors such as globalisation, regionalisation, internationalisation, production and services concentration, development of financial markets, modern technology and logistics development. Furthermore, the increasing importance of environmental protection, as well as knowledge, entrepreneurship and innovativeness also played a role.

Continually changing conditions, increased requirements of customers, as well as growing competition force business entities to take regular actions maximising the efficiency of their operation. These days decision-making without quickly available and reliable technical and economic information is impossible. Nevertheless, sole possession of the information is not sufficient in order to evaluate the efficiency of an enterprise. A reliable assessment may only be performed through a financial analysis involving technical and economic data.

For many years now the solutions to power plant technical systems that would increase their thermodynamic efficiency have been sought. Thermal power stations fuelled with solid and
liquid or gas fuels almost always involve systems with turbines driving power generators (in the case of high capacity power plants).

The issue of a large amount of noxious substances emitted by conventional power plants to the atmosphere and extensive plaster dumps formation led to research on solutions that would prevent these phenomena. Furthermore, there is a demand for increased thermodynamic efficiency of a cycle, while reducing the costs of power generation. For that purpose the use of combined systems is considered that would involve a compression ignition engine and a steam turbine cycle. When placed on a platform, such a solution would provide for the generation of electric energy in the so-called distributed generation system.

Energy generated in a distributed system, also called distributed generation, involves the generation of energy by small units or production plants directly connected to distribution networks or located in the consumer's power system. They usually generate electric energy using renewable or non-conventional energy sources, often combined with heat generation. One of the basic classifications of distributed generation sources is classification in view of the generated power value. We can differentiate:

- small distributed generation (units of capacity between 1 kW and 5 MW);
- medium-sized distributed generation (units of capacity between 5 MW and 50 MW);
- large distributed generation (units of capacity between 50 MW and 150 MW);

The aim of this work is to conduct a technical and economic analysis involving the analysis of investment and operational costs of a power plant. This would enable the determination of the costs of the power plant generating a unit of power depending on the operation of the power unit. Furthermore, the benefits derived from using a combined system and the advantages in an offshore floating power plant in view of environmental friendliness of the plant will be analysed.

2. Selected issues of economic analysis

Running a business activity becomes more and more complicated and troublesome, which is why it should be based on a complex economic analysis. Various definitions of an economic analysis can be found in economy publications. An economic analysis is defined as activities related with the evaluation of a business entity operation. It involves the division of economic processes and phenomena into their components, specification of cause and effect dependences between the analysed components and formulation of general conclusions drawn from comparative assessments[2].

Each entity running a business activity performs periodic analyses aimed at verifying whether the goals that were set for it have been met. Consequently, the entity may prepare new, more detailed development plans. An economic analysis is the basic tool used for the verification of an entity's performance. The division of an economic analysis into areas is presented in Figure 1[7].

Fig. 1. Types of economic analysis
A technical and economic analysis deals with an accurate representation of an economic situation of an entity using tracer analysis. Various areas of an entity’s activity may be the object of analysis, including material performance (products, semi-finished products, services), cost of production or services rendition, human resources management, fixed assets management, technical progress and innovativeness, goodwill, and most of all – property and financial situation.

A financial analysis deals with processing financial performance of an entity on the basis of data included in the income statement and the cash flow statement, as well as other financial statements prepared by the entity[4].

While performing calculations related with the evaluation of an investment project, the following elements must be considered:
- Investment completion time
- Investment operation time.

The investment completion time is the time in which the investor incurs expenditures for the investment launch. The investment completion time may relate only to the baseline period, i.e. the first year, or the investment may be spread over a period of time, e.g. platform construction, engine installation, steam turbine installation. In such a case investment expenditures should be divided into the periods in which they are incurred.

The investment operation time is the time in which the investor benefits from the investment. Usually, an investment starts generating profits after 1 year, unless the investment completion time is spread over several periods, and the investment starts generating profits after several of the periods are completed.

When performing a technical and economic analysis, the costs structure must be considered:
- investment costs;
- operational costs;
- depreciation;
- interest (financial costs)

Operational costs are all those costs that the project generates after the investment is launched.

Depreciation is a generic type of cost that is not included in operational costs, as it is not a type of expenditure.

If loans (foreign capital) are taken out, then the item called interest, i.e. financial costs, appears.

### 1. Project economic efficiency assessment

The following indexes are used for the economic efficiency assessment of an investment.

1. Net Present Value (NPV) which for the time of the system utilization of N years from the moment the investment is put into operation results from cash flows added to each other that are planned in the subsequent years of operation (including the year zero).

\[
NPV = \sum_{t=0}^{N} \frac{CF_t}{(1 + r)^t}
\]

(1)

where: \( t \) - current year of operation, \( N \) - total number of years of operation, \( CF_t \) - cash flow in a given year \( t \), \( r \) - discount rate

For a technical solution that aims at an economic optimum, the NPV is maximum. This in turn gives the following function of an objective:
\[ NPV \rightarrow \text{max} \] (2)

2. Net Present Value Ratio (NPVR) involves the ratio of the project net value to the investment expenditures \(J_0\) necessary in order to reach the NPV:

\[ NPVR = \frac{NPV}{J_0} \] (3)

NPVR is an auxiliary index that enables the selection of an investment variant when comparing projects that are similar in terms of structure, investment expenditures, operation time, etc.

3. Internal Rate of Return (IRR) specifies the discount rate at which the Net Present Value calculated for the entire period of operation equals zero. An investment is profitable only when the Internal Rate of Return is greater than discount rate \(r\).

4. Simple Pay Back Period (SPBP) and Discounted Pay Back Period (DPBP) specify the minimum number of years for which the sum of actual cash flows and discounted cash flows for the year in which the investment is put into operation equals zero:

\[ \sum_{t=0}^{SPBP} CF_t = \sum_{t=1}^{SPBP} CF_t - J_0 \] (4)

\[ \sum_{t=0}^{DBP} CF_t \left(\frac{1}{1+r}\right)^t = \sum_{t=1}^{DBP} CF_t \left(\frac{1}{1+r}\right)^t - J_0 \] (5)

Using the definition of pay back periods, this article also introduces the notions of simple and discounted investment value in the subsequent years of its operation. These values are obtained by summing up CF cash flows from the year 0 to the analysed year \(N\).

As results from dependences from (1) to (5), the basic element of an investment economic efficiency assessment are Net Cash Flows. For the entire period of operation of the analysed power plant net cash flows were calculated using the following formula:

\[ CF = -J_0 + J_k + S_n - K - P_d + A - R + L \] (6)

where:

- \(J_0\) - total incurred investment expenditure \((J_0 = J_w + J_b)\),
- \(J_w\) - part of investment expenditures financed using the entity’s own funds,
- \(J_b\) - part of investment expenditures financed using bank loans,
- \(K\) - production costs (including depreciation and bank interest),
- \(P_d\) - income tax,
- \(A\) - depreciation of fixed assets,
- \(R\) - loan instalment,
- \(L\) - investment object liquidation value.

4. Calculations and results

The calculations of a floating power plant construction involved the comparison of two low-speed combustion engines – Wartsila 9RTA96C and MAN Diesel & Turbo 9K98MC-C7.1-TII – for the load of 90% CMCR (Contract Maximum Continuous Rating), placed on an offshore platform.
The calculations were based on the following assumptions:

- Electric energy generation will take place in a combined system including a compression ignition engine and a turbine, as well as a power grid connection.
- Operation time of the plant amounts to 15 years.
- Discount rate amounts to 8%.
- It was assumed that the investment will be financed in full using the investor’s resources.
- The calculations do not include prices nor costs increase factors.
- Exchange rates according to the National Bank of Poland as of 18 March 2014 – USD 1 = PLN 3.043 / USD, EUR 1 = PLN 4.2295 / EUR.
- Operation of the power unit amounts to 6500 h per year.
- Heavy fuel, density of 890 – 960 kg/m³
- The floating power plant will be placed on a mobile platform weighing approximately 200 t.

In accordance with the obtained information, the estimated cost of 1 kg of the structure amounts to EUR 7. Consequently, the cost related with the platform purchase (KP) will amount to 200 000 kg * EUR 7 / kg * PLN 4.2295 / EUR = PLN 5921300. When calculating a compression ignition engine price it was assumed that 1 kW of power costs USD 200.

Therefore the cost of the Wartsila engine:

\[ \text{CEW} - \text{Wartsila compression ignition engine cost (Cost Engine Wartsila)} \]
\[ \text{CEW} = 46332 \text{ kW} \times \text{USD 200} / \text{kW} \times \text{PLN 3.043} / \text{USD} = \text{PLN 28197655.2} \]

MAN engine cost:

\[ \text{CEM} - \text{MAN compression ignition engine cost (Cost Engine MAN)} \]
\[ \text{CEM} = 48762 \text{ kW} \times \text{USD 200} / \text{kW} \times \text{PLN 3.043} / \text{USD} = \text{PLN 29676553.2} \]

Steam turbine (TP) cost:

\[ \text{TM 1000 – Makila TITurbomeca, applied power: 3 897 kW, cost: PLN 6052000} \]
The team operating the floating marine power plant includes 10 people (WZ)

Team remuneration: 10 people * average daily rate of USD 200 = USD 2000 / day; Yearly cost of the power plant team remuneration will amount to PLN 2221390

Cable connection to the grid (PK):

\[ \text{PLN 57.55 for 1 kW; connection to the grid at the distance of 200 m} \]

Cost of connecting the Wartsila engine system to the grid:

\[ \text{PLN 57.55 / kW} \times 46332 \text{ kW} = \text{PLN 2666406.6} \]

Cost of connecting the MAN engine system to the grid:

\[ \text{PLN 57.55 / kW} \times 48762 \text{ kW} = \text{PLN 2806253.1} \]

Price of 1 m³ of heavy fuel is approximately PLN 2200 (KFj)

\[ \text{Wartsila} \]
\[ \text{m}_{\text{FD}} = \text{be} \times \text{Nd} \]
\[ m_f = 166.8 \text{ g/kWh} \times 46332 \text{ kW} = 7728177.6 \text{ g/h} \]
\[ m_{fD} = 7.728 \text{ t/h} \]

**Man**
\[ m_{fD} = 174.9 \times 48762 = 852847.8 \text{ g/h} \]
\[ m_f = 8.528 \text{ t/h} \]

Let us assume that \( 1 \text{ m}^3 = 920 \text{ kg/m}^3 \), i.e. \( 0.92 \text{ t/m}^3 \)

The Wartsila engine uses 7.728 t/h of heavy fuel, i.e. it requires 8.4 m\(^3\)/h, so the fuel cost amounts to \( 8.4 \text{ m}^3/\text{h} \times 6500 \text{ h} \times \text{PLN} \ 2200/\text{m}^3 = \text{PLN} \ 120120000 \) (KF)

The MAN engine uses 8.528 t/h, i.e. it requires 9.269 m\(^3\)/h, so the fuel cost amounts to 9.269 m\(^3\)/h \times 6500 h \times \text{PLN} \ 2200/\text{m}^3 = \text{PLN} \ 132546700 \) (KF)

The performed calculations enabled the determination of total investment expenditures as the sum of costs related with the purchase of a floating platform, a compression ignition engine and a steam turbine, and the construction of a cable connection to the grid, which is expressed as:

\[ J_0 = CE + TP + KP + PK \] (7)

and operational costs:

\[ K = KF + WZ \] (8)

Using dependences (7) and (8) Table 2 was compiled that contains power units calculation results for the two engines, depending on their power capacity and fuel consumption:

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>WARTSILA</td>
<td>MAN DIESEL &amp; TURBO</td>
</tr>
<tr>
<td>( N_D )</td>
<td>kW</td>
<td>46332</td>
<td>48762</td>
</tr>
<tr>
<td>CE</td>
<td>PLN</td>
<td>28197655.2</td>
<td>29676553.2</td>
</tr>
<tr>
<td>PK</td>
<td>PLN</td>
<td>2666406.6</td>
<td>2806253.1</td>
</tr>
<tr>
<td>( J_0 )</td>
<td>PLN</td>
<td>42837361.8</td>
<td>44456106.3</td>
</tr>
<tr>
<td>KF</td>
<td>PLN</td>
<td>12012000</td>
<td>132546700</td>
</tr>
<tr>
<td>K</td>
<td>PLN</td>
<td>122341390</td>
<td>134768090</td>
</tr>
</tbody>
</table>

The calculation results compiled in Table 2 show the dependence of investment costs on the compression ignition engines power capacity and the operational costs on the unitary wear of the main engine. That is why the selection of proper power units operation parameters during the technical and economic analysis is of such importance.

On the basis of the assumptions and calculations included in part four of the article it was possible to determine the annual generation of electric energy (\( E_{el} \)):

\[ E_{el} = N_D \times \text{engine operation time (t)} \] (9)

Using dependence (9) the level of power generation by the analysed engines was calculated:

**Wartsila**

\[ E_{el} = 46332 \text{ kW} \times 6500 \text{ h} = 301158000 \text{ kWh} = 301158 \text{ MWh} \] (10)
On the basis of results listed in Table 2 and the level of power generation by the individual power units, the estimated cost of generating 1 MWh of electric energy was calculated ($KPE_{el}$): 

$$KPE_{el} = \frac{K}{E_{el}} \text{ PLN/MWh}$$  \hspace{1cm} (12)

Wartsila

$$KPE_{el} = \frac{122341390}{301158} = 406.24 \text{ PLN/MWh}$$  \hspace{1cm} (13)

MAN

$$KPE_{el} = \frac{134768090}{316933} = 425.23 \text{ PLN/MWh}$$  \hspace{1cm} (14)

The obtained costs of electric energy generation by the analysed power units (13), (14) show the direction of future optimization works related with the technical and economic analysis. The possibility of increasing the efficiency of power units and the engines operation time should be considered, as it may result in a reduction of operational costs that have a direct influence on the cost of generating 1 MWh of electric energy. The purchasing price of 1 m$^3$ of heavy fuel used for the calculations is based on wholesale prices offered by market wholesalers. Preferential conditions for a floating power plant were not analysed. This article presents general information about the practical application of technical and economic analysis tools when evaluating investment projects.

3. Summary

The method of presenting information in the article was to describe the issues of an economic analysis of electric energy generation in a floating power plant in a clear and complete manner.

The proposed concept of power generation in floating power plants has the following advantages:

- increased generation of power in the north of Poland,
- diversification of primary energy sources that reduces coal consumption and increases liquid fuels consumption,
- possibility of residual heavy fuels combustion in the engine,
- reduction of the amount of coal transported from the south of Poland or imported on ships,
- lack of slag and ash,
- reduced CO$_2$, NO$_x$ emissions as a result of the increase in the system efficiency and the reduced emission resulting from the engine structure. What is more, the reduction of SO$_x$ emissions due to the application of sulphur-recovery systems,
- shorter construction time, when compared to a conventional power plant and the possibility of gradual launching. First: the compression ignition engine is put into
operation, then, during its operation, a combined system with a steam turbine is constructed,

- lack of complications related with water cooling the condenser, minor impact on the environment related with water management,
- mobile possibilities – offshore platform

In order to verify the profitability of the construction of a floating power plant it is necessary to perform a complex analysis related with cash flows involving total net sold generation value.

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LABORATORY STAND FOR INVESTIGATIONS OF THE QUICK-CHANGING TEMPERATURE OF GAS FLOWING IN THE PISTON COMPRESSOR OUTLET CHANNEL

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Abstract
This paper presents a laboratory stand, created as a part of work on the doctor dissertation, intended for investigation of heat and flow processes taking place in the Espholin H3S piston compressor outlet channel. The test stand elements and their configuration within the stand are described. Discussed are also problems connected with the quick-changing measurement technique, solutions which have been proposed so far in that technology and premises are determined of choosing most suitable methods for measuring quick-changing temperature of outlet gases in piston machines as a diagnostic parameter.

Key words: piston compressor, heat and flow processes, gas temperature measurement, diagnostics.

1. Introduction

Planned investigations of heat and flow processes in the outlet channel of an air compressor are directed at diagnostic problems. Their main purpose is developing a methodology of evaluation of the technical state of elements bounding the working spaces of a compressor, based on the state parameters of thermodynamic medium observed in an outlet channel with adequate measurement instrumentation. For realization of a so formulated research objective it is necessary to build and to outfit a laboratory test bed allowing experimental tests to be carried out in the conditions as close as possible to the operating conditions of the real object. Measurements of the quick-changing pressures and temperatures of outlet gases [1, 4, 5], which were carried out on engines and piston compressors, allow to isolate several important metrological problems, which are to be taken into account in planning experimental tests both on a laboratory stand and in real object diagnostics. The most important of them are considerable inertia of the temperature sensors and the influence of external conditions on the obtained measurement result. However, proper selection of adequate measurement methods and instrumentation allows to minimize most of the shortcomings connected with measurements of quick-changing temperatures.
2. Diagnostics of piston compressors

Piston compressors are widely used aboard ships, first of all for compression of the following gases [6]:
- air, mainly for:
  - starting main and auxiliary engines,
  - driving the mechanism of propeller pitch change,
  - driving the main engine disengaging couplings and reversing mechanisms,
  - scavenge of kingston valves, pipelines and other water spaces,
  - scavenge of fire extinguishing installations,
  - ship horn (tyfon),
  - fuel atomization in boiler furnaces,
  - starting the turbine combustion engines – the fuel atomization, ignition and combustion process assist,
  - working medium in the pneumatic control systems,
  - household applications (e.g. hydrophores).
- refrigerating media (in refrigerating and air conditioning installations).

Therefore, a very important operational feature is diagnosis of that equipment without interference into its internal structure. One of the essential methods is parametric diagnosing based on the measurements and comparative analysis of static temperature at the end of gas compression. In the case of stabilized processes, that parameter allows to obtain such information, among others, as:
- indicating places where the greatest energy losses occur,
- defining the type of energy process disturbances in a compressor,
- identification and location of defects of compressor working space structural subassemblies and elements.

During compressor operation it is possible to watch many diagnostic symptoms indicating disturbances, e.g. decreased output and rotational speed, knocks in the mechanical system, smoke, too low lubricating oil pressure at the oil pump outlet [2]. Some of the mentioned compressor disfunctions may cause also an increase of temperature at the end of the compression process.

The most often occurring causes of compressor output decrease are the following:
- fouled air filter, which causes decreased pressure of the air suction (and in addition increased end temperature of compression) as well as reduced flow throuput in the pumping valve due to difficult to remove deposits of charred oil (as an additional consequence of the increased end temperature of compression),
- increased temperature at the end of air compression, considerably higher than the calculation value (due to lowered cooling effectiveness and too high cooling water temperature, caused e.g. by deposits of limestone on the cooling space walls).

In view of frequent occurrences of various nonoperational conditions in the use of compressors, it is necessary to develop diagnostic methods allowing to detect them early and to remove their initial causes [3]. It will make possible to apply a technical service planning strategy in accordance with the equipment actual technical condition and therefore will increase durability and reliability with considerable reduction of the service labour consumption.

In developing an optimum diagnostic method, proper methodical questions should be formulated regarding the way the diagnostic tests should be carried out:
1. What parameters and where to measure them?
2. In what way and when to be measured?
3. What equipment and measurement technologies to be applied?
4. How to draw the conclusions?
5. Which of the selected diagnostic parameters brings most diagnostic information and simultaneously is easy to measure?

Then, by means of proper methods of diagnosing (a.o. experimental tests with the use of diagnosing systems and numerical simulation of heat flow processes), the questions should be answered in such a way that by interpretation of the obtained diagnostic test results relations can be found between the received values of diagnostic parameters (symptoms) and known recognizable inaptitude states most often occurring in a piston compressor operation process.

3. Object of investigation

In order to obtain diagnostic information about the technical condition of structural elements confining the working spaces of an air compressor, a laboratory stand has been set up including the following subassemblies: an electric motor driven Espholin H3S type air compressor, an air tank and a piping system. The whole installation is mounted on a shock-absorbing foundation. Additionally, standard measurement instruments: manometer, thermometer and thermocouple are installed and also safety-valves protecting the compressed air-filled pressure system against exceeding the admissible pressure, which increases safety of the stand operation. This test stand will provide facilities for verification of the developed mathematical models of heat-flow processes in the whole compressor system, with a possibility of using them also in the combustion engine diagnostics.

Main subassembly in the stand is a two-stage Espholin H3S piston air compressor with interstage cooler, of the following dimensions:
- stage I – piston diameter D=95 mm, piston stroke S=80.1 mm (±0.1);
- stage II – piston diameter D=50 mm, piston stroke S=80.1 mm (±0.1).

Main operating parameters of the compressor are shown in Table 1.

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotational speed</td>
<td>750 min⁻¹</td>
</tr>
<tr>
<td>Volumetric output</td>
<td>425 dm³/min</td>
</tr>
<tr>
<td>Effective output</td>
<td>305 dm³/min</td>
</tr>
<tr>
<td>Power</td>
<td>5.5 kW</td>
</tr>
<tr>
<td>Internal thread on the output side</td>
<td>0.5”</td>
</tr>
</tbody>
</table>

The Espholin H3S compressor is driven (through belt transmission) by an electric motor of 5.0 kW power at 1460 min⁻¹ rotational speed and 6.0 kW power at 1755 min⁻¹ speed. Fig. 1 shows general view of the Espholin H3S piston compressor assembly during initial stage of building the test stand.
In the discussed compressor system a pressure tank (with 15 bar working pressure) of 120 dm$^3$ capacity is mounted as a compressed air accumulator. Fig. 2 presents the pressure tank valve head with the inlet and outlet stub pipes. Fig. 3 and 4 present general view and schematic diagram of the test stand.

![Pressure tank valve head](image)

**Fig. 1. General view of the Expholin H3S piston compressor assembly before mounting on the foundation**

Air temperature measurements in the test stand compressor outlet channel are carried out with a K type T-201p-K-0,5-20-1,5-1-M6-1-S-6-200 thermocouple, its technical parameters given in Table 2.

**Table 2. Technical parameters of the temperature gauge**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature gauge</td>
<td>K type, class 1</td>
</tr>
<tr>
<td>Gauge diameter</td>
<td>d=0.5 mm</td>
</tr>
<tr>
<td>Gauge length</td>
<td>L$_c$=20 mm</td>
</tr>
<tr>
<td>Casing material</td>
<td>Inconel 600 (72% nickel and 14 - 17% chromium alloy)</td>
</tr>
<tr>
<td>Conductor length</td>
<td>L$_p$=1.5 $\cdot$ 10$^3$ mm</td>
</tr>
<tr>
<td>Conductor insulation</td>
<td>glass fibre x 2/ braided screen</td>
</tr>
<tr>
<td>Process terminal type</td>
<td>M6 thread</td>
</tr>
<tr>
<td>Execution</td>
<td>small spring, penetration weld</td>
</tr>
<tr>
<td>Work temperature</td>
<td>up to 200$^\circ$C</td>
</tr>
</tbody>
</table>
The test stand (Fig. 3 and 4) will be used for performing the following research tasks:
- investigation of the heat-flow processes in a piston type air compressor,
- the stand can be easily adapted, depending on the needs, for application of different measurement techniques and tools (replaceable measurement section),
- possibility to introduce, into the compressor and the connected network, virtual (simulated) operational inaptitude states (e.g. simulation of the piston ring wear by "undercuttings", flow throttling with an orifice),
- didactic purposes.

4. Measurement problems

Many methodological problems may be encountered during gas quick-changing temperature measurements from the side of temperature gauge or measurement signal converter and also from disturbances caused by the measurement stand environment. In order to develop an effective method of impulse-diagnosing of a heat-flow system, method based on
temperature measurements of gas flowing with high velocity, the influence of measurement disturbing factors should be analysed and ways of minimizing their impact should be worked out.

One of the basic disturbances significantly influencing the accuracy of gas temperature measurement results is inertia of the applied gauge. The most often used in this kind of measurements are K type thermocouples (chromel-alumel), manufactured in broad range of the measuring element lengths and diameters and also in versions with or without casing. In order to reduce the measurement inertia to a minimum, the thinnest possible measuring element should be used, but due attention has to be paid to its less durability and resistance to defects (these features appear additional problems in this measurement method) with decreasing diameter. Gauge without the measuring element casing is much less durable, but also has less inertia. Thermocouple is a solution matching both aspects, its weld penetrates the casing material (Fig.5), which reduces the measurement inertia and maintains considerably greater durability. An effective solution is to use a thermocouple with its measurement terminal sticking 1-1.5 mm out of casing.

![Fig. 5. Schematic view of cased thermoelement](image)

The determining parameter of thermocouple inertia is its time-constant, dependent on the thermocouple mechanical structure and size, thermophysical properties of the material used as well as thermodynamic aspects and character of the gas flow. Figure 6 presents graphical interpretation of the thermocouple time-constant. In spite of the time-constant value being determined by measurement element manufacturers, calibration of gauges appears necessary. The most up-to-date method of calibration is the constant-point method, based on the use of phase transitions (melting, solidification, triple point) of fine metals and water. Calibrated temperature gauges are placed in heated or cooled cells filled with fine metals. Thanks to a precise control system, the created phase transition state can be kept for several or some scores of hours, maintaining constant material temperature. Calibration by the constant-point method may be carried out in:

- Mercury triple point, Hg (-38.8344 °C),
- Water triple point, H₂O (0.01 °C),
- Galium melting point, Ga (29.7646 °C),
- Zinc solidification point, Zn (419.527 °C),
- Aluminium solidification point, Al (660.323 °C),
- Copper solidification point, Cu (1084.62 °C),
- Palladium melting point, Pd (1553.5 °C) by wire method.
An important factor influencing the accuracy of quick-changing temperature measurements is the way of gauge mounting. Main obstacle is thermocouple heating from the material of heat channel (or other supporting element) where it is screwed or soldered in during the measurement. This has a decisive impact on the measurement results. The only method of preventing or significantly reducing such disturbance is efficient insulation of the gauge, e.g. by applying a water-cooled shield (Fig. 7). However, this is connected with possible leakage and water penetration to the flow channel.

5. Existing measurement conceptions

Combination of the two-way correction method and the recovery coefficient determination method is presented as a solution of the problem of measurement of time-dependent gas temperature in a broad input function frequency range. In effect, a dynamic-error-free temperature signal is obtained, resolved into components corresponding to the static temperature (resulting from the medium thermodynamic parameter changes) and dynamic temperature (connected with the gas flow velocity) [5].

Essence of the method proposed by the Author of the above quoted publication is using two transducers of different (unknown in advance) dynamics, measuring the same input signal. The Author proposes a special design Prandtl probe to be used. Apart from the classic measurement of total and static pressure, that probe ensures gas static temperature measurement (no dynamic component of temperature has an influence on the thermocouple indications). The measurement is carried out by means of a miniature thermocouple placed at...
the point corresponding to the static pressure measurement (on the probe gauging tip perimeter) and thermally insulated from the probe [5].

Other Authors [1] developed a solution based also on quick-changing temperature measurements by means of two thermoelements of different diameters, made from a material with good conductivity and low thermal inertia (proposed by H. Pfriem in 1936). The difference in comparison with the above described method consists mainly in building a test stand with specially prepared disc, in which were placed two thermocouples of different diameters and an anemometer to perform measurements of gas parameters in identical instantaneous conditions.

6. Evaluation of the proposed methods

Aspiring to setting up an own test stand, and in consequence to developing a measurement method of flowing gas quick-changing temperature, one should evaluate the existing conceptions, taking into account several aspects:

1. technical capability of carrying out the measurement (adapting an existing measurement system to the required measurement conditions),
2. appropriateness of the stand for planned research (type and method of performed measurements),
3. costs of the measurement instrumentation.

After analysis of the above mentioned aspects, a most effective method should be proposed for realization of the planned programme of investigations to be performed on the built laboratory stand, the method allowing to minimize most of difficulties occurring during quick-changing temperature measurements, with simultaneous simplicity and integrity of the stand structure and relatively low investment costs. The proposed solution should also be applicable to temperature measurements on other similar stands (e.g. combustion engines), and particularly to carrying out diagnostic investigations on real objects.

7. Summary

The thermodynamic medium temperature may provide diagnostic information on the technical state of structural elements bounding the piston machine (mainly compressors, but also combustion engines) working spaces. This is connected with necessary measurements of the flowing gas quick-changing temperatures and their conception is an important subject of the contemporary metrology (thermometry). In order to develop an innovative, but at the same time based on the existing solutions, method of measuring that diagnostic parameter, a “golden mean” must be proposed meeting the requirements of high accuracy of measurements (which determines the reliability of diagnosing), and relative simplicity and economy of the method. Therefore, the most suitable solution for temperature measurements of quick-changing heat-flow processes, taking place in a piston compressor outlet channel, seems to be a method based on measurement of quick-changing temperatures by means of two thermoelements with different diameters and with water-cooled insulation screen.

References:


ANALYSIS OF PARAMETERS VARIABILITY OF COMPRESSION RING AND CYLINDER LINER COLLABORATION DURING ENGINE OPERATION

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Abstract

Construction of piston-rings-liner assembly of contemporary engines of low and medium power secures a long time reliable operation. In the case of engines of higher output, e.g. marine and railway ones still occur failures caused by improper collaboration of piston rings and cylinder liner. One can mention variations in collaboration surface and difficulties with supply and proper distribution of lubricating oil over the entire surface of cylinder liner among the most important ones.

Changes in value of parameters characteristic for compression ring and cylinder liner collaboration that occur during engine run have been analyzed in the following study. Attention has been paid to the ability of ring contact with the liner, especially those features that cause the formation of light slots between ring face and liner surface.

Drafts presented in the paper show the results of compression ring pressure distribution and the geometry of light slots carried out for technical data of the earth moving machine engine. Simulation computations take into account both the effect of engine operation cycle phase and selected stages of its run as well.

Keywords: IC engine, liner deformation, piston ring, ring elastic pressure

1. Introduction

During engine operation the compression ring installed in groove is pressed against the wall by radial forces resulting from ring own elasticity and gas force that acts on ring’s inner face. When the value of elastic pressure remains approximately constant, the ring wall pressure caused by gas pressure changes itself cyclically. Generally, this phenomenon should be regarded as advantageous one because it leads to the better ring sealing properties during those strokes when the gas pressure in combustion chamber is highest one, i.e. strokes of compression and expansion. This phenomenon gives better ring fit to the deformed cylinder liner preventing or limiting the gas blow-by. However, it should be noted that the ring high wall pressure could negatively affect the engine durability. High pressure could lead to high stress in ring material (which can cause the ring break) and bring about an oil film rupture (ring collaborates with liner under conditions of boundary or even no lubrication).
These problems were analyzed by the Authors in their earlier studies, concerning above all on the analysis of ring wall pressure distribution (using the previous papers concerning this problem as an example, e.g. [1,3]). The mentioned investigations concerned rather simple problems starting from the case of merely ring pressure against ideally circular wall [5] and next - the deformed wall [6,7]. Further the scope of investigations has been widened and besides the ring own elasticity it encompassed the effect of gas forces as well [8]. In present paper the range of investigations has been further broadened including collaboration of compression ring with the worn and deformed cylinder wall taking into considerations also the gas forces.

It should be stressed that the effect of oil layer present on the wall was omitted in this study. Such assumption is necessary for the applied method of computations which allows the determination of certain quantities characteristic for ring and wall collaboration, including forces crucial for full contact as well as location and size of light slots formed on cylinder wall. Taking the oil layer into consideration investigations possible in future will cause the verification of some conclusions, especially those concerning location and shape of slots.

2. Ring pressure against the deformed cylinder wall

It has been proved in [8] that the ring circumferential pressure distribution $p_m(\varphi)$ against the worn and deformed cylinder wall can be determined using the following formula, (symbols as in Fig. 1):

$$p_m(\varphi) = \frac{E \cdot I}{h_p \cdot r_m} \left[ K_z(\alpha) \cdot r_m - \left( z_a + z_b(\varphi) + 2 \cdot z_b''(\varphi) + z_b''''(\varphi) \right) \right],$$  

(1)

where:

- $E$ – Young’s modulus,
- $I$ – ring cross-section moment of inertia,
- $r_m$ – radius of ring in cylinder neutral layer (see Fig. 1a),
- $h_p$ – ring axial height,
- $z_a$ – cylinder constant deformation,
- $z_b(\varphi)$ – cylinder wall deformation, variable along the cylinder circumference, given by the $\varphi$ angle (and its second and fourth derivative).

Quantity $K_z(\alpha)$, later called a ring characteristic dynamic coefficient, can be described with the following formula [4]

$$K_z(\alpha) = K + \frac{p_g(\alpha) \cdot h_p \cdot r_m^3}{E \cdot I},$$  

(2)

where:

- $p_g(\alpha)$ – pressure that presses on the ring to cylinder wall (its value depends on pressure over and below the ring); this pressure can be expressed as the function of crank angle $\alpha$,
- $K$ – a ring characteristic constructional coefficient given by

$$K = \frac{p_m \cdot h_p \cdot r_m^3}{E \cdot I}. $$  

(3)

Values of constructional and dynamic ring characteristic coefficients are equal, i.e. $K_z(\alpha) = K$ when there is no gas pressure and the ring wall pressure $p_m(\varphi) = p_m$ results exclusively from ring own elasticity and does not depend on crank angle

a)  

b)  

c)
The deformed cylinder (or more precisely – course of its circumferential line) has been expressed as a sum of Fourier series harmonics

\[ z_b(\varphi) = \sum_{h=1}^{n} A_h \cos(h\varphi + \delta_h), \]

where the quantities of \( A_h \) and \( \delta_h \) are amplitude and phase shift of the series successive harmonics \( h \), respectively. It should be mentioned that the presented way of description of cylinder wall geometry assumes the invariable deformation along the entire cylinder height, which does not take into account the real shape of the profile (measurements on worn cylinders show a significant wear in the area of ring TDC).

Besides the cylinder deformation itself an even wear of its surface \( z_a \) occurs during long-time engine run, which also affects the ring circumferential wall pressure (Eq. 1).

Using dependencies (1) and (4), the following equation has been obtained after necessary transformations:

\[ p_n(\varphi) = \frac{E \cdot I}{h_p \cdot r_m} \left[ K_\alpha(\alpha) \cdot r_m - \left( z_a + \sum_{h=1}^{n} (h^2 - 1) A_h \cos(h \cdot \varphi + \delta_h) \right) \right]. \]

Analysis of circumferential wall pressure distribution allows a definition of excessive pressure areas on cylinder wall as well as those where pressure is equal to zero and light slots can occur. A trial of definition of such areas extreme points by the angle \( \varphi_0 \) defining position of points on cylinder circumferential line (as in Fig. 1c) was carried out in [8]. For a mathematic description of ring circumferential line with just one harmonic of Fourier series an angle lowest value corresponding to the beginning of slot is given by

\[ \varphi_0(\alpha) = \arccos \left( \frac{K_\alpha(\alpha) \cdot r_m - z_a}{A_h(h^2 - 1) \times \delta_h} \right), \]

For more complex cylinder deformations (when a higher number of harmonics is needed for description of circumferential line) such a trial leads to more complicated dependencies and definition of demanded angles would be extremely difficult. A possible solution that leads to an approximate result is an expansion of a cosine function of Eq. 6 in a power series, according to [2]:

\[ \cos x = 1 - \frac{1}{2!} x^2 + ... + \frac{(-1)^n}{(2 \cdot n)!} x^{2n}, \quad |x| < \infty. \]
Using only initial elements of the series (and assuming that \( \delta_h = 0 \)) a \( p_m \) pressure has been defined as

\[
p_m(\varphi) = \frac{E \cdot I}{h_p \cdot r_m^4} \left[ K_z(\alpha) \cdot r_m - \left( z_a + \sum_{h=1}^{n} (h^2 - 1)^2 A_h - 0.5 \varphi^2 \sum_{h=1}^{n} h^2 (h^2 - 1)^2 A_h \right) \right],
\]

(8)
as well as the minimum value of \( \varphi \) angle corresponding to the beginning of slot on cylinder circumferential line

\[
\varphi_o(\alpha) = \sqrt{\frac{2 \sum_{h=1}^{n} (h^2 - 1)^2 A_h - K_z(\alpha) \cdot r_m + z_a}{\sum_{h=1}^{n} h^2 (h^2 - 1)^2 A_h}}.
\]

(9)

Estimation of the relative difference between boundary angles calculated according to (6) and (9) for geometry of typical cylinder shows that it remains on the level of 5%.

Application of presented equation is limited only to the cases for which ring does not contact with liner surface with its entire circumference (or at least touches it without any pressure). This corresponds to the situation when the numerator of Eq. 9 has positive value. For a simple case when the cylinder deformation is described by a single harmonic the amplitude of deformation is given as

\[
A_h = \frac{K_z(\alpha) \cdot r_m - z_a}{(h^2 - 1)^2}.
\]

(10)

3. Determination of compression ring against the deformed cylinder liner – computational example

A presented further example concerns a trial of computational determination of compression ring wall pressure distribution against cylinder surface of increasing wear along with engine mileage.

Technical data and course of wear assumed in presented example concerns an engine of earth moving machine (International Harvester DT 466). Most important data of compression ring are presented in Table 1.

<table>
<thead>
<tr>
<th>Tab. 1. Technical data of exemplary IC engine compression rings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quantity</td>
</tr>
<tr>
<td>Cylinder diameter ( d ) [m]</td>
</tr>
<tr>
<td>Ring neutral radius ( r_m ) [m]</td>
</tr>
<tr>
<td>Axial height ( h_p ) [m]</td>
</tr>
<tr>
<td>Radial thickness ( g_p ) [m]</td>
</tr>
<tr>
<td>Young modulus ( E ) [Pa]</td>
</tr>
<tr>
<td>Mean pressure ( p_a ) [MPa]</td>
</tr>
<tr>
<td>Tangential force ( F_t ) [N]</td>
</tr>
<tr>
<td>Stiffness ( EI ) [Nm²]</td>
</tr>
<tr>
<td>Parameter ( K ) [--]</td>
</tr>
</tbody>
</table>

The course of cylinder circumferential line varies itself along with engine mileage and therefore requires a higher number of harmonics. In order to ease the analysis of results obtained the period of engine run has been divided in four stages corresponding with suitable sets of data \( z_a \) and \( z_b \), (see Table 2).

<table>
<thead>
<tr>
<th>Tab. 2. Cases of cylinder deformation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case No</td>
</tr>
<tr>
<td>---------</td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>
Values of cylinder liner deformation and wear given in Table 2 were selected on the basis of measurements carried out on a real engine and the selection was aimed at achievement of unambiguous conclusions from the analyzed problem.

According to Eq. 5 the knowledge of the $K_z(\alpha)$ coefficient is needed for determination of ring wall circumferential pressure. Values of this coefficient (determined in [9]) corresponding to the engine full load operation (Fig. 2) are used in presented example.

<table>
<thead>
<tr>
<th>Case</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>(Fig. 4a) A new cylinder, not worn and not deformed</td>
</tr>
<tr>
<td>2</td>
<td>(Fig. 4b) 0 $A_4 = 10$ Liner installation; cylinder deformation described by a single harmonic of 4th order</td>
</tr>
<tr>
<td>3</td>
<td>(Fig. 4c) 50 $A_2 = 30, A_4 = 15$ Initial stage of operation; worn and deformed cylinder</td>
</tr>
<tr>
<td>4</td>
<td>(Fig. 4d) 200 $A_2 = 100, A_4 = 25, A_6 = 5$ Further stage of operation; worn and deformed cylinder</td>
</tr>
</tbody>
</table>

Cylinder surface profiles corresponding to the cases collected in Table 2 are shown in Fig. 3 (for better clarity the $z_a$ wear is not present). Assumption of phase shift makes that amplitudes of all harmonics taken into account sum up for $\phi = 90^\circ$ (0.5 rad) angle.

![Fig. 2. Course of $K_z(\alpha)$ coefficient value vs. $\alpha$ angle at engine full load](image)

![Fig. 3. The course of cylinder circumferential line (line 1) expressed as the sum of respectively: second (2), fourth (3) and sixth (4) harmonic of Fourier series determined for data collected in Table 2 (the $z_a$ wear was not taken into account)](image)

After determination of the cylinder circumferential line for each of analyzed cases corresponding to successive stages of engine operation the ring pressure distribution against the deformed cylinder wall was computed with particular attention paid to the areas where this pressure falls down to zero (Fig. 1c shows the way to identify this area).

As the result of carried out calculations the ring pressure distribution along the cylinder circumferential line was determined (using the angle increment $\Delta\phi = 2^\circ$) repeating calculations
for consecutive positions of ring between the dead centers for stroke of compression and expansion (with the angle increment $\Delta\alpha = 10^\circ$ CA). Charts presented in Fig. 4 cover the area of highest probability of light slot presence, i.e. the one for $\varphi$ angle between 45 and 135° ($0.25\pi$ and $0.75\pi$ rad, as indicated in Fig. 3c).

**Case 1**
An ideal cylinder

**Case 2**
Deformed cylinder:
$A_1 = 10 \mu m$

**Case 3**
Deformed cylinder:
$z_0 = 50 \mu m,$
$A_2 = 30 \mu m, A_4 = 15 \mu m$
4. Summary and conclusions

The analysis of pressure distribution for selected stages of engine operation shows that differentiation of ring pressure increases along with the increase of wear $z_a$ and deformation $z_b$ of cylinder surface. For a new cylinder this distribution is even and changing itself under influence of gas forces (Fig. 4, case 1) undergoes the substantial differentiation even for minor deformation of its surface. Already for the 4th order harmonic of only 10 $\mu$m amplitude (case 2) an area of zero ring pressure appears during compression stroke and gas blow-by can occur. This area disappears during expansion stroke thanks to the gas pressure far higher than the ring elastic pressure. Further deformation increase results in rise of slot circumferential area, simultaneously encompassing higher and higher cylinder surface (beginning of this process is gradually situated close to TDC). The pressure distribution and increase in slot area is particularly unfavorable influenced by such cylinder deformation which requires consideration of Fourier series higher harmonics for its mathematical description (Case 4). This conclusion has been confirmed by the results of supplementary investigations (see Fig. 5) concerning the ring pressure against the deformed wall described by a single harmonic of the same amplitude (5 $\mu$m) but higher order. The investigation prove that the increase in harmonic number substantially affects the stress gradient value and size of slot area. However, it can happen for a higher number of harmonics that the achieved resultant could be lower than the one obtained for a single harmonic because of a mutual shift among the harmonics of different order.

Carrying out the additional research (no results in this paper) showed that even high values of cylinder wear $z_a$ (of order of few hundred micrometers) affects the pressure distribution to far lower extent than the small deformations $z_b$ (of order of several micrometers).

[Fig. 4. Analyzed cases of ring wall pressure distribution]

[Fig. 5. Distribution of ring pressure against deformed wall described by a single harmonic of 5 $\mu$m amplitude obtained at compression stroke: (a) second order harmonic, (b) fourth order one, (c) sixth order one]

Generally, on the basis of detailed conclusions and observations one can conclude that:
ring elastic pressure provide its full contact with the cylinder wall only for minor deformations,
ring wall pressure depends above all on gas pressure in strokes of compression and expansion, particularly close to the TDC,
cylinder deformations that require description using harmonics of higher order cause higher gradients of ring wall pressure and favors the formation of light slots to higher degree than harmonics of lower order,
more the course of circumferential line is close to the circle, lower is the risk of slots formation and blow-by,
higher engine mileage affects its wear and deformations, and finally the increase of slots geometry and gas blow-by.

The presented investigations were carried out assuming the lack of oil layer on cylinder wall which should be stressed once again. Taking the oil film into consideration would result in complete or partial filling the slots with oil. This eventually will cause the change in pressure distribution and will lead to smaller area or even to complete disappearance of light slots.

References

OBSERVATION OF THE THERMOSIPHON EFFECT IN THE CIRCULATION OF ACETONE AS WORKING FLUID IN MODERN COOLING SYSTEM

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Abstract

Packaging and thermal management of electronic equipment that are located i.e. in a novel marine power plants or computer server rooms has led to the demand for new and reliable methods for electronic cooling. Because of bigger and bigger power levels and miniaturization of the electronic devices, lack of free space in marine power plant, typical cooling techniques such as conduction and forced convection are not able to cool such a high heat flux. The increasing integration of electronic systems requires an improved cooling technology that supposed to be designed for high thermal performance, low mass, and able to work in harsh environments.

In this paper presented a prototype of thermosiphon loop heat exchanger developed in Institute of Energy Gdansk University of Technology. This thermosyphon loop is heated from below horizontal side and cooled from upper horizontal side, the working fluid that circulate inside the loop is acetone.

Keywords: marine power plants, heat transfer, heat exchangers, termosiphons. Loop Heat Pipes

1. Introduction

The important function of a natural circulation loop (i.e. thermosyphon loop) is to transport heat from a heat source to a sink. Fluid flow in a closed thermosyphon loop is created by the natural forces that develop from the density gradients induced by temperature differences in the heating and cooling sections of the loop. An advanced thermosyphon loop consists of evaporator, where the working liquid boils; and the condenser where the vapour condenses back to liquid; and liquid and vapour lines connecting these two exchangers. Heat is transferred as the vaporization heat from the evaporator to the condenser. The thermosyphon is a passive heat transfer device, which makes use of gravity for returning the liquid to the evaporator. Thermosyphons are less expensive than other cooling devices because they feature no pump.

There are many engineering applications for thermosyphon loops such as, for example, solar water heaters, air heat recovery systems, thermosyphon reboilers, nuclear power plants, emergency cooling systems in nuclear reactor cores, electrical machine rotor cooling, gas turbine blade cooling, thermal diodes and electronic device cooling.
2. Literature review

A thermosiphon, gravity-assisted wickless heat pipe, has been used as a practical heat transfer device due to its simple structure Faghri 1995 [1] or Pioro 1997 [6]. The heat transfer between two ends of the thermosiphon occurs with negligible temperature difference because a thermosiphon utilizes liquid–vapor phase change phenomenon of fluid. The only disadvantage of thermosiphon in many application is its gravitational dependence and the limited operating temperature range that is specifically determined by the choice of working fluid. There has been a number of studies about thermosiphons. Khandekar et al. [2] studied the overall thermal resistance of closed two-phase thermosyphon using pure water and various water based nanofluids (of Al₂O₃, CuO and laponite clay) as working fluids. They observed that all these nanofluids show inferior thermal performance than pure water since nucleation sites were closed by the deposition of the nanoparticles. Lee et al [3] studied double-evaporator thermosiphon for cooling high temperature superconductor system using nitrogen as the working fluid under sub-atmospheric pressure condition. Lee et al [3] conducted that double-evaporator thermosiphon can be a useful heat transfer device that can simultaneously cool-down the thermal loads vertically separated under a tight space limitation.

3. Design and Experimental Setup

3.1. Loop Thermosyphon Design

The experimental thermosyphon loop was constructed on the basis on existing test up stand for testing the performance of evaporators filled with wick made up with mixture of sintered material powder. This experimental test up stand previously worked as Loop Heat Pipe and was described in literature by Mikielewicz et al. [4] and Mikielewicz and Szymaniński [5]. The only element that has been changed is evaporator – at this construction evaporator is empty cylindrical tube made by cooper, heated over the entire length. This setup stand was also set vertically, to utilise the gravity forces for circulating the working fluid.

3.2. Experimental Setup

The experimental setup has cylindrical evaporator. Evaporator heat source is a resistance wire wrapped around evaporator. The wire length is 1,3m and resistance of 32Ω . This wire was connected to a DC laboratory power supply with adjustable voltage and current level. Condenser is a typical shell and tube heat exchanger where the shell is made by cylindrical cupper tube of Ø254 mm diameter and stainless steel tube of Ø6,35 mm diameter. The condenser was cooled by tap water at a temperature of approx. 7,5°C and the mass flow was measured by a laboratory rotameter. The liquid and vapour lines are smooth stainless steel tubes of a length 5000mm and internal diameter Ø3,87mm. The whole system was insulated using polyethylene foam. Appearance of experimental set up presented at Figure 1 and details of termosyphon tested are given in Table 1.

The loop was equipped with 8 thermocouples “type T” and sensitivity of 40μV/°C. The thermocouples were connected to Pico Technology Thermocouple Data Logger Type USB TC-08. The thermocouples were located at the most important measuring points of the system that are: 1 - evaporator casing, 2 - vaporator outlet/vapour line inlet, 3 - vapour line outlet/condenser inlet, 4 - condenser outlet/liquid line inlet, 5 - liquid line outlet/evaporator inlet, 6 - condenser cooling water inlet, 7 - condenser cooling water outlet, 8 - ambient. The system was equipped with two pressure transducers connected to displays and powered by a laboratory DC power supply. The first pressure transducer measured the internal pressure
inside the system (Swagelok PTU-F-AG60-12AI), and the second one measured the
differential pressure between evaporator inlet and outlet. This differential pressure transmitter
was made by Regin company (DTK100-420). View and distribution of pressure transmitters
were shown in Figure 1.

Tab. 1. Details of thermosyphon

<table>
<thead>
<tr>
<th>Working fluid</th>
<th>Acetone</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature range</td>
<td>60°C- 80°C</td>
</tr>
<tr>
<td>Evaporator casing</td>
<td>Cupper</td>
</tr>
<tr>
<td>Heated part</td>
<td>210 [mm]</td>
</tr>
<tr>
<td>Diameter</td>
<td>Ø14 [mm]</td>
</tr>
<tr>
<td>Wall thickness</td>
<td>1 [mm]</td>
</tr>
<tr>
<td>Vapour line</td>
<td>Stainless steal</td>
</tr>
<tr>
<td>Internal diameter</td>
<td>Ø3.87 [mm]</td>
</tr>
<tr>
<td>Length</td>
<td>5000 [mm]</td>
</tr>
<tr>
<td>Wall thickness</td>
<td>1.24</td>
</tr>
<tr>
<td>Liquid line</td>
<td>Stainless steel</td>
</tr>
<tr>
<td>Internal diameter</td>
<td>Ø3.87</td>
</tr>
<tr>
<td>Length</td>
<td>4940.4</td>
</tr>
<tr>
<td>Wall thickness</td>
<td>1.24</td>
</tr>
<tr>
<td>Condenser</td>
<td>Cupper</td>
</tr>
<tr>
<td>Length</td>
<td>270 [mm]</td>
</tr>
<tr>
<td>Internal diameter</td>
<td>Ø3.87 [mm]</td>
</tr>
</tbody>
</table>

Fig 1. Scheme of experimental test-up stand
4. Experimental Result and Discussion

The purpose of preliminary test is to measure the maximal pressure maximum pressure difference obtained before and after the evaporator for acetone as the working fluid. Tests were performed for 6 heater power settings (30W, 40W, 50W, 60W, 70W and 80W) and the various flow rate of condenser cooling (30 l/h, 50 l/h, 100 l/h, 150l/h). The results are shown in a line graph (Figure 2) showing the dependence of the pressure difference obtained before and after the evaporator on heating power.

![Figure 2](image_url)

*Fig. 2 The dependence of the pressure difference obtained before and after the evaporator on heating power*

4. Conclusions

As shown in Figure 2, the best thermosiphon performance is achieved using applying to condenser cooling of flow rate of 150l/h. For the heater power of 30-80W the pressure rise before and after the evaporator achieved in a range from 1.08-1.28 [kPa]. It gives us a good predictions for future studies of pressure rises and heat transfer at the termosiphon loops. It also give us new hints to improve the design of thermosiphon loop.

References


MODELLING OF TOXIC COMPOUNDS EMISSION IN MARINE DIESEL ENGINE DURING TRANSIENT STATES AT VARIABLE PRESSURE OF FUEL INJECTION

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Abstract

Transient states are an important part of the spectrum of engine loads, especially the traction engines. In the case of marine diesel engines, transient states are of particular importance in reducing the analysis of motion units for special areas and maneuvering in port, the participation of transient states in the load spectrum significantly increases, also, the emission of toxic compounds from this period increases proportionally. The factors which determine the value of the emission are the forces shaping transient states and the technical condition of the engine itself. To describe the transient states, authors propose the use of multi-equation models, the presented material focuses on the analysis of changes in toxic compound concentrations during transients at varying pressures of the injector opening, which is a typical regulatory parameter that undergoes relatively frequent changes in the process of using the engine. This paper presents a description of transient states using multi-equation models, and the analysis of their relevance. It also presents a comparison of toxic compounds concentration at modified angles of fuel injection advance.

Keywords: diagnostic, theory of experiments, marine diesel engine, exhaust gas toxicity, multi-equation models

1. Introduction

Transient states are exceptional marine diesel engine operating conditions. They arise in the absence of thermodynamic equilibrium in the engine cylinders and are an important part of the engine load spectrum, especially of traction engines, thereby without affecting the emission of toxic compounds. Engine research in this area is forced because of homologation, where the main problem comes down to the optimization of the combustion course with variable engine load described even through urban driving tests.

In the case of marine diesel propulsion, the importance of transient states, in the above sense, is less prominent because of the relatively small proportion of transients in the engine load spectrum. If, however, such an analysis is subjected to the movement of individuals in
specific areas or maneuvering in port, the proportion of transients in the engine load spectrum grows significantly and is worthy of special consideration. Proportionately to this growth increases the emission of toxic compounds, caused by the impact of those states. This should be explained by the fact that transients interfere with cylinder thermodynamic equilibrium, which occurs during the fixed charges. This interrupts the combustion process by causing temporary changes, primarily to the stream of fresh charge of the cylinder, but also the amount of fuel delivered. Thus, the air-fuel ratio changes temporarily, which results in the changes in air excess ratio, leading to increased emissions of combustion products created due to the local oxygen deficit. A further consequence of the appearance of increased amounts of carbon monoxide (CO) and unburned hydrocarbons (HC) is to lower the combustion temperature, which determines the reduced NOx emissions.

The deciding factor in the emissions of toxic compounds derived from transient states is primarily the value of force, which causes these conditions. But this is not the only factor. Another factor affecting the emission of toxic compounds derived from transients that has to be taken into consideration, is the condition of the engine. This condition, described with the structure parameters while using the engine, is constantly changing, which is responsible for the processes of wear. This change enhances the formation of toxic compounds during transient states, as these processes, though short, are so dynamic that the instantaneous concentrations frequently exceed ZT values of the steady states. Therefore, it is expected that the engine with its structure parameters changed due to wear, will be more sensitive to the effects of transients and thus it will be easier to determine its technical condition [5].

The correct course of combustion in the cylinder depends primarily on well-functioning supply system, which is to ensure process repeatability mainly fuel injection. Because of this repetition, not only the beginning and the end of the injection is important, but also its course. The correctness of the first criteria (the beginning and the end of injection), in the classic power systems, is largely protected by the high-pressure fuel pump, through adjustable parameters such as injection timing and fuel dose. These parameters, although significant in view of their impact on the combustion parameters, especially in the case of fuel injection timing, undergo rather small changes during the use of the engine, and if present, they are usually the consequence of an incorrect adjustment of fuel equipment. Authors have examined the issue of the impact of injection timing on the combustion parameters during transients in earlier studies [11]. Although, as previously mentioned, the beginning and end of the injection correspond to the fuel pump, its course corresponds to the injector, and more specifically the parameters that describe its work. The most important regulatory parameter which determines the shape of the injection, its accuracy and above all its repeatability, is the opening pressure of the injector. This parameter, compared to the ones previously mentioned, undergoes the most common changes during the operation of the engine, and despite its effect on the combustion process is incomparably smaller than, for example, that of the fuel injection timing, it still must be taken into account in the analysis of the combustion process. The parameter determines the quality for fuel atomization, and thus determines the preparation of a homogeneous combustible mixture in the cylinder, which is especially important in states of thermodynamic imbalances that occur during transients, when the extortion on the power supply is much larger and this issue it the main topic of this paper. Other parameters, pertaining to the injector and having an impact on the course of the injection, are focused on the parameters that describe the geometry of the atomizer, which, as it stands, also undergoes changes during the engine's operation, e.g. due to erosive fuel interaction. This issue can be found in the earlier studies of the authors [10].

The paper will present the modeling of transient states with a variable angle of injection timing and their impact on the changes in the basic concentration of toxic compounds.
2. Identification of a dynamic process of multi-equation model

Building on the experience of authors [6,7,8,9,10, 11] with modeling of toxic compounds concentrations, it was decided to implement the multi-equation models, proven during steady state, for the analysis of dynamic processes, whereby it is assumed that the change process of gas toxicity occurs throughout a time, which makes it dynamic. Therefore, the model was described as multi-equation system of linear differential equations. Since the measurement of the concentration of toxic compounds is a discrete measurement, discrete-time signal (time series) is a function whose domain is the church of integers. Thus, a discrete-time signal is a sequence of numbers. Such sequences are referred to as recorded in the functional notation. The adoption of such a notation was striving to minimize the impact of errors including the approximation of functions that would have to occur when using the continuous functions.

Discrete-time signal \( x[k] \) is often determined by sampling \( x(t) \), a continuous signal in time. If the sampling is uniform, then \( x[k] = x(kT) \). Constant \( T \) is called the sampling period. Course of the dynamic process in time depends not only on the value of force at a given time but also the value of extortion in the past. Thus, the dynamic process (system) has a memory where it stores consequences of past interactions.

The relations between the input signals \( x_1[k], x_2[k], \ldots, x_n[k] \), and output signals \( y_1[k], y_2[k], \ldots, y_m[k] \), \( k = 0,1,2,\ldots \), will be described by a system of linear differential equations.

\[
\begin{align*}
y_1[k+1] &= a_{11}y_1[k] + a_{12}y_2[k] + \cdots + a_{1m}y_m[k] + b_{11}x_1[k] + b_{12}x_2[k] + \cdots + b_{1n}x_n[k] + \xi_1 \\
y_2[k+1] &= a_{21}y_1[k] + a_{22}y_2[k] + \cdots + a_{2m}y_m[k] + b_{21}x_1[k] + b_{22}x_2[k] + \cdots + b_{2n}x_n[k] + \xi_2 \\
& \vdots \\
y_m[k+1] &= a_{m1}y_1[k] + a_{m2}y_2[k] + \cdots + a_{mn}y_m[k] + b_{m1}x_1[k] + b_{m2}x_2[k] + \cdots + b_{mn}x_n[k] + \xi_m
\end{align*}
\] (1)

where:
- \( y_i[k], i = 1,2,\ldots,m \) - output signal values at \( k \),
- \( x_j[k], j = 1,2,\ldots,n \) - input signal values at \( k \),
- \( a_{ij} \) - is a coefficient found in \( i \)-th equation with \( j \)-th output signal, \( i,j = 1,2,\ldots,m \)
- \( b_{ij} \) - is a coefficient found in \( i \)-th equation with \( j \)-th input signal, \( i = 1,2,\ldots,m, j = 0,1,\ldots,n \)
- \( \xi_i \) - is a non-observable random component in \( i \)-th equation.

In analogy to (1), the system of equations (2) can be written in matrix form

\[
y[k+1] = Ay[k] + Bx[k] + \xi
\] (2)

where:

\[
A = \begin{bmatrix} a_{11} & a_{12} & \cdots & a_{1m} \\ a_{21} & a_{22} & \cdots & a_{2m} \\ \vdots & \vdots & \ddots & \vdots \\ a_{m1} & a_{m2} & \cdots & a_{mm} \end{bmatrix}, \quad B = \begin{bmatrix} b_{11} & b_{12} & \cdots & b_{1n} \\ b_{21} & b_{22} & \cdots & b_{2n} \\ \vdots & \vdots & \ddots & \vdots \\ b_{m1} & b_{m2} & \cdots & b_{mn} \end{bmatrix}
\]

\[
y[k] = \begin{bmatrix} y_1[k] \\ y_2[k] \\ \vdots \\ y_m[k] \end{bmatrix}, \quad x[k] = \begin{bmatrix} x_1[k] \\ x_2[k] \\ \vdots \\ x_n[k] \end{bmatrix}, \quad \xi = \begin{bmatrix} \xi_1 \\ \xi_2 \\ \vdots \\ \xi_m \end{bmatrix}
\]
Later denoting:

\[ C := [A | B] = \{c_{ij}\}_{mx(m+n)} \quad (3) \]

and

\[ z[k] := \begin{bmatrix} y[k] \\ x[k] \end{bmatrix}, \]

the system of equations (1) is shown in reduced form

\[ y[k + 1] = Cz[k] + \xi \quad (4) \]

Identification of the system of equations (1) and (4) will be based on the selection of the coefficients using the set of measurements on the real object of input and output signals. The problem of aforementioned selection the authors present, among others, in [6,7,8,9,10].

3. Study of dynamic process in engine fuel supply system through multi-equation models

The object of this research was the engine fuel supply system (opening pressure of the injector) of a single-cylinder test engine [8]. The experimental material was collected by trivalent developed a complete plan [4]. The originality of the presented experimental material lies in the fact that the implementation of the various measurement systems (measuring points) of the aforementioned experiment plan were carried out by a programmable controller, equipped with the dynamometer control system. This allowed a high repeatability of dynamic processes. The period between an onset of the clipping of injection system components and the re-stabilization of output quantities was adopted as the duration of the dynamic process. The time (about 106 seconds) was selected based on previous experience of the authors.

In order to identify the impact of the technical condition of the fuel supply system on the parameters of the engine power during dynamic processes, sets of input quantities (preset parameters) and output quantities (observed parameters) were defined. For the purpose of this study a set of input quantities X was limited to three elements, that is: \( x_1 \) - engine speed \( n \) [r/min]; \( x_2 \) - engine torque \( T_{tq} \) [N·m]; \( x_3 \) - injector opening pressure \( p_{wtr} \) [MPa]. The study was conducted in accordance with the approved complete plan, for three values of speed, i.e. 850, 950 and 1100 [r / min]. For each speed, torque \( T_{tq} \) increased and thus created a transient state, consequently for the load of 10, 20, 30, 50, 70 [N]. For speed of 850 r / min, afraid of a large engine overload, the loads of 50 and 70 N were omitted. Similarly, this was done to the speed of 950 r / min and a 70 N load. Injector opening pressure varied by ±3MPa, yielding three values, i.e. face value – 20 MPa, increased regulatory spring tension of the injector, which increased opening pressure of the injector – 23 MPa, and reduced injector opening pressure – 17 MPa. Repetitive transients were obtained this way. Graphic interpretation of the test program is shown in Figure 1. 36 repetitive transients were obtained this way. Graphic interpretation of the test program is shown in Figure 1.
Similar treatment was applied to the set $Y$ of output quantities, limiting the number of its elements to only the primary toxic compounds in exhaust manifold: $y_1$ – concentration of carbon monoxide in the exhaust manifold $C_{CO(k)} \text{[ppm]}$; $y_2$ – concentration of hydrocarbons in the exhaust manifold $C_{HC(k)} \text{[ppm]}$; $y_3$ – concentration of nitrogen oxides in the exhaust manifold $C_{NOx(k)} \text{[ppm]}$, $y_4$ – tsp exhaust gas temperature [$^\circ \text{C}$], $y_5$ – air-fuel ratio $\lambda$.

Figure 2 and figure 3 show a graphical representation of transients recorded at a speed of 1100 [rev/min] and the change in torque from 30 to 50 [Nm] (Fig. 2), and from 50 - 70 [Nm] (Fig. 3) for concentrations of CO and HC.

Statistical identification was made using GRETL [1]. Estimation of the equation coefficients for specific output variables was performed using the least-squares method and it had to verify the significance of its parameters and, consequently, the rejection of insignificant values, which consequently led to a significant simplification of the models. Given the large amount of experimental material, as well as due to the authors’ detailed methodology of the analysis, for the purpose of this study presented are only the most characteristic of cases which occur at the highest loads that have been achieved during the experiment.

From the analysis of the experimental material as well as the multi-equation models describing it, it is clear that both the concentration of carbon monoxide $C_{CO}$, hydrocarbon concentrations $C_{HC}$, and concentrations of nitrogen oxides $C_{NOx}$ are correlated primarily with excess air ratio $\lambda$ and with the size of the input of the presented experiment - an injector opening pressure $p_{wir}$. This arrangement seems to be obvious, as it is the amount of oxygen in the combustion chamber that determines the concentration of a particular ZT value. In turn, the value of the excess air ratio in the combustion chamber, especially its local values, is determined by the correctness of the fuel supply, in particular the correctness of the spray. As demonstrated in the introduction, the decisive factor affecting the correctness of the fuel atomization is the injector opening pressure.
Figure 2. The concentration of hydrocarbons HC for the transient at \( n = 1100 \) rpm and load change from A: \( T_{\text{iq}} = 30 \text{ Nm} \) to \( T_{\text{iq}} = 50 \text{ Nm} \); B: \( T_{\text{iq}} = 50 \text{ Nm} \) to \( T_{\text{iq}} = 70 \text{ Nm} \). \( C_{\text{HC}(17, 20, 23)} \) – HC concentration for the (17) reduced, (20) nominal, (23) increased opening pressure of the injector, where \( t \) - the time of the transient.

The resulting relevance values of model parameters are similar to those presented in previous works of injection timing interactions [11], wherein the interaction is far greater, especially in the case of the \( NO_x \) concentration. Similarities are also found in the case of the impact of other input values of the experiment. It has been observed, inter alia, that the rotational speed has a greater effect on the concentrations of CO, and in the case of HC an important factor determining the value of its concentration is the load (particularly evident in the case of the maximum torque load).

The results of the presented analysis highlight the significant advantage of multi-equation models, the possibility of multi-criteria analysis of the variables in the case where these values are in mutual correlation. Analysis of these relationships in one model reflects the reality more accurately (because there are obvious interactions between, for example, CO and HC).
and, for example, $\lambda$, and thus allows for a broader interpretation of the test problem. In the present case, significant interactions were observed between the concentration of CO and HC and a negative correlation between the concentrations of these compounds and NO$_x$ concentration, which seems to be logical considering the processes of formation of these compounds in the cylinder.

Having a good model fit to the values obtained as a result of the experiment on the engine is indicated by a small value of the sum of residuals between these results. These values, in the case of raw residues, for a model describing the concentration of HC are a maximum of 30 ppm. In the case of models describing the concentration of CO, the maximum error does not exceed 80 ppm, which is a very good value considering the maximum value of indicators that, during the experiment, was 3000 ppm. A uniform distribution of the regression residuals from the average values also indicates a proper fitting of the models.

![Graph](image)

**Fig. 3.** The concentration of carbon monoxide CO for the transient at $n = 1100$ r/min and load change: A: $T_{cq} = 30$ Nm to $T_{cq} = 50$ Nm; B: $T_{cq} = 50$ Nm to $T_{cq} = 70$ Nm; CCO (17, 20, 23) - the concentration of CO for the (17) reduced, (20) the nominal, (23) increased opening pressure of the injector, where $t$ - the time of the transient.
Despite the obvious advantages, multi-equation models do not provide direct information on the quality of changes, in this case - changes in the concentration of various toxic compounds resulting from the changes in opening pressure of the injector. Only comparing the courses of the experiment, or the analysis of the obtained models, gives a picture of the phenomenon. As it is known from observation, depending on the value of force, the course of the transient can vary significantly. These differences are largely in the intensity of the course of each phase of the transient. Most frequently the course of a typical transient can be divided into two phases. The first one is characterized by the highest growth rate, accompanied by a sharp increase in ZT concentration, which is usually several times higher than the concentration in the steady state. The second phase of the transient is characterized by a much less violent course, with a monotonic character and approaches the value of the steady state concentrations in an asymptotic way (fig. 2, 3).

In such case, it is desired to apply the criteria that would be useful in the objective assessment of the comparative levels or emissions from transients. The use of an evaluation index is one of the methods used commonly in similar cases. In earlier studies, the authors present a proposal for such indicators [11], however they do not exhaust the topic, and may be a separate source of discussion.

As mentioned above, the concentrations of individual toxic compounds derived from transients are characterized by a certain regularity and repetition, and therefore a tool had to be found that would be deprived of the above-mentioned disadvantages of the indicators, while being able to be described in the precise and objective nature of the changes in the concentrations of individual toxic. It seems that the described method would be the analysis of the correlation of individual transients. This method determines the correlation of the researched transient state and that of the transient adopted as a model describing the phenomenon. Analysis of the correlation function allows you to specify the degree of correlation and its nature. Analyzing the components of the function can infer the said transient nature, that is, the participation and intensity of the individual phases. Graphic representation of the correlation analysis is a scatter diagram presented in figures 4 and 5.

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**Fig. 4** The concentration of hydrocarbons HC for the transient at $n = 1100$ r/min and load change from $T_{tq} = 30$ Nm to $T_{tq} = 50$ Nm: CHC (17, 20, 23) – HC concentration for (17MPa) reduced (20 MPa) nominal, (23MPa) increased opening pressure of the injector
Figure 4 shows the linear correlation function of the concentration of unburnt hydrocarbons HC at an injector opening pressure of 23 MPa (green) relative to the nominal injector opening pressure (20 MPa), where the correlation coefficient was $r = 0.95$. Black color indicates a correlation $r = 1$ (for the nominal values of the injector opening pressure), while the red color indicates the correlation function of HC concentration at reduced pressure to open the injector (17 MPa) also relative to the nominal opening pressure of the injector. The correlation coefficient in this case was smaller, with $r = 0.75$. Smaller values of the correlation coefficient were affected by the dispersion of points around the correlation function, which indicates an unstable transient process (multi-equation model fit is nevertheless significant because the largest residual value is 22 ppm). Analogously, correlation analysis may be performed for the carbon monoxide (fig. 5).

Fig. 5 The concentration of carbon monoxide for transient $n = 1100$ rev / min and load change with $T_{iq} = 50$ Nm to $T_{iq} = 70$ Nm: CCO (17, 20, 23) - the concentration of CO for (17MPa) reduced (20 MPa) nominal, (23MPa) increased opening pressure of the injector

The correlation coefficients indicate a high match of the correlation functions in the two cases under consideration. And so, for increased opening pressure of the injector (green), the correlation coefficient is $r = 0.78$, while for the reduced pressure, $r = 0.87$.

While analyzing the collected material, another regularity can be seen, namely, the higher the opening pressure of the injector, the higher the concentrations of CO and HC, and higher exhaust temperature. Accompanied by a decline in the value of excess air (measured in the exhaust manifold), which, despite seemingly better fuel fragmentation and thus a better mix of cargo, should be explained by a decrease in the local values of air ratio with the entry into burning more mass of (better prepared) fuel. It is evidenced by the increase of dynamics of combustion (which rises along with the increase of the opening pressure of the injector). At the same time a NOx concentration decrease can be noticed, to whose creation at least a local excess of oxygen is necessary.
4. Summary:

In the course of this study, the following conclusions have been raised:

- multi-equation models provide a good match to the empirical results,
- using a multi-equation model makes it possible to predict, and thus refine the modeling of concentration change (emission) during the transient,
- there is a need for a thorough analysis of the correlation of the dependent variables throughout the experiment,
- a discussion of the accuracy of different methods to estimate emissions, even using the method of spline functions is possible.

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THE POSSIBILITIES OF FISHING CUTTER ENERGETIC EFFICIENCY IMPROVEMENT THROUGH THE APPLICATION OF THE RENEWABLE ENERGY SOURCES

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Abstract

Both the limitations in the worldwide resources of the natural fuels and the requirements in respect of the environment protection growing stricter with time have caused in the recent years an interest in the non-conventional energy sources, also in the power systems of the fishing boats and cutters. The article presents the analysis of the possibilities of the application of the renewable energy sources on this watercraft. Amongst the numerous various sources only such have been taken into consideration which have the potential for the application on the fishing cutters, in varying extent for the propulsion purposes or for the auxiliary purposes, thus the wind and solar energy.

Key words: environment protection, renewable energy sources, power system, fishing cutters

1. Introduction

The renewable energy sources comprise the three primary sources, ie the solar, geothermal and gravity energy. Some as for instance the solar energy can be applied directly to generate heat or electric energy while the others only upon the natural conversion, eg wind or biomass or through the conversion initiated and conducted by people in various installations.

One of the physical properties of the renewable energy is its small density in comparison with the conventional energy, because for instance the solar radiation is characterised by the density <1.33 kW/m² and the wind by the density <3 kW/m² [1]. This property of the renewable energy renders hard its application on fishing cutters having the limited surface area. Nevertheless the technological progress taking place within the application technology of the renewable energy on land, and also more and more frequently on ships, provides the incentive to analyse the possibilities of its application also on smaller watercraft such as fishing cutters or fishing boats.

Under the guidance of own experience and the professional references within the application of the renewable energy sources on board the ships, it has been assumed that the same sources could be well applied on the fishing cutters and fishing boats [2, 3, 4, 5, 6, 7]. Thus two renewable energy sources have been adopted for the consideration, ie the wind power and solar energy. On account of the significantly smaller dimensions of the fishing cutters and the fact of not applying thereon the steam-powered plants it has been deemed aimless to apply solid biomass as a fuel to be burnt in boiler. Nevertheless it is worthwhile to note here the possibility of application of the liquid biomass, so eg bio-oils or alcohol for the classic Diesel engines used on cutters. In such situation
in terms of the power system arrangement and its energetic efficiency there are practically no
differences in comparison with the power system utilising the Diesel oil. The only difference
would possibly be the fuel heater necessary for the operation during the wintertime, on account of
the higher viscosity of this fuel than in case of the Diesel oil. Another possibility would be to
apply biomass in the volatile form so biogas. The main component of biogas is methane, thus the
solution of the power system of a cutter utilising this fuel is more or less the same as in case of
natural gas as a fuel. In such case the application of the renewable energy source practically does
improve the energetic efficiency either.
The energetic efficiency of the fishing cutter power system is not always however the objective
ration indicating or proving the energy consumption of the fishing process and the degree of the
environment pollution with the exhaust gas emissions. Therefore it is purposeful to apply the
energetic efficiency ratio including these two aspects thus the application of the renewable energy
sources causing the decrease of the CO₂ emission.

2. Fishing Cutter Energetic Efficiency Ratios

The fishing cutter energetic efficiency ratio could be defined as the primary energy
consumption per one tonne of the caught fish, ie:

\[ \varepsilon = \frac{GW_d}{3600M} \]  

(1)  

where:
\( G \) – hourly consumption of primary fuel, kg/h,
\( W_d \) – fuel calorific value, kJ/kg,
\( M \) – weight of the caught fish, Mg.

It is worthwhile to note the universality of the aforesaid ratio, because it is independent of the
kind of the fuel applied and moreover it provides for the application of the renewable energy
sources. If this is biomass, then since this is not a primary fuel, the value of the ration shall be
equal to zero. The similar situation will be with any other renewable energy source utilised as the
sole energy source on the cutter. The simultaneous application of the primary fuel and eg the wind
power or solar energy will cause the fuel consumption reduction thus the also the ratio decrease.

The disadvantage of the ratio proposed is however the dependence of its value from the length
of the route covered by the cutter onto the fishing grounds and back. The further the fishing area is
from the port, the worse the ratio becomes. Therefore it would seem justified for instance to apply
this ratio only during the operations at the fishing area or for a longer period of operation, eg a
year. For the purpose of the evaluation of the voyage to the fishing area and back to the port with
the caught fish the ratios could be proposed such as for the transport ship which have been
elaborated by International Maritime Organisation (IMO).

Presently all the newbuildings exceeding 400 GRT must have the determined Energy Efficiency
Design Index – EEDI [8]. Moreover, there has been implemented the voluntarily determined
Energy Efficiency Operational Indicator – EEOI that allows the current evaluation of the ship’s
transport efficiency. EEOI is defined as

\[ EEOI = \frac{\sum_j FC_j C_{ij}}{m_{\text{arg}} D} \]  

(2)  

where:
\( FC_j \) – mass of the fuel consumed during the voyage (at sea and in port) by the main and auxiliary
engines and incinerator,
\( J \) – fuel kind,
\( C_{Fj} \) – conversion factor expressed by the ratio of CO\(_2\) mass from burning the consumed fuel of \( j \) kind,

\( m_{\text{cargo}} \) – mass of cargo carried (tonnes) or the work performed (number of containers TEU or passengers) or capacity (GT) for passenger ships,

\( D \) – distance in nautical miles over which the cargo was carried or the transport operation was done [9].

The value of the conversion factor \( C_F \) depend on the coal content in fuel. For instance for the Diesel oil \( C_F = 3.206 \) [tCO\(_2\)/t\(_{\text{fuel}}\)] and for LNG \( C_F = 2.75 \) [tCO\(_2\)/t\(_{\text{fuel}}\)] [9]. This is inter alia why LNG is so strongly promoted fuel for the ships in EU. Reaching the small value of EEOI is possible chiefly through any activities favouring the reduction of the fuel consumption by ships, also in port. The basic actions to reduce EEOI while at sea comprise eg the speed reduction, utilisation of the exhaust gas waste heat but also the application of the wind or solar energy.

The aforesaid regulations do not apply to the fishing cutters yet, also in view of the vessel size. However, for the purposes of the analysis, in terms of the evaluation of the fishing cutter energy efficiency and the reduction of the negative impact on the water ecosystems, the voluntary application of the EEOI indicator is justified.

Assuming the EEOI indicator for the evaluation of the fishing cutter energy efficiency in the relation (2) the caught fish mass \( M \) should be put in place of the carried mass of cargo \( m_{\text{cargo}} \). It would be also reasonable to alternatively determine the indicator in consideration of the fuel consumption not only during the fish transport to the port, but also during the operations on the fishing grounds (changing the position, trawling). EEOI indicator presented by means of the relation (2) can be determined for one voyage or for the specified ship’s operation time or also for the specified number of voyages [9]. Similar propositions could be suggested for such determination of the EEOI indicator in respect of the fishing cutters.

3. The Application of the Wind Power on Fishing Cutter

The general characteristics of wind in terms of the possibilities of utilising its energy on a ship has been presented in the earlier works by this author, inter alia [7]. In case of the fishing cutters the utilisation of the wind energy is brought down to three solutions in practice: the application of the sail propulsion unit, “towing kite” unit or the wind turbine for the propulsion of the electric current generator.

3.1 Sail Propulsion

Until the piston type steam engine has been invented the sail propulsion, next to the rowing oars, has been the primary and basic propulsion source utilised on boats and fishing cutters. On the fishing boats for the long time there has been applied the square kind of sail so called lug sail [10]. The sails have been used on fishing cutters for many years and even long after the piston type Diesel engine has become the universally used propulsion source. They have been acting on these ships as the auxiliary propulsion unit. The example of such vessels are inter alia the fishing cutters built in Polish shipyards after the WWII. The figure 1 shows the silhouette of the very popular fishing cutter Meyerform 17 afterwards referred to as B368 built after the war in Gdańsk Shipyard [11]. She had the sails of the ketch of the area 33 m\(^2\). Similar type of sails with the surface area the Storem 4B (KS-17) cutters have had which were built within 1958-1961 in Stocznia Szczecińska, and later, ie since 1961 until 1970, in Szczecińska Stocznia Remontowa [Szczecin Repair Shipyard]. This type of sails/rigging has also been applied on the other vessels such as eg K15 or KS-177. A certain exception is KU-134 which had gaff sails of slup type.

The increase in the power output and efficiency of the Diesel engines, relatively low fuel prices as well as the troublesome operation and the restrictions caused by the basic fault of the sail
propulsion which is the necessity of tacking in case of the unfavourable wind direction made the sails disappear from the fishing cutters.

In view of the new circumstances, imposing upon the shipowners the necessity to improve the energetic efficiency, advisable also in reference to the cutters, this is the author’s opinion that the consideration of the return to the sails becomes a necessity. The sail propulsion should be computer-aided and fully automated in operation. The utilisation of such sails with the favourable wind, at least only the passage over to the fishery or back, even, owing to the scarce surface area, acting only as an auxiliary propulsion unit, may provide substantial economic benefits resulting from the savings on some amounts of fuels and the ecological benefits ensuing from the smaller CO₂ emission.

3.2 Towing Kite

The modern overhead propeller consists a kite connected with the ship by means of a 100 – 500 m long rope which constructionwise resembles a paraglide. This arrangement utilises the bigger wind velocities occurring at the heights within 100 and 500 m above sea level. The application of the kite currently is reduced only to its utilisation as the ship’s auxiliary propulsion system [7].

These solutions have already been applied on large sea-going vessels, but can also be adapted for the smaller ship’s such as fishing cutters. The towing kite forms an attractive propulsion unit because the force with which it acts directly at the ship generates the towing capacity without any additional energy losses resulting from its conversion. Assuming the average value of the propulsion efficiency of the motor ship defined according to [12] as the ratio of the towing force to the power carried to the propeller shaft tail-end, equal to ca 0.5, then the equivalent engine power output must be twice as big as the towing kite power in order to give the same speed to the ship.

The diagram of the forces acting on the cutter is shown in the figure 2 [13].

Although the application of the towing kite does not only depend on the full wind, but also beam wind and even one fourth of the wind power, then similarly as in case of the sail propulsion also in
case of the towing kite propulsion there is one basic deficiency, ie the necessity of tacking about in case of unfavourable wind direction.

![Diagram](image)

\[ P_{Dw} = \frac{F_K v_s}{\eta_n} \]

\( \text{Fig. 2. The forces acting on the cutter and wind towing power } P_{Dw} \)

3.3 Wind Turbine as the Propulsion for the Power Generator

As the wind turbine power output grows along with the square of the rotor diameter, the tendency should be to apply the biggest diameters possible [7]. However, on fishing cutter the rotor diameter size must be limited on account of the small size of the vessel and the possibility of affecting the stability. In order to determine the possible power output the calculations have been made for three different possible rotor diameters, ie 0.5m, 1 m and 1.8 m. The graph showing the relation of the turbine power output and the wind speed for the assumed diameters is presented in figure 3. As evident from the graph the power values obtained are not big. With the wind force corresponding to 6 to 7 in Beaufort scale (ca 16 m/s) the power approximately 3 kW can be obtained from the turbine with the biggest diameter, ie 1.8 m. With the rotor diameter of 1 m it would be 1 kW and in case of the smallest turbine just ca 300 W.

![Graph](image)

\( \text{Fig. 3. The power outputs of the wind turbine dedicated for cutter for different rotor diameters in the function of the wind velocity} \)

Currently on the market there are available numerous wind power generators dedicated for the sea-going yachts and the small inland boats. These are chiefly the turbines with horizontal rotation axis
rotors [commonly referred to as the horizontal axis wind turbines HAWT] fitted with 3 blades. These are lightweight structures and resistant to sea environment. The diameters of the wind turbogenerators intended for yachts are within ca 0.5 m up to ca 2 m. Therefore from just this range the diameter values have been adopted for the aforesaid calculations. These turbines also perform excellent on fishing cutters and fishing boats.

The table 1 shows the annual theoretical amount of generated electric power subject to the wind velocity and fuel amounts possible to save by cutter under assumption of the fuel consumption by the cutter engine in the amount of 0.240 kg/kWh in respect of turbine with rotor diameter of 1.8 m (Zephyr) [13].

<table>
<thead>
<tr>
<th>Average annual wind velocity [m/s]</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annual electric power generated [kWh]</td>
<td>260</td>
<td>660</td>
<td>1260</td>
<td>2050</td>
<td>2970</td>
<td>3930</td>
</tr>
<tr>
<td>The amount of fuel saved [kg]</td>
<td>62</td>
<td>158</td>
<td>302</td>
<td>492</td>
<td>713</td>
<td>943</td>
</tr>
</tbody>
</table>

The offer of the wind turbine manufacturers dedicated for the smaller watercraft covers also the vertical axis wind turbines [VAWT]. An example of vertical axis design is the turbine with the Savonius rotor of Maglev System CXF-400 type developing the 400 W power with the 12 m/s wind speed. The rotor diameter is 1.2 m and its weight is 30 kg [14]. The parameters of such turbines are not that promising as those of horizontal axis design turbines. First of all they are heavier, larger and more expensive. Nevertheless they are the object of the further investigations inter alia in terms of their application on fishing cutters due to the advantage which is the low wind velocity with which they initiate, ie ca 1 m/s. The research has shown that eg with the 12 V batteries they are efficiently charged even with the small wind speed. If the charging current from the wind generator is sufficient at the time of sailing during the day, night or during the standstill, then it is possible to reduce the charging current from the engine driven generator which transposes onto the reduction of the fuel consumption.

From the foregoing analyses it can be seen that the application of the wind power generators on fishing cutters can produce concrete savings in the amounts of the fuel consumed in the annual perspective.

4. The Application of the Solar Energy on the Fishing Cutter

In practice the solar radiation energy is most often converted into heat (photothermal conversion) or into the electric energy (photo-voltaic conversion). On large watercraft as for the time being the photo-voltaic conversion has been applied. In case of fishing cutters characterised by small share of heat in the general power balance, in comparison with the large ocean-going vessels, application of only photo-voltaic conversion is most justified.

Since a single photo-voltaic cell (PV) has low voltage, insufficient to supply directly the electric power consumers, it is necessary to connect many cells. As an example connecting in a series 36 cells (typical module) made of crystalline silicone allows to obtain the voltage within 15-16 V – sufficient to charge 12 V lead-acid batteries [15]. The module means most frequently the smallest set of interconnected photo-voltaic cells complete with the mechanical construction protecting from the external conditions. The modules are also connected to form bigger elements referred to as the photo-voltaic panels. In most occasions they have the dimensions approximately 1600x900 mm. From such surface the peak energy to be obtained approximates 240 W. The solar cells are hermetically closed, thus resistant to the weather conditions which allows their manufacturers to issue long-term guarantees (often ca 25 year). The market prices of the cells recently remain below 1 €/W (average ca 0.8 – 0.9 €/W).
There are many small boats and yachts in operation worldwide which utilise the photo-voltaic cells to supply the propulsion system with electric motor or just to charge the batteries. The common feature of these vessels are their minor dimensions, operation in the absence of rolling and small sailing speed which is associated with the insignificant demand for the propulsion power. On sea-going yachts the photo-voltaic cells have been used to charge batteries. They often co-work with the wind turbogenerator. The flexible panels are particularly worth mentioning as they can be attached to the even and level surfaces of the superstructure or deck, eg by gluing. Although the fishing cutters are not any big vessels, then the required main propulsion power output values are so big that the application of PV cells dedicated to generate the electric power to fully cover the propulsion system power demand is not realistic. The small dimensions of cutters, their construction as well as their specific tasks make the arrangement of the PV cells on board of these vessels harder than on small cruising ships or yachts.

On the other hand it is justified to utilise the PV cells on the fishing cutters for the purpose of charging batteries. The photo-voltaic panels which are used on yachts have been checked in marine conditions and are excellent choice for the installation on fishing cutters. However, it should be considered that the photo-voltaic cells are more exposed to mechanical damages on the cutters. Therefore the PV cells should be positioned in such way that the fishing operations are carried out without exposing them to mechanical damage. Namely, care should be taken so that the module cannot get into collision with boom, metal rigging elements such as eg shackle or fishing equipment. They are likely to break the board and thus cause the irreparable module destruction. In practice this means the limitation of the surface available for the installation of the photo-voltaic panels. The PV cell location should ensure the best possible exposure to sunlight, therefore little shadow. Often on account of the PV cell capacity it is better to apply several smaller modules in numerous locations instead of one big unit. On cutters, owing to the aforesaid limitations, this becomes of particular importance.

The most advantageous place to locate the cells on fishing cutter is the superstructure roof/top and possibly partially also its side walls. Another attractive place in terms of the surface area available are the hold hatch covers, but on the other hand this is a location where the cells are very likely to get damaged. For instance for the cutter of type Storem 7 (fig 4) of 20.67 m length overall and beam of 5.59 m the area estimated as available for the PV modules on the superstructure is ca 10 m², and including the hatch cover this is ca 12.5 m².

Fig. 4. The prototype Storem 7 cutter and the proposed PV module installation locations [16]
The reinforced panels, e.g., SOL-FL HD model made by GTB-SOLARIS, can be mounted on the hatch cover. The boards in these cells are laminate-coated, mounted on white, flexible plastic board which acts as the carrying element. Additionally, the panel is reinforced by means of the aluminium sheet that protects the panels against the dents (caused by the uneven substrate). It makes them partially resistant to the indentations resulting from walking on the panels under which there is an uneven surface. Because the cells are flexible, they can be mounted also on a curved surface.

Assuming on the basis of the cell manufacturer’s data that on the superstructure according to the STC [standard test conditions] from 1 m² (at the temperature of 25°C with the radiation of 1000 W/m²) the maximum power output to be obtained is 150 W and from panels on the hatch cover 130 W, then maximum to be obtained jointly is ca 1800 W. These are values of peak power but it is worthwhile to note that they significantly exceed those obtained by the wind generators analysed earlier.

In order to determine the electric energy actually accumulated owing to the installed PV cells one should take into consideration the actual radiation value and time of sun operation within specified time (usually within a year). An often used indicator for the electric energy generation is that representing the ratio of the generated electric energy in relation to 1 W of peak power (frequently this unit is referred to as Wp). The value of this ratio will not be constant because the sunlight exposure conditions may vary in different years. The differences are generally not big and basing on such data the annual gains of electric energy can be estimated which allow to determine the amounts of fuel saved on a fishing cutter. The table 2 shows such data for CIS type cell (with the absorber containing Cu(In,Ga)Se₂) [28].

<table>
<thead>
<tr>
<th>Year</th>
<th>Radiation on module surface, Wh/m²</th>
<th>Energy generated per 1Wp for the stationary module in STC, Wh/Wp</th>
</tr>
</thead>
<tbody>
<tr>
<td>2000</td>
<td>904 342</td>
<td>734</td>
</tr>
<tr>
<td>2001</td>
<td>976 125</td>
<td>767</td>
</tr>
<tr>
<td>2002</td>
<td>1 002 156</td>
<td>811</td>
</tr>
<tr>
<td>2003</td>
<td>934 826</td>
<td>764</td>
</tr>
</tbody>
</table>

Thus on the average it can be assumed that the module generates ca 770 Wh/Wp per year. If this value is adopted for the modules proposed in the foregoing example of the cutter with the cells for which the peak power has been estimated as 1800 W, this means a possibility of generating annually 1386 kWh. This is a value of a level obtainable by one of the most efficient 2 wind turbines Airdolphin Mark-Zero/Pro with the wind velocity ca 5-6 m/s (table 1). Therefore the savings in fuel consumption will be similar and amount to ca 400 kg per year under assumption that the specific fuel consumption of the engine would be the same.

The data shown in table 2 refer to the cells produced more than 10 years ago. Presently the ratio of electric energy generation as stated by some manufacturers such as eg REC SOLAR for the cell Solar 260PE is at the level as high as even 1150 Wh/Wp [17]. In case of the application of the panels with such cells it would be possible to obtain even 2070 kWh electric energy per year which corresponds to the savings of ca 500 kg fuel annually.

5. Summary

It is hard to explicitly state which of the presented solutions related with the utilisation of the wind and solar energy should be considered in the first place. This depends to a large extent on the financial possibilities of the ship owners. The analyses show that the market offer is abundant in the wind turbogenerators and photo voltaic modules differing largely in prices.
This would be worthwhile to pay attention to the possibility of hybridisation of the power systems where there can be simultaneously wind turbogenerators, solar modules as well as sails. The hybrid system where besides the Diesel engine there is the electric motor acting also as the generator, supplied from the batteries charged from the photo-voltaic cell, wind turbogenerator or from the land is shown in figure 5.

Charging the batteries is also possible by Diesel engine when the electric machinery acts as generator. The propeller can be driven by Diesel engine only, or only by electric motor or by both engine and motor together. The operation of the cutter with the utilisation of the electric motor only apart from the emission reduction has also such merit that the power system does not emit noise.

References

[9] MEPC.1/circ.684 Guidelines for voluntary use of the ship energy efficiency operational indicator (EEOI)
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“Conducting the expertise of the restructuring and modernising plans for Polish fishing fleet, at the example of chosen vessels, aiming to reduce the negative impact on the water ecosystems”