PUBLISHER :

Address of Publisher & Editor's Office :

GDAŃSK UNIVERSITY
OF TECHNOLOGY
Faculty
of Ocean Engineering
& Ship Technology

ul. Narutowicza 11/12
80-952 Gdańsk, POLAND
tel.: +48 58 347 17 93
fax : +48 58 341 47 12
e-mail : sekoce@pg.gda.pl

Account number :
BANK ZACHODNI WBK S.A.
I Oddział w Gdańsku
41 1090 1098 0000 0000 0901 5569

Domestic price :
single issue : 20 zł

Prices for abroad :
single issue :
- in Europe EURO 15
- overseas US$ 20

ISSN 1233-2585

POLISH MARITIME RESEARCH

in internet
www.bg.pg.gda.pl/pmr.html

POLYTECHNICA GEDANENSIS
PATRIAE MARIQUE FIDELIS

POLISH MARITIME RESEARCH

No 3(53) 2007 Vol 14

CONTENTS

The papers published in this issue have been reviewed by :
Prof. Cz. Dymarski ; Prof. J. Girtler
Assoc. prof. Z. Korczewski ; Prof T. Szelangiewicz

Editorial

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

Engineering, **Computing & Technology**, **Mechanical Engineering**,

which could find applications in the broad domain of maritime economy. Hence there are published papers which concern methods of the designing, manufacturing and operating processes of such technical objects and devices as : ships, port equipment, ocean engineering units, underwater vehicles and equipment as well as harbour facilities, with accounting for marine environment protection.
The Editors of POLISH MARITIME RESEARCH make also efforts to present problems dealing with education of engineers and scientific and teaching personnel. As a rule, the basic papers are supplemented by information on conferences , important scientific events as well as cooperation in carrying out international scientific research projects.

Scientific Board

The catamarans *George* and *Energa Solar*

Dariusz Duda,
Wojciech Litwin
Gdansk University of Technology

ABSTRACT

Since a few years students of the Faculty of Ocean Engineering and Ship Technology, Gdansk University of Technology , have designed and built untypical floating units. Until last year their efforts were focused mainly on leg-driven boats. The boats of interesting design have taken part in yearly competitions : the International Waterbike Regatta. Their advanced design made it possible to compete with the best boats from Croatia, Holland, Germany, Turkey and Italy. Recently the students have designed and built a solar-energy-driven boat. It is the catamaran Energa Solar which took part in a prestigious regatta : the Frisian Nuon Solar Challenge carried out in Holland in summer 2006.

Keywords: leg-driven boats, solar- energy - driven boats, renewable energy

INTRODUCTION

The Scientific Circle of Students *Korab* has acted for many years at the Faculty of Ocean Engineering and Ship Technology, Gdansk University of Technology, however presently an extraordinary activity was shown by the students associated in it.

The revival happened already in 2004 when a group of students attempted to design and build a leg-driven boat intended for taking part in the International Water-bike Regatta whose 6th edition had to be performed in Bremen in May 2005.

The RW-4 catamaran designed and built within a short time, made successively its debut : it was ranked sixth among 28 boats built by students from Croatia, Holland, Germany, Turkey and Italy. The success has motivated the team to an even greater effort. Basing on experience gained during the regatta it was decided to build an entirely new boat which could be able to compete for a place on podium. In the end of 2005 the Rector of Gdansk University of Technology obtained an official inquiry from the side of the organizers of the regattas of solar bikes, Frisian Nuon Solar Challenge, if somebody of the University would be interested in starting. The students of the Scientific Circle promptly declared their will to take part. However, as revealed clear in a short time, the task imposed on themselves, namely the building of two boats practically in parallel appeared difficult to be obtained, as the International Waterbike Regatta was scheduled for the end of May and the Frisian Nuon Solar Challenge – for the end of June, i.e. during examination session.

DESIGNING AND BUILDING THE CATAMARAN *GEORGE*

Debuting in 2005, the catamaran RW-4 was reliable and fast, and of excellent manoeuvrability due to its azimuthing propeller. Its main drawback was large weight resulting from the application of simple polyester composites (for hulls) and usual structural steel (for frame).

In designing the new boat, was taken into account the specificity of the regatta within the frame of which seven highly different competitions such as : sprint, slalom, acceleration test, forward-and-back sprint, long distance run, bollard pull test, are carried out.

Therefore mass reduction of the boat and achieving its possibly high speed even in expense of lowering its manoeuvrability were assumed design priorities.

The analysis of the competed structures revealed that each of them had important disadvantages. Therefore the team presented three different novel design concepts. It was decided to build a catamaran driven by a pulling azimuthing propeller placed fore. To limit number of mechanical gears and to unload the boat structure the crew members were assumed to occupy places along the hulls (Fig.1).

Fig.1. *The catamaran George .*

The team planned to single-handedly design and build the hulls but cost and time consumption of such undertaking forced it to drop out the plan. The model tests realized by Ship Design and Research Centre (the CTO) in Gdańsk (Fig. 2), demonstrated that the previously applied hulls of typical canoe form are of a low resistance at an assumed speed of motion. Hence it was decided to use a more modern canoe hull. Owing to the *Gemini* company which made available hull moulds the students unaidedly produced two laminated hulls.

Fig. 2. *Model tests carried out by the CTO .*

The designed structural frame was computationally controlled by using the Finite Element Method. Most troubles were associated with the highly loaded pipe connecting both hulls in the bow part of the catamaran. It transferred wave-generated loads and torque and bending due to azimuthing propeller thrust.

The connecting frame was built of a high-grade aluminium. All frame elements were carefully prepared by the students and welded together by the company *Aluminium Ltd* at *Wisła* shipyard.

The propulsion system was so designed as to reduce, as much as possible, torque in the azimuthing propeller vertical shaft. It is important since the torque interacts with the propeller rotation mechanism that may be troublesome for swain. For this reason the rotational speed of the first two stages of the reduction gear was increased up to the assumed rotational speed of the propeller. As demonstrated later during practical tests, the torque acting in the rotation mechanism was negligible and did not make any difficulty for swain during manoeuvres. The body of the intersecting-axis gear which operated under water, was modernized for two reasons. Its flange connection with the propeller column**,** provided for by its producer, was feasible but of large gabarites, that could detrimentally influence resistance to motion of the immersed part of propulsion system. Hence the reduction gear was disassembled and the upper connecting flange removed out of the body by machining. Then the propeller column was glued with the gear body. Additionally it was assessed that the propeller was to be moved away from the propeller column, hence the existing propeller shaft was replaced with a longer one and one sliding radial bearing was added.

The propeller, a crucial element of the boat, was produced by the CTO whose help appeared invaluable. The catamaran's design process took a few months, but its building and assembling – only four weeks.

First trials performed on *Motława* river showed that the catamaran's centre of gravity should be shifted aft as the boat was slightly trimmed by the head.

After some modernization work resulting from the trials the catamaran was finally completed one day before leaving for the regatta.

After traveling via Bohemia, Slovakia, Hungary, Serbia and Bulgaria the team finally reached Turkey. There after hard fighting it managed to win 1st place in general classification. The challenge cup founded 28 years ago by the German team, first time reached Poland.

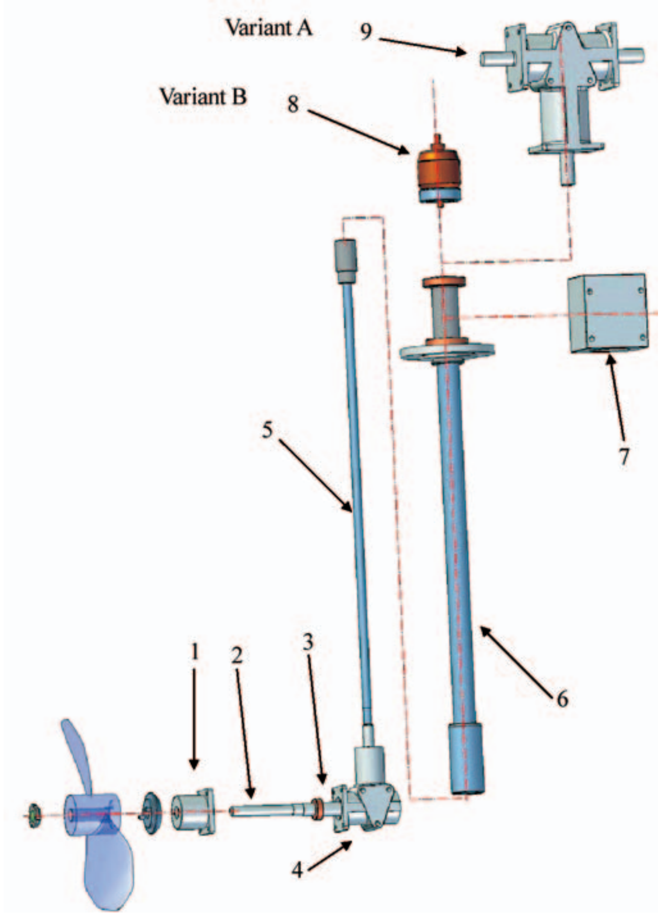


Fig. 3. *The simplified assembling drawing of the azimuthing propeller with vertical shaft;* ***Variant A*** *– for the catamaran George,* ***Variant B*** *– for the Energa Solar;* ***1*** *– seating of additional bearing supporting the shaft,* ***2*** *– the longer shaft made of stainless steel,* ***3*** *– additional sliding bearing supporting the shaft,* ***4*** *– modernized intersecting-axis reduction gear,* ***5*** *– vertical propulsion shaft,* ***6*** *– propeller column fitted with bearings and rotation mechanism flange, 7- rotation mechanism body,* ***8, 9*** *– propulsion engine.*

The team will face hard work in the next year as the International Waterbike Regatta of 2007 has to be held in Gdańsk.

DESIGNING AND BUILDING THE CATAMARAN *ENERGA SOLAR*

To be carefully acquainted with the rules was the first step in preparation of the team, made just after submitting its participation in the regatta. The regulations of over twenty pages were very extensive and detailed. It imposed values of maximum dimensions of the boat, its mass and number of crew members, as well as it even determined the minimum weight (mass) of each crew member amounting to 70 kg.

The regatta organizers established three separate classes of boats :

★ The Class A – for one-person boats having solar cells of less than 100 kg mass and 6 m length at most
★ The Class B – for two-person boats having solar cells of less than 150 kg mass and 8 m length at most
★ The Open Class –for boats of 8 m length at most, without any other limitations.

As Dutch partners suggested that the Class A would have the greatest number of competitors it was decided to build an one-person boat complying with relevant requirements for that class.

The team faced difficulties in gaining financial support for building the boat. As in the case of the catamaran *George* it was the Rector of Gdansk University of Technology and the Head of the Faculty of Ocean Engineering and Ship Technology who financially supported the building of the boat. However it was crucial to find a strategic sponsor. Ultimately, the company *Energa S.A.* accepted position of the main sponsor.

To take part in the regatta it was necessary to fulfill a few conditions. The regatta's organizer required a.o. to submit the boat's technical plans and schematic diagram of electric network. The team was concerned about if crew safety is the only thing of this request or perhaps also the gaining of experience on the basis of the submitted designs.

It was important that the company *Sharp* declared to equip each team with the same number of identical solar cells. They were only lent and had to be returned after the regatta. However this way all the teams had the identical energy sources, that - in the conditions of limited funds - was crucial and helpful in getting even to each other. The energy source for the A-class boats consisted of six panels whose total power output could be up to 600 W depending on solar operation.

The regatta's organizers allowed to store the produced energy in electric batteries of the capacity limited to about 40 Ah at 24V voltage. No charging them from land was allowed and only solar radiation had to be used.

The organizers attached high importance to safety issues therefore each boat had to be fitted with a fire extinguisher, water draining pump, emergency electric „death man switch" etc.

The planned route of the regatta run along canals of Holland. It was divided into six stages and its total length reached 230 km. In designing the boat it had to be remembered that the boat must be durable and reliable enough to cope with several day trip in variable ambient conditions.

As in the case of the catamaran *George*, the boat's concept was discussed for a relatively long time. It was sure that an azimuthing propeller will be applied. The team has already gained some experience concerning the boats of similar mass and propulsion power. Hence, knowing advantages of the catamaran concept and necessity of fitting six cell panels of a mass close to that of the boat itself, the team was convinced that the twin-hull unit should be chosen. The problem was to get appropriate hulls as the firm which already accepted the building order withdrew from it in the last moment. The hull of the canoe used up to that moment appeared to short and its displacement insufficient. In this situation 6 m hulls of one of typical sailing catamarans was chosen. The order for two such GRP hulls was placed with one of their producers. To limit their weight thickness of laminate was kept at minimum and original heavy decks were removed. As prompt delivery of the hulls was not possible it was also not possible to finally design and build the frame, without which to determine location of centre of buoyancy necessary for arranging solar cell panels and position of swain, was impossible.

Two months before the regatta the delay connected with building the frame and hulls, increased since it was necessary to focus the team's effort on the water bike because the starting date of the International Waterbike Regatta was nearer and nearer.

As late as in the end of May the work upon the solar - energy - driven catamaran started again when only four weeks remained to the date of departure for taking part in the regatta. The approaching examination session made the situation even worse as the students had less and less time to build the boat. Nevertheless the frame and hulls were completed on schedule.

An azimuthing propeller was used to drive the catamaran. From the very beginning of designing process of the boat two different design solutions were considered (Fig. 1 and 2). It was possible either to place the electric motor within an underwater pod and apply direct propulsion or to drive the propeller through an intersecting-axis reduction gear and vertical shaft and to place the motor over the propeller's column. Tests of the electric motor on a special test stand showed that to apply a rotational speed reduction gear would be more favorable. Unfortunately the directly driven azimuthing propeller was already ready to use and its modernization was rather impossible. Time was running and funds lacking. Therefore elements of the propeller with vertical shaft was promptly ordered and an intersecting -axis gear able to reduce rotational speed in half, was purchased.

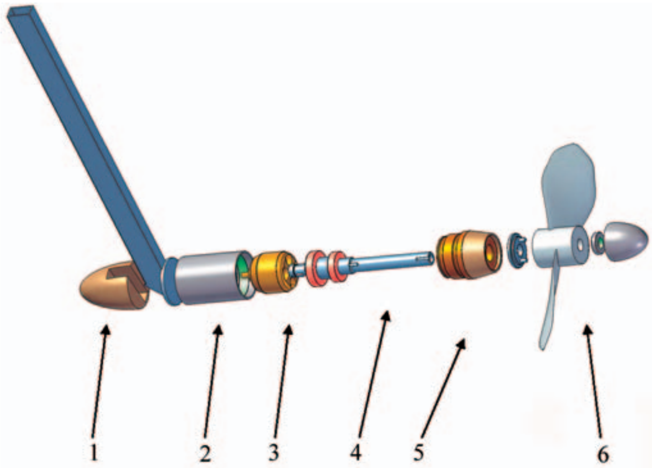


Fig. 4. *The simplified assembling drawing of the azimuthing propeller with the electric motor placed in the pod;* ***1*** *– fairwater,* ***2*** *– motor's seating,* ***3*** *– driving motor,* ***4*** *– stainless steel shaft with pair of bearings,* ***5*** *– second part of the propeller's pod with the seating of rolling bearings,* ***6*** *- propeller .*

To reduce the propeller's resistance to motion the reduction gear was subjected to the modernizations already proved in the propulsion system of the catamaran *George* (Fig. 1), i.e. the flange connecting the pod with propeller's column was removed and the propeller itself shifted away from the reduction gear.

Fig. 5. *The catamran Energa Solar .*

To drive the propeller a non-typical three-phase electric motor fitted with permanent magnets containing neodymium, was used. Owing to its relatively large diameter and a large density of magnetic field generated by the permanent magnets it produced a relatively large torque. In spite of its small mass of 680 g its output power reached almost 2kW. Results of tests performed on a special test stand showed that the motor operated correctly within the assumed range of its rotational speed. It was assumed that during the regatta the motor has to operate on the power level of 500 – 700 W. Unfortunately, during the tests some troubles associated with the motor's controller overheating, appeared. The controller fed from DC source (accumulator battery) produced three-phase alternate current feeding the motor. Its efficiency was relatively high reaching even about 90%. In effect large amount of heat was emitted. Due to compact design of the controller, heat exchange was difficult and resulting increase of temperature triggered the thermal cut-out to operate until the temperature drops enough. Because of lack of time it was not possible to order another controller. Hence the previously purchased controller was subjected to modernization work. The controller's circuit plate packets were divided into three separate segments and glued onto a radiator fitted with a fan (Fig. 6). The solution appeared effective and turned out to be correct during the regatta.

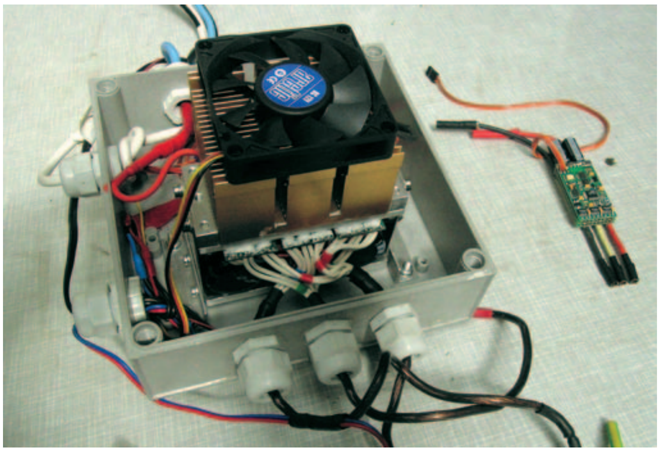


Fig. 6. *The DC/AC controller; on the left, inside casing - its modernized version, on the right – its original version before modernization .*

One of the prerequisites for taking part in the regatta was to submit to the organizers the electric network documentation for careful review in advance the regatta. Necessity of such control has been demonstrated by fires broken out on two boats as a result of overloading the electric network. The electric network of the catamaran was based on two 12V gel batteries connected in series. Each of them was placed in another hull. Left from the swain post a box containing all electric and electronic devices, was located (Fig. 3), except of the motor's controller which was placed near the driving motor. Initially it was planned that in the case of propulsion system's failure the fast switching from 24 V to 12 V voltage in the electric network, would be possible. Such option could make it possible to install a typical overboard motor used for propelling small boats. However tests carried out in the model basin of the CTO demonstrated that the catamaran's speed fitted with a 300W outboard motor would be very low, hence the concept has been abandoned.

To drive the boat a special propeller designed and manufactured by the CTO was applied.

In the course of the catamaran's manufacturing efforts were undertaken to reduce its weight as much as possible. In spite of that its mass was exceeded by almost 15 kg. It was expected that the regatta organizers would penalize the team with penalty minutes, but they came to the conclusion that the

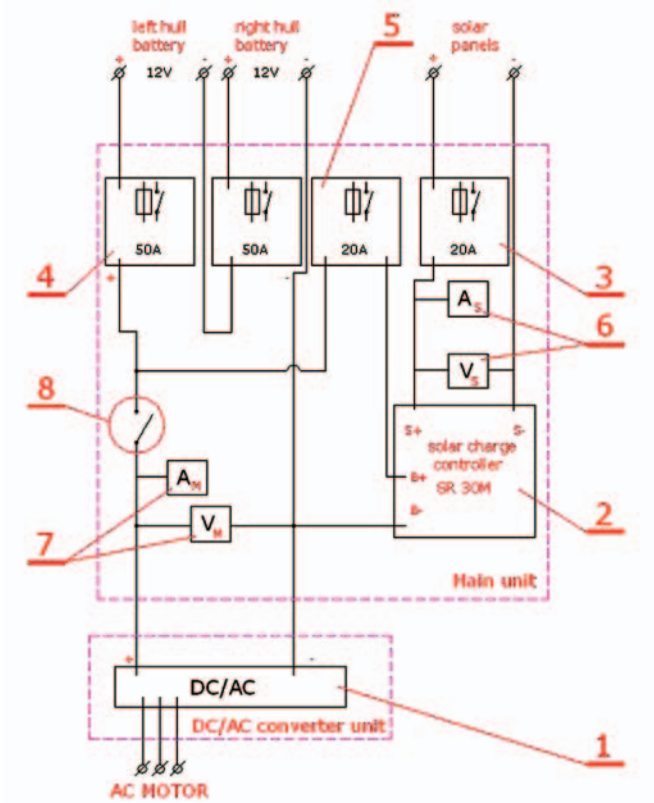


Fig. 7. *Schematic diagram of the electric network of the catamaran Energa Solar;* ***1*** *– module of AC/DC controller;* ***2*** *– controller charging accumulators,* ***3*** *– cut-out in solar cells circuit,* ***4*** *– cut-out in circuit of accumulators,* ***5*** *– cut-out in feeding circuit,* ***6*** *– power measurement system of solar cells,* ***7*** *– power measurement system of motor,* ***8*** *– safety cut-out switch .*

overweight would make harm to the team itself and no penalty was finally given.

A week before the regatta the boat was two times practically tested. Results of the tests appeared positive : the boat was fast and responsive. It was given the name of *Energa Solar*.

The preparations were continued till the day of departure. Despite some troubles the team succeeded in being in time for the regatta. The boat, after thorough technical control, was approved for taking part in the regatta.

Six days of the race were full of dramatic events. What's most important the team succeeded in finishing the race in spite of some technical problems. Unfortunately the organizers punished the team for introducing the changes to the electric system. The penalty wrote off chance of winning a place on podium and finally the team was ranked seventh among 14 boats of A class.

CONTACT WITH THE AUTHORS

Wojciech Litwin, Ph. D.
Dariusz Duda, Ms. C.
Faculty of Ocean Engineering
and Ship Technology
Gdansk University of Technology
Narutowicza 11/12
80-952 Gdansk, POLAND
e-mail : wlitwin@pg.gda.pl
e-mail : dadu@pg.gda.pl

Application of fuzzy inference to assessment of degree of hazard to ship power plant operator

Tomasz Kowalewski
Antoni Podsiadło
Wiesław Tarełko
Gdynia Maritime University

ABSTRACT

This paper presents application of fuzzy logic to assessment of degree of hazard to ship power plant operator. For the assessment a system of computer-aided identification of hazardous zone within ship power plant, was used. The system's variables representing the subject-matter knowledge in safety design area were transformed into fuzzy sets by means of appropriate linguistic variables and membership functions. The assessing of safety level of operator with the use of fuzzy inference was performed by means of an expert system programmed in the PROLOG LPA language.

Keywords: safety, operator, ship power plant, assessment, fuzzy inference

INTRODUCTION

Complex technical objects (e.g. large transport units, electric power plants and power systems, chemical systems) can create significant hazards to safety of their operators. Sea-going ship is a complex technical object specific from safety point of view. According to many domestic and worldwide elaborations and reports the ship power plant can be deemed the most hazardous place on ship.

In ship power plant many dangerous and noxious factors which constitute various hazards to its operators appear simultaneously. For instance the following can be listed: displacing machines and transported objects, moveable elements, falling elements, pressurized fluids, slippery and uneven surfaces, limited spaces, hot and cold surfaces, caustic and toxic substances. Therefore it would be advisable to indicate, just in early design stages, such places in a ship power plant, which create potential hazards to operators.

The question appears whether it would be possible to limit consideration of ship power plant only to potentially most hazardous zones during designing its safety. In the authors' opinion it is possible provided use will be made of information on structure and function of a power plant, contained in preliminary design documentation, as well as its consideration will be limited only to the zones where an operator (operators) carrying out certain service operations concerning a given technical object, appears. The idea was realized in the computer aided advising system for identification hazardous zones in ship power plant [1]. The system proposes certain safety design strategies for the zones distinguished by it.

The system's concept consists in cooperation between designer and computer where :

- the designer provides appropriate information to the system on the basis of analysis of preliminary design of ship power plant, as well as his own knowledge, intuition and experience
- the computer processes the put-in data, calculates indices and orders distinguished hazardous zones of the power plant taking into account possibility of creating potential hazard to its operator.

The description of the computer-aided identification system of hazardous zones within ship power plant was presented in [2], its elements (variables and functions) as well as a way of modelling the subject-matter knowledge for computer purposes - in [3], and a way of determining its decision variables - in [4] , [5].

In the identification system in question the additive method of estimation of degree of hazard to ship power plant operator, was chosen. A drawback of the method is the linear ranking of potential influence of particular states (symptoms) of distinguished variables, which could not be capable of modelling real influence of endangering and noxious factors on creating a hazard to operator. The drawback can be eliminated by using the fuzzy assessment and inference method. It correctly models the uncertainty associated with the assessment of influence of particular variables on the creating of a hazard to operator.

This work is aimed at consideration of possible application of fuzzy logic to assessing hazardous zones to ship power plant operators.

THE COMPUTER - AIDED IDENTIFICATION SYSTEM OF HAZARDOUS ZONES WITHIN SHIP POWER PLANT

Operation of the computer - aided hazardous zone identification system

The operation of the system in question is realized in such a way that its user – making use of a knowledge base and taking into account various limitations – performs assessment of the system's state, and on this basis he determines a degree of hazard to operator by using a set of measures. The decision

–making procedure consists in choosing the values of input variables, which are capable of causing a definite effect to operator, i.e. hazard to his life or health. Basing on the values of input variables – determined by the system's user – the system calculates value of hazard degree assessment index for considered element of hazardous zone and for n-th identification task, in accordance with the following relationship :

$$I_{H,m}^{(n)} = I_{OF,m}^{(n)} + I_{FF,m} = k_{n,m} w_n c_n + \sum_{j=1}^{J} w_j c_{j,m} \quad (1)$$

where :

- $I_{H,m}^{(n)}$ – index of influence of endangering and noxious factors on the creating of potential hazard to operator, which occur during carrying out service operations realized in compliance with the considered *n-th* procedure concerning *m*-th structural unit
- $I_{OF,m}^{(n)}$ – index of influence of service factors on the creating of potential hazard to operator, which occur during carrying out service operations realized in compliance with the considered *n-th* procedure concerning *m*-th structural unit
- $I_{FF,m}$ – index of influence of functional factors on the creating of potential hazard to operator, which occur during carrying out service operations realized in compliance with the considered *n-th* procedure concerning *m*-th structural unit
- $k_{n,m}$ – number of service operations appearing in *n*-th procedure for *m*-th structural unit
- w_n – weighing factor for *n*-th procedure
- c_n – value of service variable for *n*-th procedure
- w_j – weighing factor for *j*- th functional variable
- $c_{j,m}$ – value of functional variable for *m*-th structural unit
- J – number of functional variables.

If the task of identification of hazardous zone element is considered for a set of service procedures it will be necessary to calculate the hazard index in compliance with the following realtion :

$$I_{OF,m} = \sum_{n=1}^{N} I_{OF,m}^{(n)} \quad (2)$$

where :

- $I_{OF,m}$ – total index of influence of service factors on the creating of potential hazard to operator, which occur during carrying out all service operations distinguished by considered procedures concerning *m*-th structural unit
- N – number of considered procedures.

The assessment index of hazardous zone element – considered according to a set-in criterion (number and kind of service procedures) – is determined by using the following relationship :

$$I_{H,m} = \frac{I_{OF,m}}{\sum_{m=1}^{M} I_{OF,m}} + \frac{I_{FF,m}}{\sum_{m=1}^{M} I_{FF,m}} \quad (3)$$

where :

M – number of structural units distinguished by the considered procedures.

Decision variables of the hazardous zone identification system

As assumed according to [1], ship power plant operator will find himself in a potentially hazardous zone only when he performs definite service operations. Degree of hazard is a function of factors resulting from operation of machines and devices, access to working place, kind of performed operation as well as position of operator. The first two are associated with functional structure of power plant and constructional form of considered unit and its environment, whereas the two remaining – with kind of service task carried out in given external conditions. For this reason the sets of the factors were conventionally split into the group of functional factors and that of service factors. For purposes of the identification system in question both the groups of factors were transformed into the set of input variables for the system.

The performed classification of the factors hazardous and noxious to operator as well as the awareness of main and auxiliary working processes carried out in ship power plant, as well as of service states of ship and its power plant, made it possible to distinguish the set of 10 input variables to the hazardous zone identification system, where 6 of them belong to the group of functional factors, and 4 to that of service factors. The detail description of the procedure for distinguishing the input variables is given in [4] and [5].

In this paper only the group of functional factors is considered. The names and states (symptoms) of functional input variables used for assessing particular structural units are presented in Fig.1.

In the system in question to each of the distinguished variables is attributed the weighing factor $\mathbf{w}_j$ of constant value and five states (symptoms) of the values : 1, 0.75, 0.5, 0.25, respectively, or 0 in the case if a given symptom of considered variable does not involve a hazard to operator.

The task of the system's user is to attribute – on the basis of his knowledge, experience and intuition – appropriate values to the variables during solving a chosen identification problem of zone hazardous to operator. In the case when a variable takes zero value its symptoms are not taken into account on the system's dialogue screens (Fig.1) and then the system's user does not mark any of those symptoms whereas the system automatically attributes zero value to the variable in question.

FUZZY INFERENCE ASSESSMENT OF DEGREE OF HAZARD TO TECHNICAL OBJECT'S OPERATOR

In considering hazard to operator all influencing factors, namely functional and service ones should be taken into account. However for purposes of analysis of possible application of fuzzy inference to assessing degree of hazard to operator the authors' attention was paid only to functional factors, i.e. those associated with realization of operational processes in various service states. To this end, the functional variables were transformed into linguistic ones and fuzzy estimates were attributed to their particular symptoms. Next, to particular linguistic variables appropriate membership functions were attributed. Trapezoid is one of the most often appearing forms of membership functions [7, 8, 9]. Hence the membership functions in question were represented by trapezoids. The assumed form and distribution of membership functions constitutes a preliminary proposal only. In the next phase of the research the properties of the functions will be corrected depending on the knowledge gained from the experienced experts – ship power plant operators. The particular membership functions for the decision variables (V1÷V6) as well as the resulting variable which characterizes level of hazard to operator are presented in Fig.2.

To assess degree of hazard to operator were used the programming language Prolog and software Flint which makes

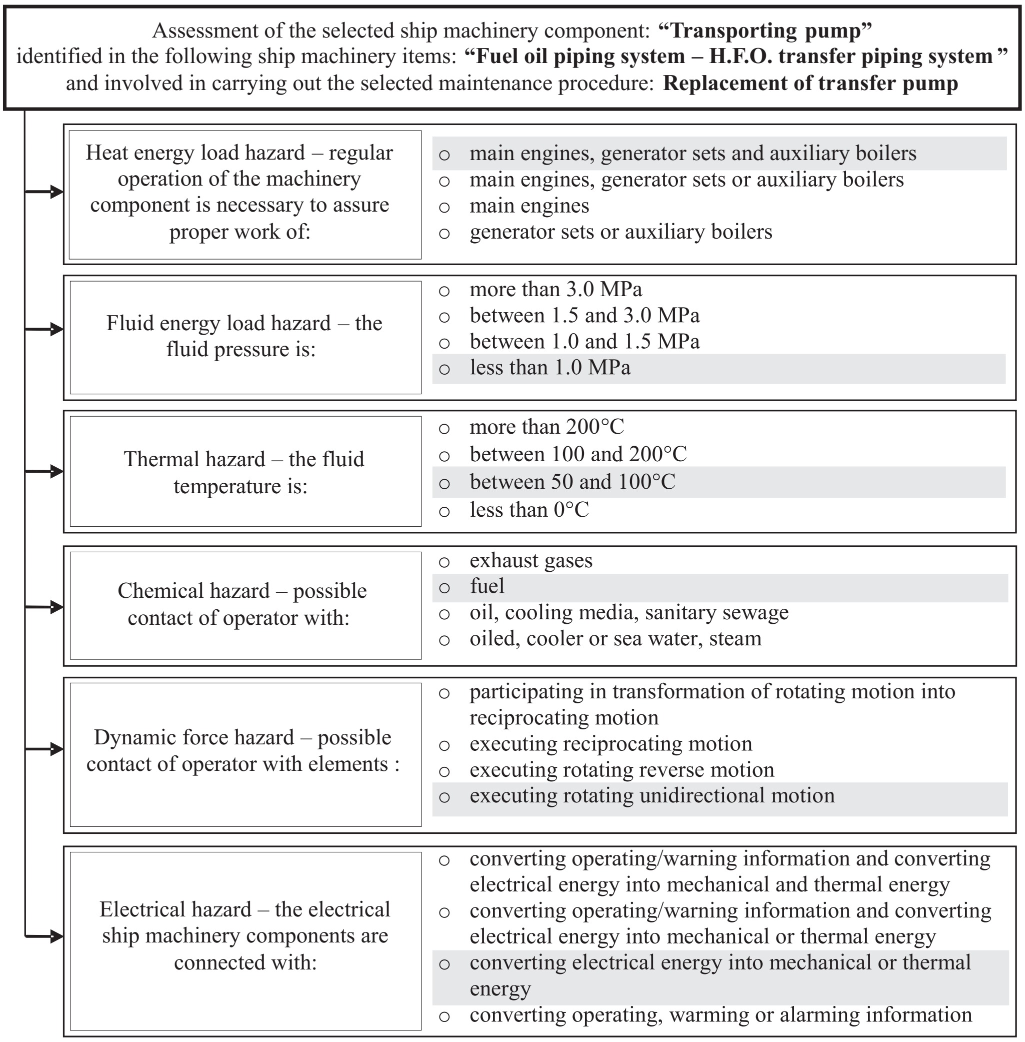


Fig. 1. *Schematic diagram of assessing hazards to operator from the side of the structural unit : “transporting pump” during realization of the procedure: “replacement of transporting pump” .*

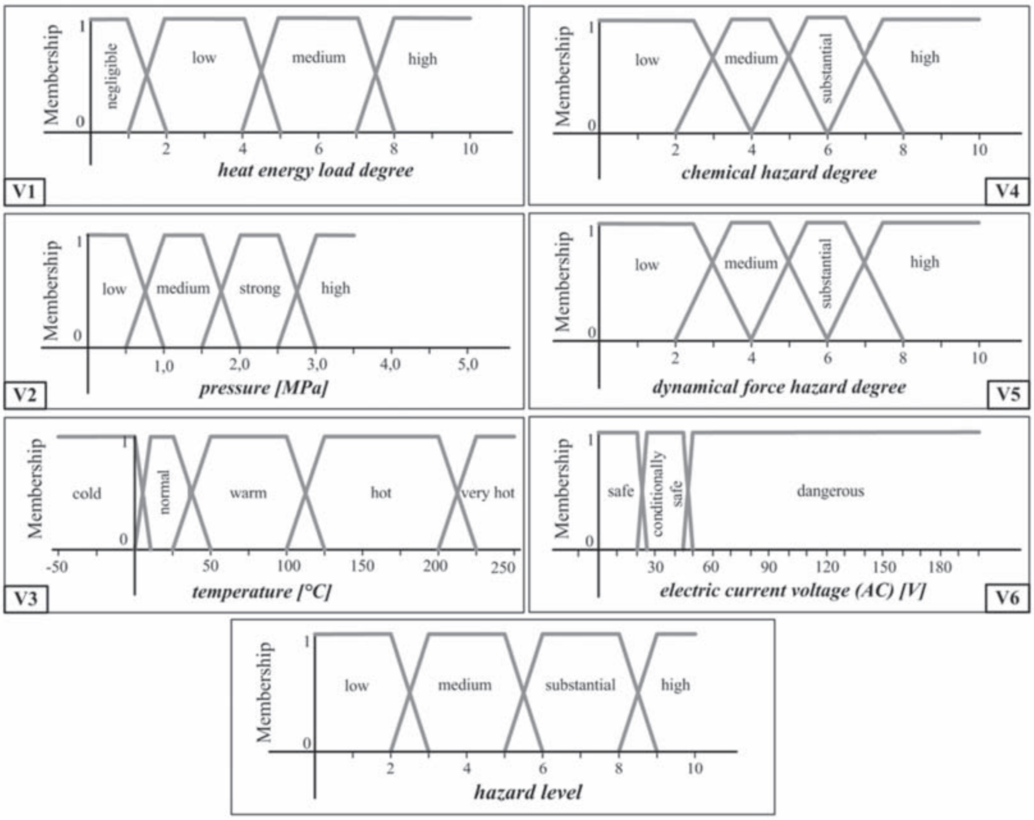


Fig. 2. *Membership functions for functional variables and hazard level .*

fuzzy inference in that language possible. The base of rules in the software Flint, for the distinguished linguistic variables and their values, has the form of the fuzzy matrix as follows :

fuzzy_matrix(t) :-
V1 * V2 * V3 * V4 * V5 * V6 -> hazard level
*small*low*normal*small*small*safe -> low*
…

The notation can be interpreted as follows : „If **V1** is **small** and **V2** is **low** and … and **V6** is **safe**, then the **hazard level** is **low**".

However in this case to fully determine all hazardous situation it was necessary to write as many as 5120 rules. Owing to the great number of variables the base of rules was simplified in such a way that hazard level is considered separately for each of the variables, that significantly decreases size of the base of rules. For instance the rules concerning the variable Z2 will have the following form :

fuzzy_matrix(pressure_reg) :-
pressure -> hazard level
low -> low
medium -> medium
high -> significant
v.high -> high.

Full estimate of hazard to operator by using fuzzy sets, performed by the system's user for a given structural unit, will be based on the estimates of particular functional variables. The partial index of hazard level assessment for the considered hazardous zone element, $\mathbf{I}_{\mathbf{FF,m}}$, will result from the fuzzy inference. The value will be obtained by means of the MAX-MIN inference method with implemented „defuzzification" by using the gravity centre method [7], [9, 10].

As an example the assessment of hazard during replacement of main engine's fuel pump is considered. In Fig.3 the functional variables used for the hazard assessment and the values attributed by the system's user, are presented.

In the considered case, the following values were attributed to the particular functional variables :

- ***V1*** *(range 0÷10) → 3*
- ***V2*** *(range 0÷3.5) →1.25*
- ***V3*** *(range -50÷+250) → 150*
- ***V4*** *(range 0÷10) → 6*
- ***V5*** *(range 0÷10) → 2*
- ***V6*** *(range 0÷380) → 220.*

During fuzzy inference process, due to aggregation of active rules, is obtained the resulting fuzzy set (Fig.4) from which – after its "defuzzification" – the final value of hazard level is achieved.

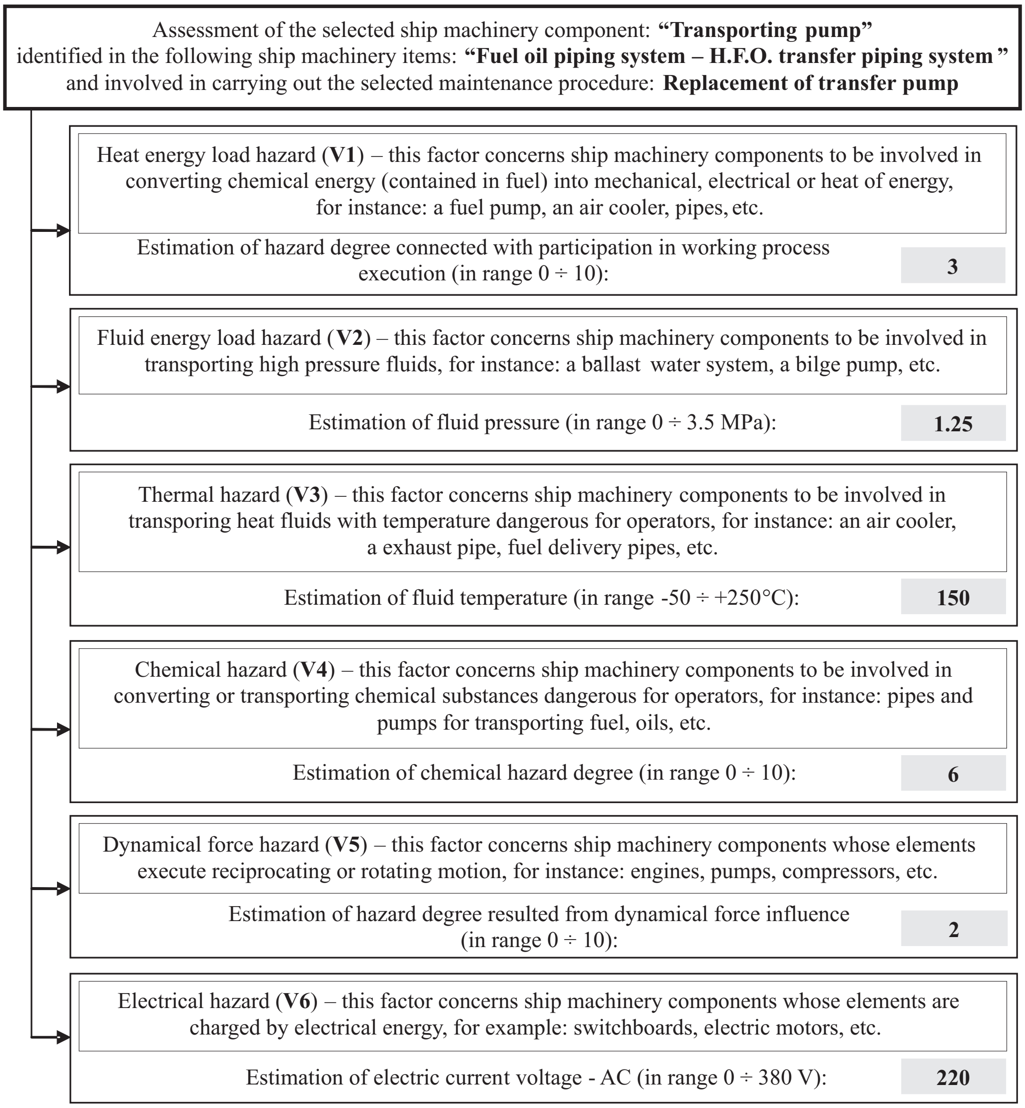


Fig. 3. *Functional variables and estimates involved by the system's user .*

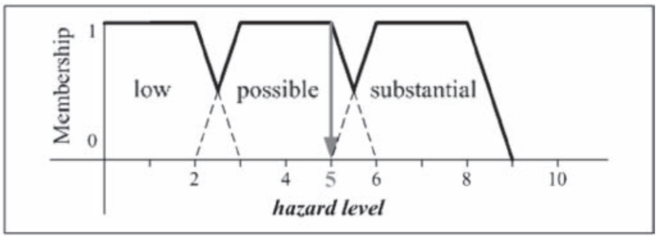


Fig. 4. *The resulting fuzzy set and final value of hazard level due to functional factors .*

CONCLUSIONS

On the basis of the obtained investigation results the following conclusions can be formulated :

- The application of fuzzy logic makes it possible to estimate operator's safety already during design process of ship power plant, especially in its early stages when information on hazardous and noxious factors to operator are not precise and associated with high uncertainty
- In contrast to the classical solutions the assessment of degree of hazard to operator by using fuzzy logic rules models more realistically qualitative aspects of human knowledge, and the inference process itself does not require performing any quantitative analyses
- The application of the special computer software which helps in using fuzzy inference, to a great extent decreases range of efforts connected with building the base of knowledge, as well as its size within the computer-aided hazardous zone identification system.

NOMENCLATURE

a, b, c – linguistic variables
$c_{j,m}$ – value of functional variable for m-th structural unit
c_n – value of service variable for n-th procedure
f(x) – function providing a membership value in [0,1] fuzzy set
$k_{n,m}$ – number of service operations appearing in n-th procedure for m-th structural unit
w_n – weighing factor for n-th procedure
w_j – weighing factor for j-th functional variable,
$\mu_A(x)$ – membership function of a given element in A fuzzy set
A, A1, A2, B, B1, B2, C1,C2 – fuzzy sets
$I^{(n)}_{H,m}$ – index of influence of endangering and noxious factors on the creating of potential hazard to operator, which occur during carrying out service operations realized in compliance with the considered n-th procedure concerning m-th structural unit
$I^{(n)}_{OF,m}$ – index of influence of service factors on the creating of potential hazard to operator, which occur during carrying out service operations realized in compliance with the considered n-th procedure concerning m-th structural unit
$I_{FF,m}$ – index of influence of functional factors on the creating of potential hazard to operator, which occur during carrying out service operations realized in compliance with the considered n-th procedure concerning m-th structural unit
J – number of functional variables
M – number of structural units distinguished by the considered procedures.
N – number of considered procedures
V1÷V6 – functional variables
X – numerical space of consideration

BIBLIOGRAPHY

1. Tarełko W. *et al.* : *Antropo-technical assumptions for designing safe technical objects – subject-matter report* (in Polish). Report on realization of the research project no. 8T07C 01120 financed by KBN. Bulletin no. 1. Basic Engineering Department, Gdynia Maritime University. Gdynia, 2004
2. Podsiadło A., Tarełko W. : *Identification of hazardous zones in ship power plants*. Proceedings of European Safety & Reliability Conference (ESREL 2005). Advances in Safety and Reliability,Vol. 2, A.A. Balkema Publishers. Gdańsk, June 2005
3. Podsiadło A., Tarełko W. : *Modelling a decision-making process of hazardous zone identification in ship power plants.* Proceedings of European Safety & Reliability Conference (ESREL 2005). Advances in Safety and Reliability, Vol. 2, A.A. Balkema Publishers. Gdańsk, June 2005
4. Tarełko W. : *Machinery function factors triggering off hazards for operators of ship power plants.* Proceedings of the 12th International Congress of the International Maritime Association of the Mediterranean (IMAM 2005),Vol. 1, Lisboa, September 2005
5. Podsiadło A. : *Operational factors triggering off hazards for operators of ship power plants.* Proceedings of the 12th International Congress of the International Maritime Association of the Mediterranean (IMAM 2005). Vol. 1, Lisboa, September 2005
6. Westwood D.: *LPA PROLOG. Version 4.3. Technical Reference.* Logic Programming Associates Ltd. London, 2002
7. Łachwa A. : *Fuzzy world of sets, numbers, relations, facts, rules and decisions* (in Polish). EXIT Academic Publishing House (Akademicka Oficyna Wydawnicza EXIT). Warszawa 2001.
8. Niczyj J. : *Multi-criterion reliability optimization and technical state estimation of bar structures by using theory of fuzzy sets* (in Polish). Scientific Research Reports of Szczecin University of Technology, No. 581. Department of Theory of Structures, issue No. 3. Szczecin, 2003
9. Piegat A. : *Fuzzy modelling and control* (in Polish). EXIT Academic Publishing House (Akademicka Oficyna Wydawnicza EXIT). Warszawa, 2003
10. Piech H. : *Inference based on fuzzy strategies* (in Polish). Publications of Częstochowa University of Technology. Częstochowa, 2005.

CONTACT WITH THE AUTHORS

Tomasz Kowalewski, M.Sc., Eng.
Assoc. Prof. Antoni Podsiadło
Assoc. Prof. Wiesław Tarełko
Faculty of Marine Engineering,
Gdynia Maritime University
Morska 81/87
81-225 Gdynia, POLAND
e-mail : tomkow@am.gdynia.pl

Photo: Cezary Spigarski

Main building of the Gdynia Maritime University .

An analysis of possible assessment of hazards to ship shaft line, resulting from impulse load

Andrzej Grządziela
Polish Naval University

ABSTRACT

This paper presents a proposal of identification of a degree of hazard to ship shaft line due to impulse load caused by underwater explosion. A theoretical analysis was made of influence of changes in co-axiality of shafts on second-kind critical velocities resulting from elastic deformations of hull structure in vicinity of shaft bearing foundations. Results are presented of pilotage tests of underwater explosion performed on training water area. A preliminary mathematical model of underwater explosion is given with taking into account mass of explosives and distance from the tested object. The drawn conclusions show a way of identification of hazard to shaft line by using spectral analysis and time - runs of vibration signals recorded at shaft bearings.

Keywords : ship shaft lines, technical diagnostics, modelling, vibrations, underwater explosion

INTRODUCTION

Ship propulsion systems are subjected to specific sea loads due to waving and dynamical impacts associated with mission of a given ship. Sea waving can be sufficiently exactly modeled by means of statistical methods. Much more problems arise from modelling impacts due to underwater explosion. In operation of contemporary technical objects including naval ships greater and greater attention is paid to such notions as : time of serviceability, repair time, maintenance and diagnosing costs [1]. Diagnosing process has become now a standard procedure performed during every technical maintenance. Out of the above mentioned, the notions of time of serviceability and maintenance costs seem to be crucial for the diagnosing process of ship power plant. Knowledge of a character of impulse loading which affects ship shaft line, can make it possible to identify potential failures by means of on-line vibration measuring systems. This way elimination of costly and time-consuming overhauls on dock leads to lowering operational costs and increasing ship fighting merits.

ANALYSIS OF FORCES ACTING ON SHAFT-LINE BEARINGS

Ship shaft lines are subjected to loads in the form of forces and moments which generate bending, torsional and axial vibrations. In most cases strength calculations of driving shafts are carried out by using a static method as required by majority of ship classification institutions. Moreover they require calculations of torsional vibrations which have to comply with permissible values, to be performed. Calculation procedures of ship shaft lines generally amount to determination of reduced stresses and safety factor related to tensile yield strength of material – Fig.1.

The above mentioned methods do not model real conditions of shaft-line operation, which is confirmed by the character of ship hull response, i.e. its deformations under dynamic loads. Much more reliable would be to relate results of the calculations to fatigue strength of material instead of its yield strength [5].

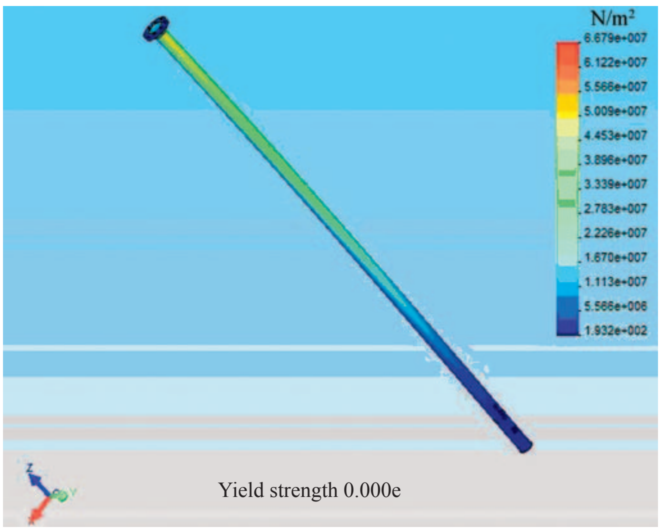


Fig.1. *Simulated static bending stresses in propeller shaft due to weight of propeller .*

In static calculation procedures no analysis of dynamic excitations, except torsional vibrations, is taken into consideration. In certain circumstances the adoption of static load criterion may be disastrous especially in the case of resonance between natural vibration frequencies and those of external forces due to dynamic impacts. To analyze the dynamic interaction a simplified model of shaft line is presented below, Fig.2.

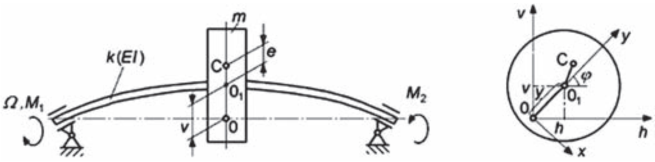


Fig. 2. A simplified shaft-line model for critical speed calculation [4] .

Let us note : M_1 - torque, M_2 - anti-torque. The system can be represented by the following set of equations :

$$m\ddot{h} + kh = me(\ddot{\varphi}\sin\varphi + \dot{\varphi}^2\cos\varphi)$$

$$m\ddot{v} + kh = me(-\ddot{\varphi}\cos\varphi + \dot{\varphi}^2\sin\varphi) \qquad (1)$$

$$(J + me^2)\ddot{\varphi} = me(\ddot{h}\sin\varphi - \ddot{v}\cos\varphi) + M_1 - M_2$$

The presented form of the equations is non-linear. Considering the third of the equations (1) one can observe that the variables h, v and φ are mutually coupled. It means that any bending vibration would disturb rotational motion of the shaft. The third of the equations (1) can be written also in the equivalent form as follows :

$$J\ddot{\varphi} = ke(v\cos\varphi - h\sin\varphi) + M_1 - M_2 \qquad (2)$$

To obtain the shaft angular speed $\Omega = \dot{e}$ constant to use time-variable torque is necessary :

$$M = M_1 - M_2 = ke(h\sin\varphi - v\cos\varphi) \qquad (3)$$

Theoretical analysis indicates that shaft bending deformation continuously accumulates a part of shaft torque. However the quantity of torque non-uniformity is rather low since shaft-line eccentricity is low; it results from manufacturing tolerance, non-homogeneity of material, propeller weight and permissible assembling clearances of bearing foundations. The condition makes it possible to predict that run-out of propeller shaft may also happen at other rotational speeds than the critical 1st kind speed calculated during design process. Taking into account the torque variability one can express Eq. (1) as follows:

$$\ddot{h} + \omega_0^2 h = e\Omega^2\cos\varphi$$

$$\ddot{v} + \omega_0^2 h = e\Omega^2\sin\varphi \qquad (4)$$

$$\ddot{\varphi} = \frac{1}{J}M(t)$$

On assumption that the average angular speed of propeller shaft equals Ω, the angle φ will change with time according to the equation :

$$\varphi = \Omega t + \varphi_0 \qquad (5)$$

and the torque M(t) can be represented as follows :

$$M(t) = M_1 - M_2 = a_\beta \cdot \cos(\beta\Omega t + \eta_\beta) \qquad (6)$$

For ship propulsion system the torque pulsation expressed by means of Fourier series is much more complex. It additionally contains components resulting from number of propeller blades, kinematical features of reduction gear as well as disturbances from main engine and neighbouring devices. In general case occurrence of only one harmonic does not change reasoning logic. After solving the third of Eqs.(4) and substituting Eq.(5) to the obtained expression one obtains the following:

$$\varphi = \frac{a_\beta}{\beta^2\Omega^2 J}\cos(\Omega t + \eta) + C_1 + C_2 \qquad (7)$$

If Ω = const then C_1 = 0. Similarly: C_2 = 0 as C_2 determines displacement of the vibration centre axis but does not change character of shaft motion, hence to neglect both the integrating constants is possible. Assuming additionally that generated vibrations are of low energy as compared with the power transmitted by the shaft, i.e. cosη = 1 and sin η ≈ η, and using trigonometric transformations one can obtain the following [4]:

$$\ddot{h} + \omega_0^2 h = e\Omega^2\cos\varphi + \frac{e\cdot a_\beta}{2\beta^2 J}\{\sin[(\beta+1)\Omega t + \eta_\beta] - \sin[(\beta-1)\Omega t + \eta_\beta]\}$$

$$\ddot{v} + \omega_0^2 v = e\Omega^2\cos\varphi + - \frac{e\cdot a_\beta}{2\beta^2 J}\{\cos[(\beta+1)\Omega t + \eta_\beta] + \cos[(\beta-1)\Omega t + \eta_\beta]\} \qquad (8)$$

From Eqs. (8) one obtains three harmonic excitations : Ω, (β + 1)Ω and(β – 1)Ω. Hence assuming that the torque pulsation occurs with the frequency β · Ω one obtains three cases of resonance for one degree of freedom of propeller shaft:

$$\Omega_{1KR} = \omega_0$$

$$\Omega_{2KR} = \frac{1}{\beta - 1}\omega_0 \qquad (9)$$

$$\Omega_{3KR} = \frac{1}{\beta + 1}\omega_0$$

where : Ω_{2KR} , Ω_{3KR} - critical rotational speeds of 2nd kind. The above given considerations deal only with one torque pulsation harmonic. In practice, at a greater number of excitations including instantaneous elastic deformations due to impulse loading, one should expect a series of critical speeds of 2nd kind whose influence will be reduced by damping due to water, stuffing-box and bearing supports [6].

For long shaft lines of ships the influence of gravity forces on critical speeds should be taken into consideration [7]. According to Eq. (10) the generated vibrations will be then performed respective to static deflection axis of the shaft.

$$m\ddot{h} + kh = me\Omega^2\cos\varphi$$

$$m\ddot{v} + kv = me\Omega^2\sin\varphi \qquad (10)$$

$$\ddot{\varphi} = 0$$

Hence the equations obtain the following form:

$$\ddot{h} + \omega_0^2 h = e\cdot\Omega^2\cos\varphi$$

$$\ddot{v} + \omega_0^2 v = e\cdot\Omega^2\sin\varphi - mg \qquad (11)$$

$$J\ddot{\varphi} = -mge\cos\varphi$$

Since in the third equation of the set (9), i.e. that for Ω_{3KR} , appears the exciting torque of the frequency/angular speed ratio β = 1 it means that one has to do with the critical state of 2nd kind for β = 1, namely :

$$\Omega_{KR(2)} = \frac{1}{2}\omega_0 \qquad (12)$$

Occurrence of such kind vibrations is conditioned by non--zero value of *e*, which – in the case of ship shaft line – appear just after dislocation of a weight along ship, a change of ship displacement or even due to sunshine operation on one of ship sides. A similar situation will happen when *e* varies due to dynamic excitations resulting from e.g. sea waving or explosion. In this case the critical speed will vary depending on instantaneous value of *e* and damping.

Theoretical analysis of operational conditions of intermediate and propeller shafts indicates that static and dynamic loads appear. In a more detailed analysis of dynamic excitations of all kinds the following factors should be additionally taken into consideration :

- disturbances coming from ship propeller (torsional, bending and compressive stresses);
- disturbances from propulsion engine (torsional and compressive stresses);
- disturbances from reduction gear (torsional stresses);
- disturbances from other sources characteristic for a given propulsion system or ship mission.

THEORETICAL BACKGROUND OF UNDERWATER EXPLOSION

Information on potential hazard resulting from underwater explosion is crucial not only for ship's commander during warfare but also for ship structure designers. Knowledge of loads determined during simulative explosions is helpful in dimensioning ship's hull scantlings [3]. Another issue is possible quantification of explosion energy as well as current potential hazard to whole ship and its moving system.

From the point of view of shock wave impact on shaft line, underwater and over-water explosions should be considered in two situations :

- when shock wave (or its component) impacts screw propeller axially
- when shock wave (or its component) impacts screw propeller perpendicularly to its rotation axis.

The axial shock-wave component affects thrust bearing and due to its stepwise character it may completely damage sliding thrust bearing. Rolling thrust bearings are more resistant to stepwise loading hence they are commonly used on naval ships [3]. The shock wave component perpendicular to shaft rotation axis is much more endangering.

Shock wave can cause: damage of stern tube, brittle cracks in bearing covers and tracks, plastic displacement of shaft supporting elements including transmission gear and main engine, and even permanent deformation of propeller shaft. For that reason contemporary naval ships are tested regarding their shock strength - irrespective of their mission – already during sea trials, Fig.3.

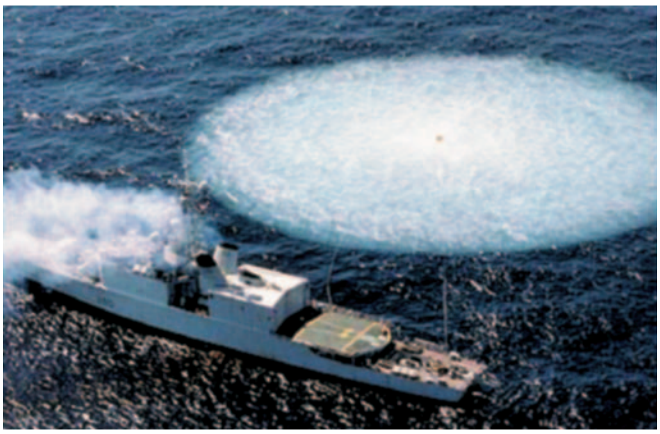


Fig. 3. Experimental tests of shock strength of frigates against underwater explosion .

The problem of influence of sea mine explosion on hull structure is complex and belongs to more difficult issues of ship dynamics. Underwater explosion is meant as a violent upset of balance of a given system due to detonation of explosives in water environment. The process is accompanied with emission of large quantity of energy within a short time, fast running chemical and physical reactions, emission of heat and gas products. The influence of underwater explosion does not constitute a single impulse but a few (2 to 4) large energy pulsations of gas bubbles [2, 8, 9]. The pulsation process is repeated several times till the instant when the gas bubble surfaces. Hence the number of pulsations depends a.o. on immersion depth of the explosive charge. The character of changes of pressure values in a motionless point of the considered area is shown in Fig.4.

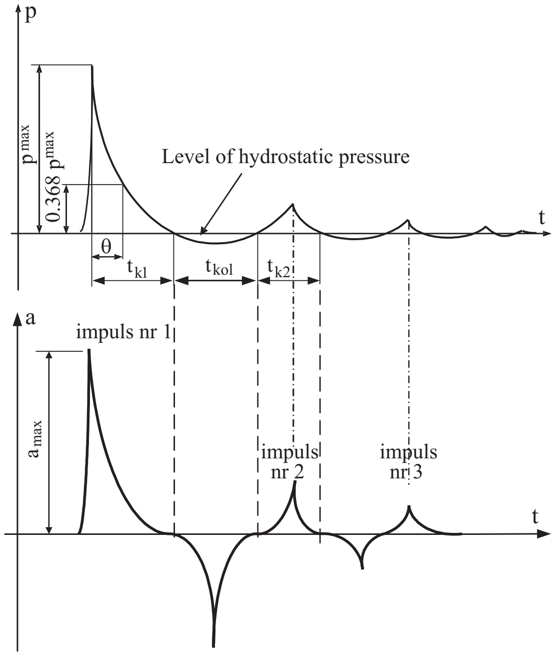


Fig. 4. Run of changes of shock wave pressure and ship hull acceleration measured on hull surface during underwater explosion .

In the subject-matter literature can be found many formulae for determining maximum pressure value, based on results of experiments, however data on a character of pulsation and its impact on ship structures are lacking.

EXPERIMENTAL TESTS

To identify underwater explosion parameters a pilotage test was performed with the use of the explosive charge having the mass m = 37.5 kg. The schematic diagram of the experiment is shown in Fig. 6. During the test were measured vibration accelerations of casings of intermediate and thrust bearings in the thrust direction and that perpendicular to shaft rotation axis. The ship course angle relative to the explosion epicentre was 45° and the shaft line rotated with the speed n_{LW} = 500 rpm. Ship's distance from the mine and its immersion depth was determined by using a hydro-location station and ROV underwater vehicle. The vibration gauges were fixed over the reduction gear bearing (p. 1) as well as on the intermediate shaft bearing (p. 2) as shown in Fig.5. The measurement directions (X and Z axes) are presented in Fig.6.

The performed test was aimed at achieving information dealing with :

- character of shock wave impact on shaft-line bearings, in the form of recorded vibration parameters
- assessment of time-run of vibration accelerations with taking into account dynamic features of the signals in set measurement points

- ⇨ assessment of possible identification of influence of pulsation of successive gas bubbles during the time-run of vibration accelerations
- ⇨ identification of features of the signals by means of spectral analysis.

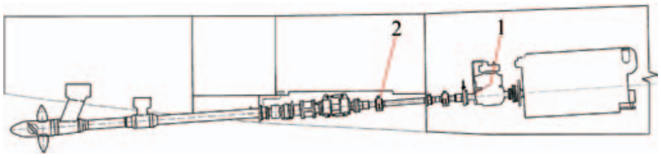


***Fig. 5.** Schematic diagram of the propulsion system of the ship in question.*
***1** - measurement point of 3-dimensional vibration at reduction gear;*
***2** - measurement point of 3-dimensional vibration at shaft-line intermadiate bearing*

Since the mass of the explosive charge was small, to reliably identify the effect of only first and second pulsation was possible during the test. The time lag of the recorded signals was the same in all measurement points, as shown in Fig.7 and 8.

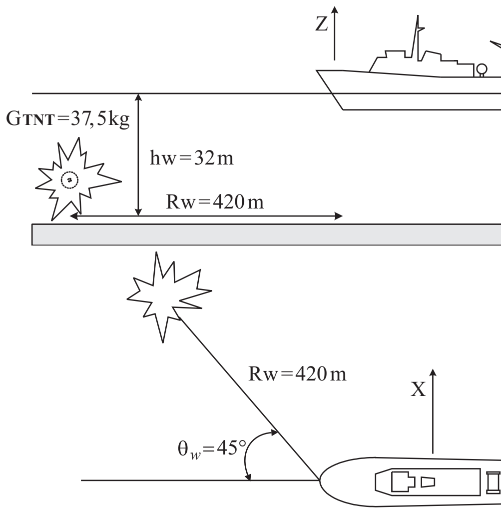


***Fig. 6.** Schematic diagram of the performed experimental test .*

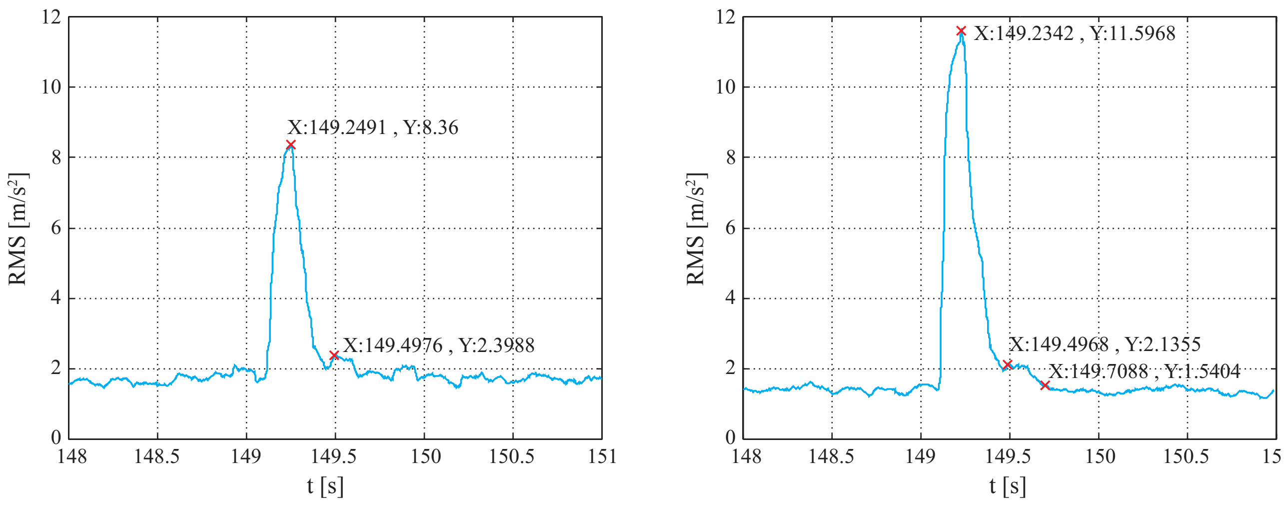
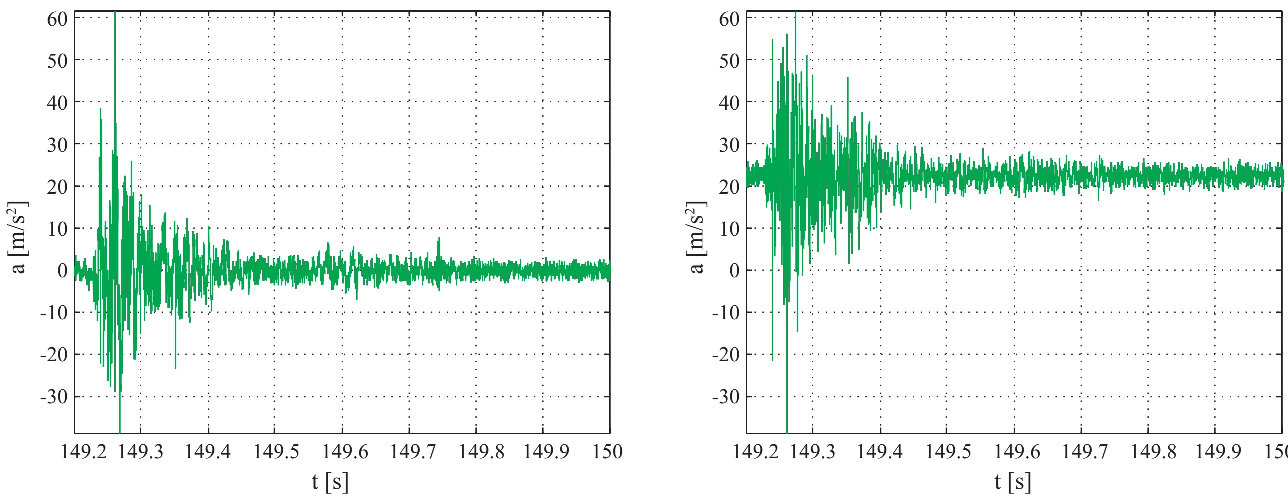


***Fig. 7.** Rms values of vibration accelerations recorded on port side (LB) and starboard (PB) intermediate bearings and thrust bearings*



$a_{z\,min} = -38.55\ m/s^2,\ a_{z\,max} = 61.57\ m/s^2$ $a_{x\,min} = -65.01\ m/s^2,\ a_{x\,max} = 41.80\ m/s^2$

***Fig. 8.** Time-run of vibration accelerations in two measurement points .*

The most important information was the assessment of the ratios of maximum and minimum values of vibration accelerations measured before, during and after underwater explosion.
The performed analysis concerned all the measurement points, Tab.1.

Tab. 1. Ratios of vibration accelerations in the measurement points K1 through K8

Indices 1 min before explosion				
Measurement points:	K2	K1	K6	K5
a_{max}/a_{min}	1.21	0.88	0.8	0.82
Measurement points:	K4	K3	K8	K7
a_{max}/a_{min}	1.24	0.9	1.02	1
Indices during explosion				
Measurement points:	K2	K1	K6	K5
a_{max}/a_{min}	13.02	9.23	11.90	10.02
Measurement points:	K4	K3	K8	K7
a_{max}/a_{min}	5.30	3.60	9.87	8.66
Indices 1 min after explosion				
Measurement points:	K2	K1	K6	K5
a_{max}/a_{min}	1.23	0.83	0.83	0.87
Measurement points:	K4	K3	K8	K7
a_{max}/a_{min}	1.34	0.92	1.06	1.08
Location	K1 – reduction gear, X axis, PB (port side)	K2 – reduction gear, X axis, LB (starboard)	K3 - reduction gear, Z axis, PB (port side)	K4 - reduction gear, Z axis, LB (starboard)
Location	K5 – intermediate shaft bearing, X axis, PB (port side)	K6 - intermediate shaft bearing, X axis, PB (port side)	K7 - intermediate shaft bearing, X axis, PB (port side)	K8 - intermediate shaft bearing, X axis, PB (port side)

It can be observed that values of the a_{max}/a_{min} ratios from before and after the explosion are close to each other; it means that the explosion impulse caused elastic deformations and next the propulsion system comes back to the initial technical state.

MODELS OF EXCITATION DUE TO UNDERWATER EXPLOSION

Analysis of dynamic impacts including impulse ones should take into account basic parameters which influence character of time-run of a given signal as well as its spectrum. The basic parameters which identify impulse impact resulting from explosion, are the following :

★ *form of impulse* which identifies kind of impulse (Fig. 9a through 9d)

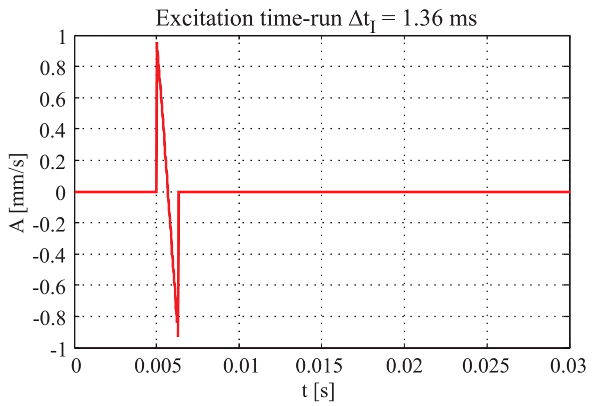


Fig. 9a. *Stepwise impulse.*

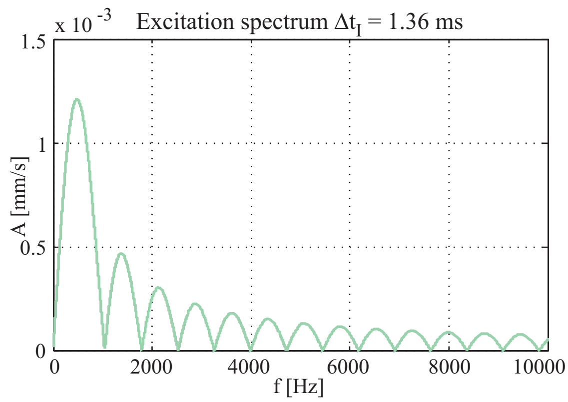


Fig. 9b. *Stepwise impulse spectrum.*

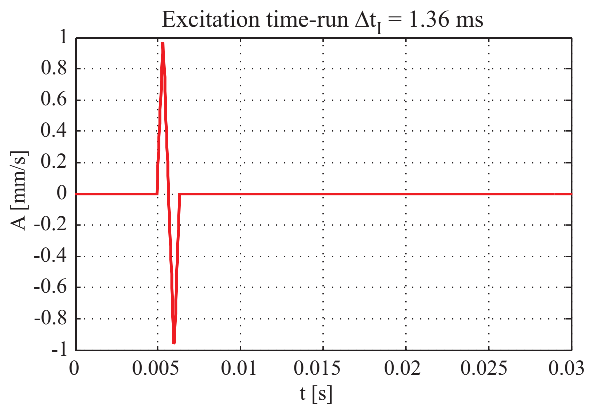


Fig. 9c. *Symmetrical impulse.*

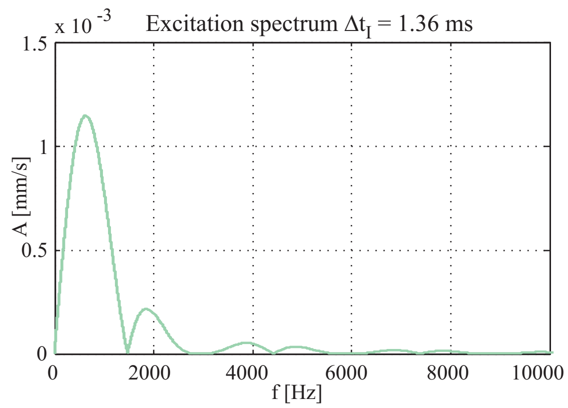


Fig. 9d. *Symmetrical impulse spectrum.*

★ impulse duration time Δt_I at the ratio $A_I/\Delta t_I$ maintained constant, which identifies explosive charge power (time of propagation of gas bubble) - Fig. 10a through 10d

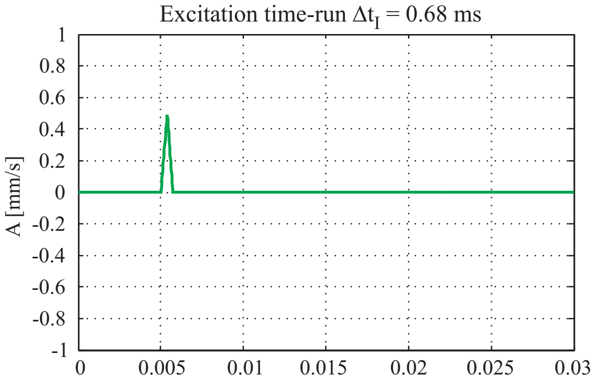


Fig. 10a. Simplified run of shock wave model in function of time (Δt_I = 0.68 ms).

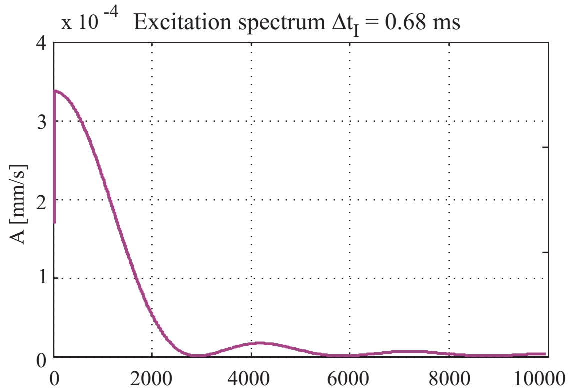


Fig. 10b. Spectrum of simplified run of shock wave model in function of time (Δt_I = 0.68 ms).

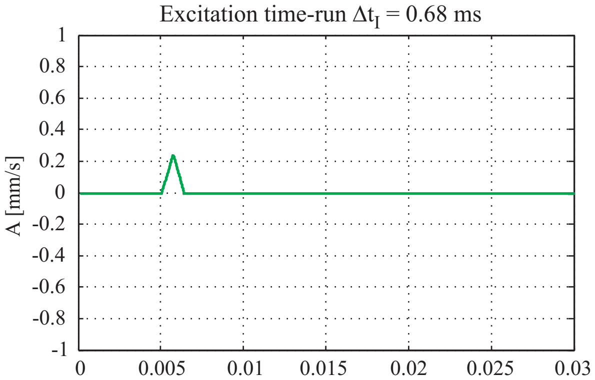


Fig. 10c. Simplified run of shock wave model in function of time (Δt_I = 1.36 ms).

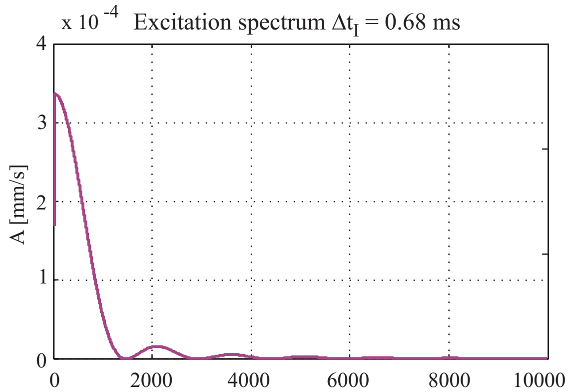


Fig. 10d. Spectrum of simplified run of shock wave model in function of time (Δt_I = 1.36 ms).

★ *influence of damping on spectrum form,* which identifies distance from explosion and - simultaneously - epicentre depth (Fig. 11a through 11d)

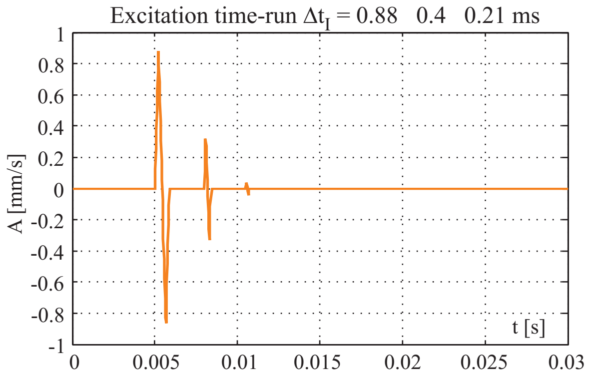


Fig. 11a. Signal which contains transformed impulses for amplitude ratios 1: 0.37 : 0.05 .

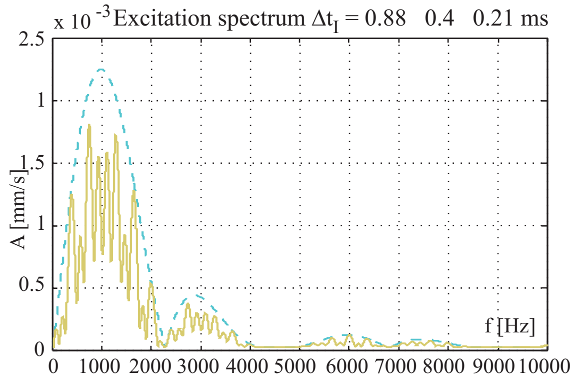


Fig. 11b. Spectrum of the signal which contains transformed impulses for amplitude ratios 1: 0.37 : 0.05 .

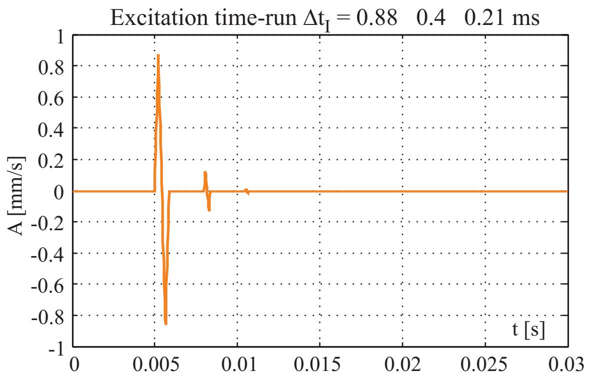


Fig. 11c. Signal which contains transformed impulses for amplitude ratios 1: 0.18 : 0.02 .

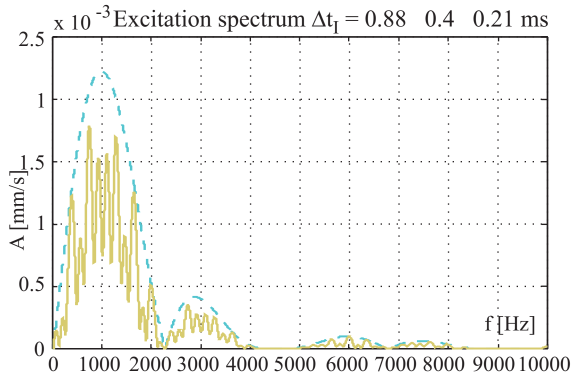


Fig. 11d. Spectrum of the signal which contains transformed impulses for amplitude ratios 1: 0.18 : 0.02 .

★ *number of excitation impulses*, which informs on distance from explosion, combined with explosive charge mass (Fig.12a through 12d)

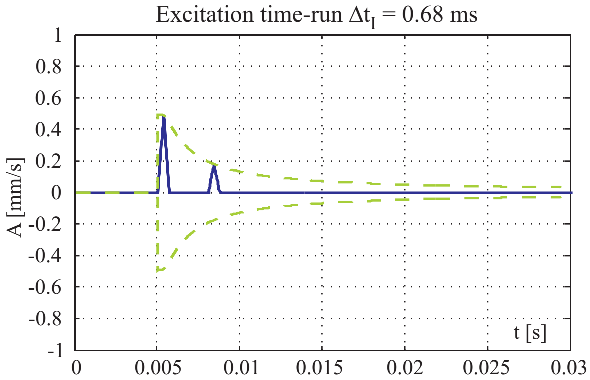


***Fig. 12a.** Run of shock wave model in function of time (two impulses) t = 0.68 ms .*

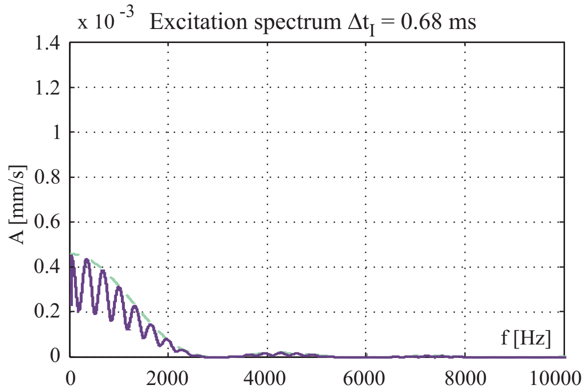


***Fig. 12b.** Spectrum of run of shock wave model in function of time (two impulses) t = 0.68 ms .*

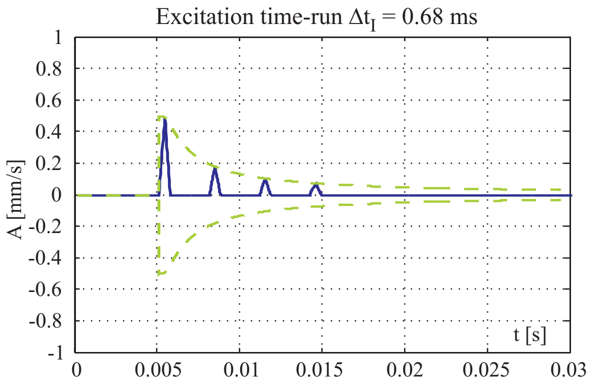


***Fig.12c.** Run of shock wave model in function of time (four impulses) t = 0.68 ms .*

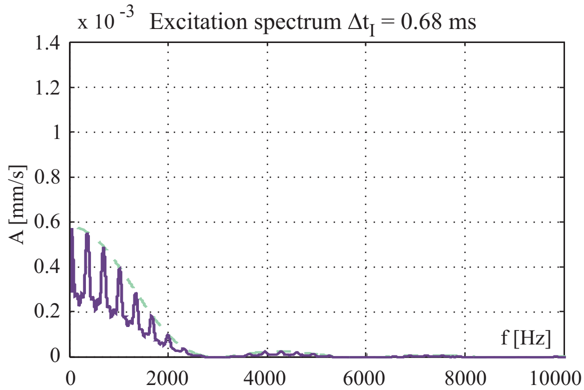


***Fig. 12d.** Spectrum of run of shock wave model in function of time (four impulses) t = 0.68 ms .*

★ *time between successive impulses*, which characterizes explosive charge mass (Fig 13a through 13d)

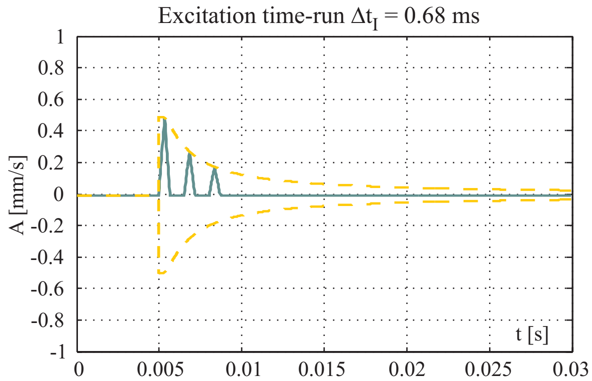


***Fig. 13a.** Run of shock wave model in function of time with taking into account effect of three successive gas bubbles (short time) .*

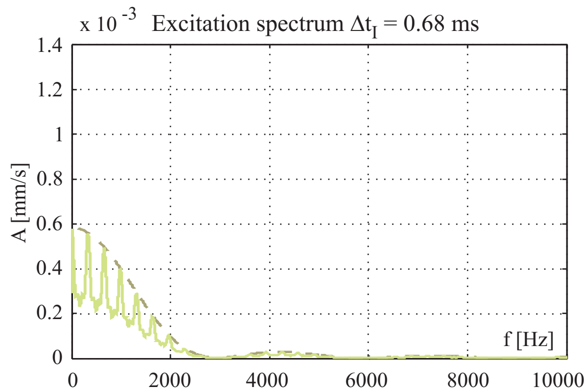


***Fig. 13b.** Spectrum of shock wave with three successive impacts of gas bubbles (short time) .*

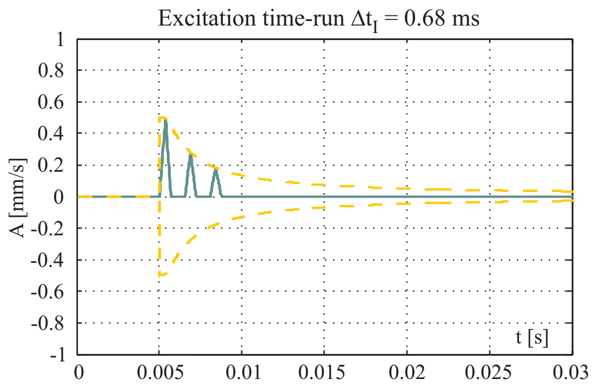


***Fig. 13c.** Run of shock wave model in function of time with taking into account effect of three successive gas bubbles (four times longer duration time) .*

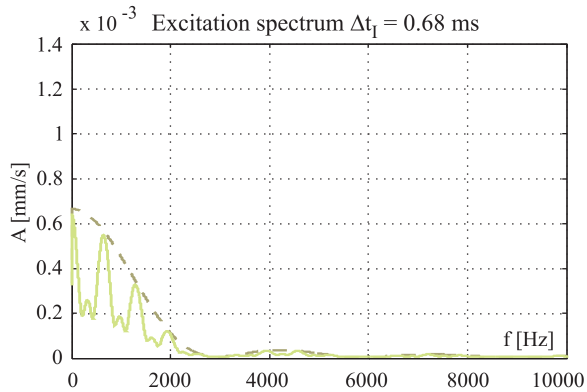


***Fig. 13d.** Spectrum of shock wave with three successive impacts of gas bubbles (four times longer duration time) .*

The modelling of run of hydrodynamic pressure changes due to underwater explosion, in the direction normal to tangent plane at any point of hull, can make it possible to identify the quantities which characterize underwater explosion. The formulae for maximum shock wave pressure and pressure values in function of time, presented in the literature [2], do not take into consideration all the components whose values are significant for assessing degree of hazard to propeller shaft operation. As observed in Fig.4 every non-disturbed underwater explosion (by shallow water, bed structure, water non-homogeneity etc) consists of at least three shock wave impacts resulting from gas bubble pulsation. In practice the form of shock wave is limited by the following factors :

- if the distance between explosion epicentre and ship hull is small then only one time-extended impulse impact may be recorded
- if the distance between explosion epicentre and ship hull is large then impacts of the second or successive pulsations are contained within measurement background and their degree of danger to propeller shaft operation is negligible (for unlimited water area it depends on explosive charge mass and distance).

The possible recording of measured shock wave pressure and accelerations on intermediate and propeller shaft bearings enables to identify some explosion parameters hence also hazards to power transmission system. Analysing the run of underwater shock wave pressure one is able to assume its time-dependent function (Fig.14a and 14b) :

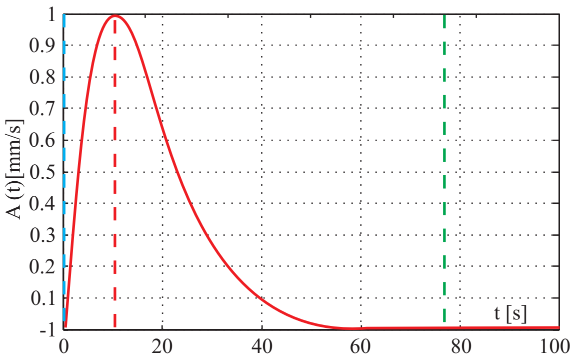


Fig. 14a. *Example of the function form for b = 1.5, c = -0.15 and k = 1*

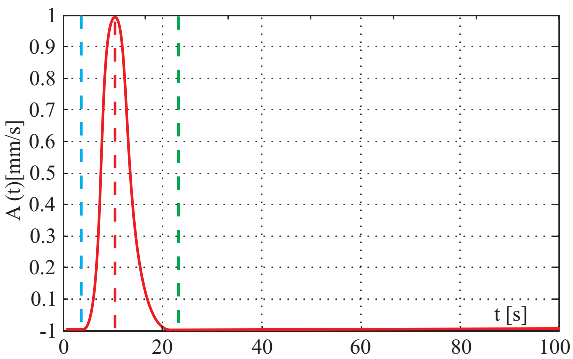


Fig. 14b. *Example of the function form for b = 1.5, c = -0.15 and k = 10*

$$A = at^{kb} \cdot e^{kct} \text{ (normalized by a)} \qquad (13)$$

For the assumed mathematical model of the first shock wave impulse the run of vibration accelerations recorded on ship hull - for the example function given in Fig. 14a - can be presented as shown in Fig.15a and 15b.

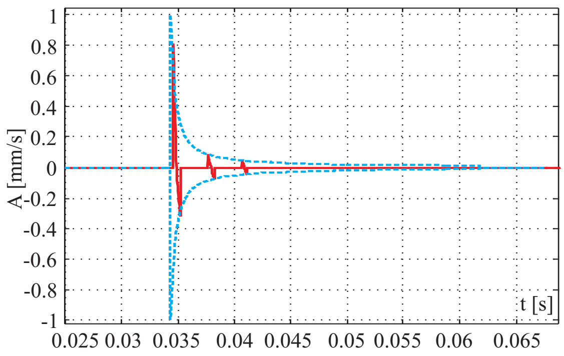


Fig. 15a. *Run of the assumed vibration acceleration model*

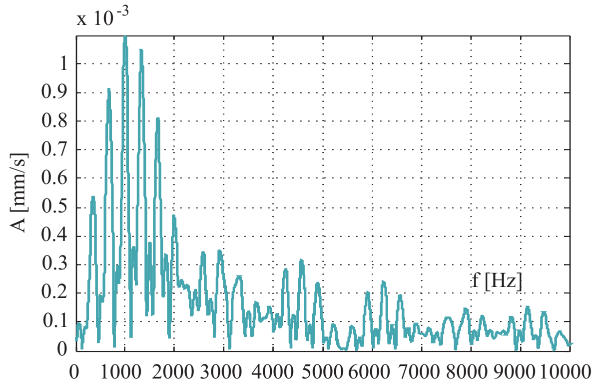


Fig. 15b. *Spectrum of the assumed vibration acceleration model*

FINAL CONCLUSIONS

- It's common knowledge that failure frequency is the most hazardous factor in marine industry, just after aeronautics. Dynamic reactions which occur on ships in service at sea are rarely able to produce wear sufficient to cause a failure.
- The possible application of an on-line monitoring system of vibration parameters of the propulsion system of mine hunter makes it possible to perform the typical technical diagnostic tests of torque transmission system and to identify possible plastic deformations of hull plating as a result of underwater explosion.
- The modelling of impulse impact form and next its identification makes it possible additionally :
  - to identify explosion power by using an analysis of the first vibration impulse amplitude and its duration time
  - to identify distance from explosion epicentre (hence a degree of hazard) by analysing signal's damping
  - to identify a kind of explosion and even characteristic features of type of used mine
  - to select dynamic characteristics of a measuring system which has to comply with requirements for typical technical diagnostics and for a hazard identification system
  - to identify elastic or plastic deformation of shaft line by using spectral assessment of its characteristic features from before and after underwater explosion.
- The presented results of modelling related to the performed experimental test do not make it possible - due to strongly non-linear character of interactions occurring in sea environment – to assign unambiguously the modelled signal features to those of the recorded ones during the real test.

- Successive experimental tests will make it possible to verify features of the signals assumed for the analysis, to be able to build reliable models.
- The wide range of stochastic dynamic loads acting on ships during its life-time makes that in the nearest future the application of on-line diagnostic techniques to ship propulsion systems, based on analysing vibration signals, will constitute an obvious tactical and technical necessity.

NOMENCLATURE

M_1 – driving torque
M_2 – anti -torque
m – mass
e – shafts axis displacement
J – moment of inertia
h, v – coordinates
Ω, ω – angular velocity
k – stiffness

BIBLIOGRAPHY

1. Cempel Cz., Tomaszewski F. (Ed.): *Diagnostics of machines. General principles. Examples of applications* (in Polish), Publ. MCNEM, Radom 1992
2. Cole R. H.: *Underwater Explosions*. Princeton University Press, Princeton 1948
3. Cudny K., Powierża Z.: *Selected problems of shock resistance of ships* (in Polish) Publ. Polish Naval University, Gdynia 1987.
4. Dąbrowski Z.: *Machine shafts* (in Polish), State Scientific Publishing House (PWN), Warszawa 1999
5. Dietrych I, Kocańda S. Korewa W: *Essentials of machine building* (in Polish). Scientific Technical Publishing House (WNT), Warszawa 1974
6. Gosiewski Z., Muszyńska A.: *Dynamics of rotary machines* (in Polish), Publ. High School on Engineering (WSI), Koszalin 1992
7. Kaliski S.: *Vibrations and waves in solids* (in Polish), State Scientific Publishing House (PWN), Warszawa 1966
8. Klatka N.: *Construction of passive defense devices*, *Part Ia and Ib* (in Polish), Publ. **WSMW** 1985
9. Rich H. L., *Effects of underwater explosions on shipboard equipment*, Bureau of Ships Journal, vol. 8, no. 4, 1959
10. www.martec.com

CONTACT WITH THE AUTHOR

Andrzej Grządziela, D.Sc., Eng.
Mechanic-Electric Faculty,
Polish Naval University
Śmidowicza 69
81-103 Gdynia, POLAND
e-mail : AGrza@amw.gdynia.pl

Conference

SymSO 2006

27th Symposium on Ship Power Plants

On 16-17 November 2006 27th Symposium on Ship Power Plants was held in Szczecin. It was organized by Division of Ship Engines and Power Plants, Department of Heat Machines and Ship Power Plants, Maritime Technology Faculty, Szczecin University of Technology. Scientific patronage over the Symposium was taken by : the Marine Technology Unit of the Transport Technical Means Section, Transport Committee, Polish Academy of Sciences (PAN) and the West- Pomeranian Regional Group of the Section on Exploitation Foundations, Machine Building Committee, PAN. Arrangement of 27th Symposium on Ship Power Plants has been included into the program of celebration of 15th Anniversary of the Maritime Technology Faculty as well as 60th Anniversary of Szczecin University of Technology, commencing in 2006.

Such symposia have been organized each year by similar Divisions, Departments and Institutes which educates in designing, construction and operation of ship power plants, being organizational units of : Polish Naval University of Gdynia, Gdańsk University of Technology, Gdynia Maritime University, Maritime University of Szczecin and Szczecin University of Technology.

The Symposia are aimed at forming a forum for exchange of scientific technical information and experience among scientific workers and persons employed in maritime economy enterprises and institutions in the area of the designing, manufacturing and operating of ship power plants and their devices, as well as of marine environment protection and for tightening relations between science and practice.

The organizers had honour to host – apart from representatives of the above mentioned universities – also experts in the field of industry and maritime economy. It was representatives of H. Cegielski Works in Poznań, MAN Diesel A/S, Wartsila Polska Co Ltd, Alfa-Laval Polska Co Ltd, PBP Enamor, Bosch Rexroth Co Ltd, as well as Szczecin Nowa Shipyard Co Ltd.

During two-day proceedings 20 papers were presented in 7 sessions and another twenty in poster presentations. Themes of the papers were assigned to the following groups : **Design**, **Diagnostics and Operation**, **and Environmental Protection and Safety**.

The shares of the particular scientific research centres in preparation of the papers were as follows :

☆ Polish Naval University	8
☆ Gdańsk University of Technology	9
☆ Szczecin University of Technology	5
☆ Gdynia Maritime University	5
☆ Szczecin Maritime University	12
☆ Polish Register of Shipping	1

The next, 28th Symposium on Ship Power Plants will be organized by Gdynia Maritime University and held in November 2007.

Approximation of the index for assessing ship sea-keeping performance on the basis of ship design parameters

Tomasz Cepowski
Szczecin Maritime University

ABSTRACT

This paper presents a new approach which makes it possible to take into account sea-keeping qualities of ship in the preliminary stage of its design. The presented concept is based on representing ship's behaviour in waves by means of the so called operational effectiveness index. Presented values of the index were calculated for a broad range of design parameters. On this basis were elaborated analytical functions which approximate the index depending on ship design parameters. Also, example approximations of the index calculated by using artificial neural networks, are attached. The presented approach may find application to ship preliminary design problems as well as in ship service stage to assess sea-keeping performance of a ship before its departure to sea.

Keywords : sea-keeping qualities, sea-keeping performance index, approximation, ship design parameters, artificial neural networks, rolling, slamming, green-water shipping

INTRODUCTION

In ship design process an optimum solution which satisfies assumed economical criteria and technical limitations, is searched for. The technical limitations contain a.o. performance of ship in rough seas, the called sea-keeping qualities. However in the preliminary design stage to take into account the whole range of ship sea-keeping qualities is very difficult and rather inaccurate when using current calculation methods. It results from a few problems the most important of them are the following :

- in the preliminary design stage values of the parameters which significantly influence sea-keeping performance of designed ship are not yet known (they result from ship hull form and mass distribution which are unknown in that stage of designing)
- simple accurate relationships between design parameters and sea-keeping qualities of ship are unknown
- in the preliminary design stage, assessment of sea-keeping performance of ship is of a descriptive character, left at designer's discretion and hence imprecise as usual.

In the papers [6 , 7] was proposed a method of approximation of selected sea-keeping qualities based on ship design parameters and represented by means of amplitude-phase characteristics of ship in regular waves.

On the basis of such approximations it is possible to determine statistical quantities of sea-keeping qualities of ship in rough waves and the operational effectiveness index (acc. [2 , 5]) which enables to assess quantitatively sea-keeping performance of a given designed ship. The operational effectiveness index $\mathbf{E_T}$ usually expresses probability of the event that ship response in given wave conditions will not exceed an assumed level. Hence the index $\mathbf{E_T}$ takes values from the interval of 0 to 1. The higher the index value the better predicted sea-keeping performance. In [8] is presented an example of application of the above mentioned index (and of the approximation acc. [6 , 7]) to selecting optimum design of ship regarding its sea-keeping performance.

The approximations presented in [6 , 7] make it possible to simply and accurately determine transfer functions of selected sea-keeping qualities. However, the assessing of the sea-keeping qualities on the basis of the operational effectiveness index is associated with the necessity of calculation of their statistical values for ship in irregular waves and probability of occurrence of the wave parameters for which ship's motions exceed an assumed level. The general algorithm of calculation procedure of the index $\mathbf{E_T}$, based on the transfer functions of sea-keeping qualities is presented in Fig.1. The using of the above mentioned algorithm for calculation of the operational effectiveness index is associated with necessity of carrying out many iterations hence the approach can not be applied to multi- criterion optimization methods as the target function as its form is then too much complex. Moreover, to calculate $\mathbf{E_T}$ index value, data on statistical distributions of wave parameters in a given sea area or shipping route should be known, that excludes the method from application to fast assessing a given ship design variant.

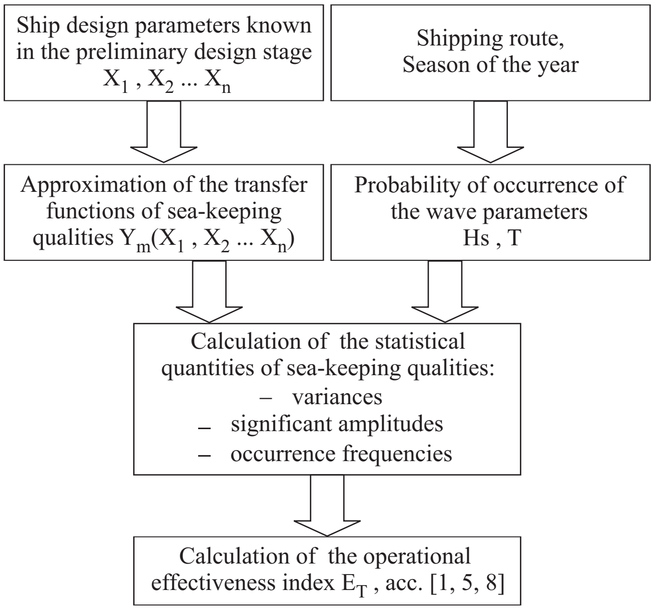


***Fig. 1.** Schematic diagram of calculation of the operational effectiveness index of sea-keeping qualities, E_T . **Notation** : $X_1 , X_2 \dots X_n$ – ship design parameters, **Hs** – significant wave height, **T** – characteristic wave period, Y_m – transfer function of sea-keeping qualities.*

CONCEPT

For the above mentioned reasons in the investigations in question another concept of determining the index $\mathbf{E_T}$ was taken into consideration. To simplify the method described in Fig.1, the approximation of transfer functions of sea-keeping qualities was replaced by the approximation of the index $\mathbf{E_T}$ depending on the ship design parameters $\mathbf{X_1}$, $\mathbf{X_2} \dots \mathbf{X_n}$ on a given shipping route. The schematic diagram of the concept is presented in Fig.2.

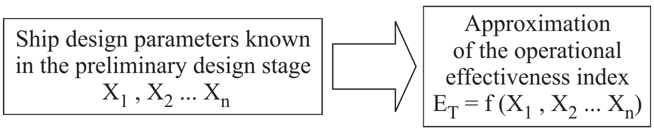


***Fig. 2.** Schematic diagram of approximation of the sea-keeping qualities index E_T on a given shipping route, where : $X_1 , X_2 \dots X_n$ – ship design parameters, **f** – searched for approximation function*

In such case the effectiveness index $\mathbf{E_T}$ on a given shipping route can be approximated in accordance with the formula (1) :

$$E_T = f (X_1 , X_2 \dots X_n) \tag{1}$$

where :

$X_1 , X_2 \dots X_n$ – ship design parameters
f – searched for approximation function.

On the basis of the assumed concept the approximation function **f** may concern all selected sea-keeping qualities or only a given one (e.g. rolling or slamming). Values of the index $\mathbf{E_T}$, approximated with the use of the function **f** , can be related to :

- a given shipping route or region of operation
- a given season or time interval
- a selected type of ships or their group
- an assumed load condition (ballast, full load)
- assumed criterion values of sea-keeping qualities.

The approximation function **f** can be determined on the basis of the set of model values of the ship design parameters $\mathbf{X_1}$, $\mathbf{X_2} \dots \mathbf{X_n}$ and values of the effectiveness index $\mathbf{E_T}$ calculated by using exact methods. The above mentioned approximations can be elaborated by using statistical methods. An example of application of the artificial neural networks to determine the approximation function **f** is presented below.

Application of the artificial neural networks to approximate the operational effectiveness index E_T

The function for approximating the ship operational effectiveness index $\mathbf{E_T}$ can be determined according to the formula :

$$X \xrightarrow{f} Y \tag{2}$$

where :

X – set of assumed ship design parameters
Y – set of values of the ship operational effectiveness index $\mathbf{E_T}$ calculated by means of exact methods
f – searched for analytical function in the form of artificial neural network, intended for the approximating of the index $\mathbf{E_T}$.

It was preliminarily assumed that the investigations in question will concern approximation of the sea-keeping qualities which have the most detrimental influence on ship's safety, to which – in accordance with [2 , 5] – the following ones belong : rolling, slamming, green water shipping onto the deck, propeller emerging, vertical accelerations at bow and bridg, and pitching. And, successive investigations showed that for the considered series of ships in assumed wave conditions vertical accelerations and pitching impair ship's safety to a small extent only. Hence only the values of the operational effectiveness indices which concern the following qualities, were taken into account as the elements of the set **Y** in the equation (2) :

- rolling – (E_{Troll})
- slamming – (E_{Tslam})
- green water shipping – (E_{Tgreen})
- propeller emerging – (E_{Tprop})
- all the above specified sea-keeping qualities in total – (E_T).

And, as the elements of the set **X** in the equation (2) the ship design parameters which significantly influence the above mentioned sea-keeping qualities, were taken into account. In compliance with the literature sources [1, 3, 5] the following ship's parameters were assumed :

- L – length
- B – breadth
- Cb – block coefficient of immersed part of ship hull
- GM_0 – initial transverse metacentric height
- T – draught.

The process of searching for the best network consisted of the following steps :

- determination of the best network structure by using genetic algorithms
- teaching the network
- testing the network
- assessing approximation accuracy of the network on the basis of test data.

For teaching the neural networks at most 50 % of all the data was utilized not to result in over-teaching the networks.

For the assessing of approximation accuracy errors in teaching and testing, which can be determined from Eq. (3), were used :

$$RMS = \sqrt{\frac{(E_{Tw} - E_T)^2}{n}} \tag{3}$$

where :

RMS – value of error
E_{Tw} – model (reference) values used in teaching or testing the neural network
E_T – values calculated with the use of the neural network
n – number of records.

Model values

As model values the operational effectiveness indices calculated for the North Atlantic in winter were assumed. Values of the indices were calculated for the series of container carriers having the design parameters specified in Tab.1.

Their sea-keeping qualities were calculated by using exact numerical methods with the help of **GRIM** [1] software. In Fig. 3 – 5 are presented tests of accuracy of the GRIM software as compared with the results obtained by recognized scientific centres (measurements performed in Wageningen model basin, calculations by Delft calculations by the **WARES** software, Ship Design and Research Centre, Gdańsk). The presented comparison shows very high accuracy and conformity of the calculations performed with the help of GRIM software.

Values of the operational effectiveness indices were calculated in accordance with the algorithm presented in Fig.1 complying with [2, 5, 8]. Criterion quantities for sea-keeping qualities were assumed according to [2]. Calculations of secondary effects of ship oscillation motions were performed for the points indicated in Fig. 6.

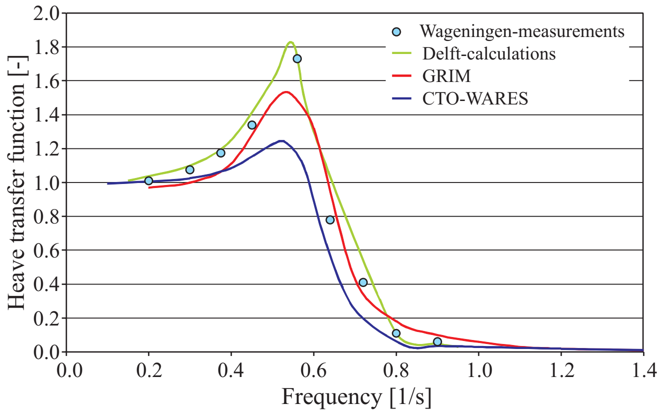


Fig. 3. *Comparison of heave transfer functions : β_w = 90°, V = 0 kn ; 200 000 dwt tanker of the length L = 310 m, breadth B = 47.17 m, draught T = 18.9 m [3]*

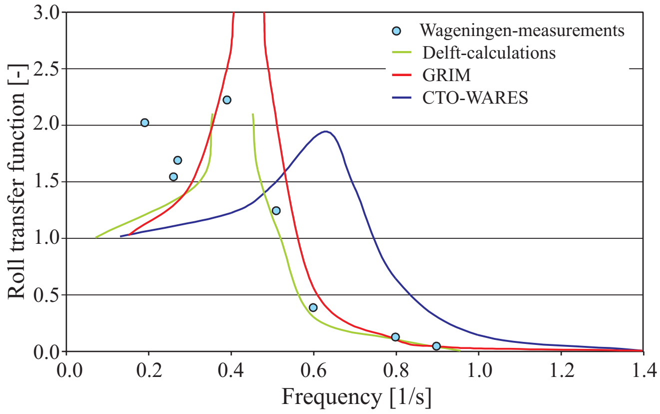


Fig. 4. *Comparison of roll transfer functions : β_w = 90°, V = 0 kn ; 200 000 dwt tanker of the length L = 310 m, breadth B = 47.17 m, draught T = 18.9 m [3]*

Tab. 1. *Model data set :*

✶ ***the ship design parameters*** *: **L** – length, **B** – breadth, **Cb** – block coefficient of immersed part of hull, GM_0 – initial transverse metacentric height, **T** – draught;*
✶ ***the operational effectiveness indices*** *: E_T – for all sea-keeping qualities, E_{Troll} – for rolling, E_{Tslam} – for slamming, $E_{Tprop.}$ – for propeller emerging, E_{Tgreen} – for green water shipping onto the deck.*

Variant	L [m]	B [m]	Cb [-]	GM_0 [m]	T [m]	E_T	E_{Troll}	E_{Tslam}	$E_{Tprop.}$	E_{Tgreen}
1	144.6	24.1	0.67	2	8.6	0.87	0.98	0.89	0.87	0.77
2	152.8	23.5	0.71	1.85	7.8	0.85	0.89	0.89	0.85	0.73
3	161	23	0.75	2.41	7.2	0.81	0.88	0.88	0.82	0.7
4	170.3	22.7	0.78	2.92	6.7	0.77	0.94	0.87	0.8	0.67
5	184.2	30.7	0.78	3	10.2	0.94	0.88	0.94	0.91	0.86
6	192.4	29.6	0.75	3	10.6	0.94	0.95	0.94	0.91	0.86
7	219.1	31.3	0.71	3.98	9.2	0.9	0.97	0.91	0.89	0.8
8	229.5	30.6	0.67	3.05	9.6	0.9	0.98	0.91	0.88	0.8
9	225	37.5	0.71	3.88	11.7	0.97	1	0.96	0.94	0.91
10	246.4	37.9	0.67	4.81	11.1	0.96	0.99	0.95	0.93	0.9
11	236.6	33.8	0.78	3.5	11.3	0.97	0.95	0.96	0.93	0.91
12	245.3	32.7	0.75	3	11.7	0.93	1	0.96	0.93	0.91
13	249.6	41.6	0.75	5.46	12.2	0.99	0.99	0.98	0.97	0.95
14	254.2	39.1	0.78	4.15	12.2	0.9	0.86	0.98	0.97	0.95
15	275.1	39.3	0.67	4.5	13.1	0.99	1	0.98	0.97	0.95
16	276	36.8	0.71	4.5	13.1	0.99	1	0.98	0.97	0.95

[1] The GRIM software was elaborated by the Department of Ocean Engineering and Marine Systems , Faculty of Maritime Technology , Szczecin University of Technology

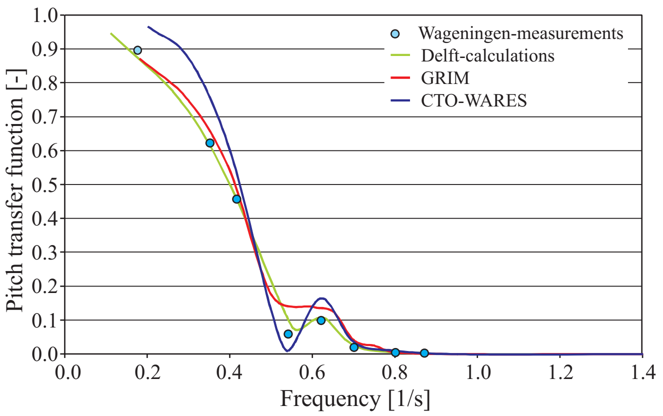


*Fig. 5. Comparison of pitch transfer functions : β_w = 180° , V = 0 kn ; 200 000 dwt tanker of the length **L** = 310 m, breadth **B** = 47.17 m, draught **T** = 18.9 m [3]*

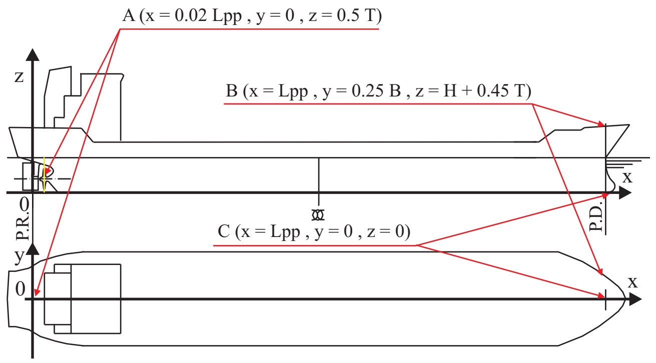


*Fig. 6. Coordinates of the points at which values of the selected sea-keeping qualities were calculated : **A** – for propeller emerging, **B** – for green water shipping onto bow deck, **C** – for slamming , where : **Lpp** – ship length between perpendiculars, **B** – ship breadth, **T** – ship draught, **H** – ship depth.*

Test data

The testing of approximation accuracy of the function $\mathbf{f_m}$ was carried out on the basis of the following data :

➡ those contained within the range of model data (interpolation) – Tab. 2
➡ those from outside the range of model data (extrapolation) – Tab. 3.

The sea-keeping qualities were calculated by using the methods described in [6, 7], whereas values of the indices presented in Tab 2 and 3 – in the same way as for the model data.

***Tab. 2.** Test data set for interpolation :*

✸ ***the ship design parameters** : **L** – length, **B** – breadth, **Cb** – block coefficient of immersed part of hull, $\mathbf{GM_0}$ – initial transverse metacentric height, **T** – draught;*
✸ ***the operational effectiveness indices** : $\mathbf{E_T}$ – for all sea-keeping qualities, $\mathbf{E_{Troll}}$ – for rolling, $\mathbf{E_{Tslam}}$ – for slamming, $\mathbf{E_{Tprop.}}$ – for propeller emerging, $\mathbf{E_{Tgreen}}$ – for green water shipping onto the deck.*

Variant	L [m]	B [m]	Cb [-]	GM_0 [m]	T [m]	E_T	E_{Troll}	E_{Tslam}	$E_{Tprop.}$	E_{Tgreen}
1	160.75	26.79	0.78	2.00	8.93	0.79	0.88	0.94	0.91	0.86
2	167.89	25.83	0.75	2.00	9.22	0.83	0.95	0.94	0.91	0.86
3	191.65	27.38	0.71	2.00	8.05	0.79	0.97	0.91	0.89	0.8
4	200.50	26.73	0.67	2.00	8.35	0.79	0.98	0.91	0.88	0.8
5	213.52	35.59	0.71	3.00	11.12	0.90	1.00	0.96	0.94	0.91
6	234.31	36.05	0.67	3.00	10.60	0.88	0.99	0.95	0.93	0.9
7	224.45	32.06	0.78	3.00	10.69	0.88	0.95	0.96	0.93	0.91
8	232.70	31.03	0.75	3.00	11.08	0.89	1.00	0.96	0.93	0.91

***Tab. 3.** Test data set for extrapolation :*

✸ ***the ship design parameters** : **L** – length, **B** – breadth, **Cb** – block coefficient of immersed part of hull, $\mathbf{GM_0}$ – initial transverse metacentric height, **T** – draught;*
✸ ***the operational effectiveness indices** : $\mathbf{E_T}$ – for all sea-keeping qualities, $\mathbf{E_{Troll}}$ – for rolling, $\mathbf{E_{Tslam}}$ – for slamming, $\mathbf{E_{Tprop.}}$ – for propeller emerging, $\mathbf{E_{Tgreen}}$ – for green water shipping onto the deck.*

Variant	L [m]	B [m]	Cb [-]	GM_0 [m]	T [m]	E_T	E_{Troll}	E_{Tslam}	$E_{Tprop.}$	E_{Tgreen}
1	114.59	19.10	0.67	1.00	6.82	0.75	0.98	0.89	0.87	0.77
2	121.31	18.66	0.71	1.00	6.22	0.69	0.89	0.89	0.85	0.73
3	127.87	18.27	0.75	1.00	5.71	0.65	0.88	0.88	0.82	0.70
4	134.85	17.98	0.78	1.00	5.29	0.65	0.94	0.87	0.80	0.67
5	257.81	42.97	0.75	4.50	12.64	0.94	0.99	0.98	0.97	0.95
6	263.04	40.47	0.78	4.50	12.65	0.85	0.86	0.98	0.97	0.95
7	284.54	40.65	0.67	4.50	13.55	0.94	1.00	0.98	0.97	0.95
8	285.59	38.08	0.71	4.50	13.60	0.94	1.00	0.98	0.97	0.95

Approximation

The **MLP** network of 5x1x1 structure (Fig.7), characterized by the statistics described in Tab.4, appeared the best for approximating the operational effectiveness index $\mathbf{E_T}$ for all sea-keeping qualities.

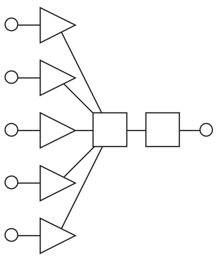


*Fig. 7. Structure of **MLP** artificial neural network for approximating the index E_T .*

Tab. 4. Statistics of regression issues for the neural network approximating the index E_T .

	teaching	testing (interpolation)	testing (extrapolation)
Coefficient of correlation R	0.99	0.94	0.98
RMS error	0.017	0.018	0.021

The searched for function approximating the index $\mathbf{E_T}$, elaborated by using the above presented neural network, is presented by means of Eq. (4), in an analytical form :

$$E_T = \frac{\left(\dfrac{2.5614}{1 + e^{-\left(\begin{bmatrix} 0.0088 \cdot L - 1.4110 \\ 0.0614 \cdot B - 1.4110 \\ 9.0909 \cdot Cb - 6.0909 \\ 0.4785 \cdot GM_0 - 1.1531 \\ 5.5556 \cdot T - 1.2203 \end{bmatrix} \times \begin{bmatrix} -0.6815 & -0.0286 & -0.1163 & 0.3014 & 2.3751 \end{bmatrix} - 0.1378 \right)}} - 0.1378 \right) + 4.1667}{5.5556} \tag{4}$$

where :

L – ship length, B – ship breadth, Cb – block coefficient of immersed part of hull, GM_0 – initial transverse metacentric height, T – ship draught.

As results from Tab. 4 and Fig. 8 and 9, the function described by Eq. (4) shows rather high accuracy both in the range of interpolation and extrapolation. Moreover, the function (4) has a very simple structure, that makes it possible to use it in the multi-criterion optimization methods based on genetic algorithms.

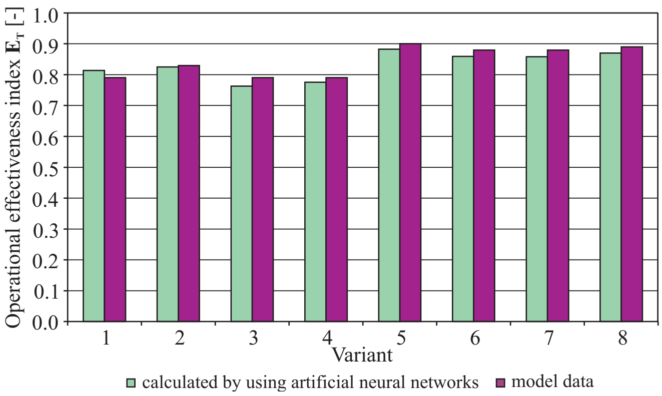


Fig. 8. Interpolation of the operational effectiveness index E_T for test variants given in Tab. 2.

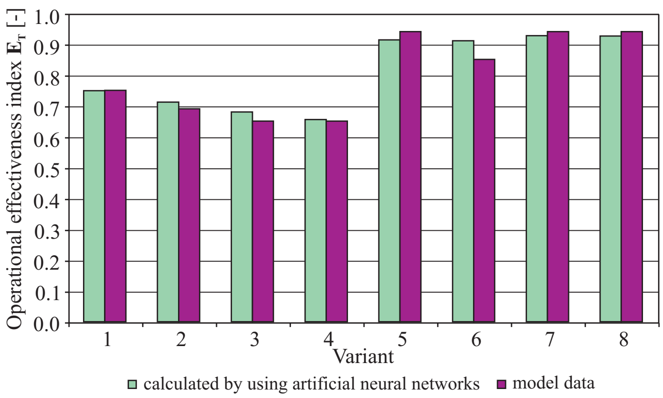


Fig. 9. Extrapolation of the operational effectiveness index E_T for test variants given in Tab. 3.

In Tab. 5 are described the artificial neural networks approximating the remaining operational effectiveness indices of sea-keeping qualities such as :

- ✦ rolling (E_{Troll})
- ✦ slamming (E_{Tslam})
- ✦ green water shipping onto the deck (E_{Tgreen})
- ✦ propeller emerging (E_{Tprop}).

The values of statistical parameters speak for relatively high approximation accuracy of the networks in question both in the range of interpolation and extrapolation. The neural networks are also of a relatively simple structure.

*Tab. 5. Types, structure and statistical parameters of the neural networks for approximating the operational effectiveness indices : E_{Troll} – for rolling, E_{Tslam} – for slamming, $E_{Tprop.}$ – for propeller emerging, E_{Tgreen} – for green water shipping, where : **R** – correlation, **RMS** – network's error .*

Parameter	Type of network	Structure	Parameter	Teaching	Testing	
					Interpolation	Extrapolation
E_{Troll}	MLP	5x8x1	R	0.880	0.850	0.620
			RMS	0.015	0.031	0.035
E_{Tslam}	MLP	4x2x1	R	0.990	0.970	0.990
			RMS	0.003	0.016	0.013
E_{Tprop}	MLP	4x4x1	R	0.990	0.950	0.990
			RMS	0.005	0.021	0.019
E_{Tgreen}	MLP	4x3x1	R	0.990	0.980	0.990
			RMS	0.008	0.044	0.042

SUMMARY

- In this paper was presented a concept of approximation process of the operational effectiveness index by making use of basic ship design parameters. The index - according to [1, 5, 8] - can be applied to assess ship sea-keeping performance both in the stage of ship preliminary design and in service.

- Example approximations of the index by using the artificial neural networks are also attached. The presented approximations make it possible to determine the operational effectiveness indices for the following sea-keeping qualities : rolling, green water shipping onto the deck, slamming, propeller emerging, and all the above given qualities together, on the basis of the following ship design parameters : length, breadth, design draught, block coefficient of immersed part of hull, initial transverse metacentric height.
- The described approximations can be used to select a ship design which is characterized by most favourable sea-keeping qualities on a given shipping route described by statistical parameters of waves, at given criterion values for sea-keeping performance.
- A way of application of the approximations may be the same as that described in [8].
- The approximations make it possible to calculate the operational effectiveness index fairly accurately and in a much simpler way than in the iterative methods described in [6 , 7].
- Owing to the simple form of the functions approximating particular indices the approximations in question can be used as target functions in multi-criterion optimization methods in the preliminary ship design stage. They can be also used as a simplified method to assess a given design variant from the point of view of sea-keeping performance (without any necessity of determining ship response to wave action and taking into account occurrence probability distribution of wave parameters on a given sea area).
- The presented investigation concept can be also applied to approximation of sea-keeping performance indices, carried out for :
  - a greater group of ships of different types
  - various shipping routes and areas
  - various seasons of the year.
- The approximation method in question may also find its application to the issues associated with ship operation in the phase of voyage planning. In this case ship master could be provided with useful information on sea-keeping performance of his ship on a given shipping route and in a given season, well in advance of the ship's departure, without any necessity of carrying out complex calculations.

NOMECLATURE

B – ship breadth
Cb – block coefficient of immersed part of ship's hull
E_T – operational effectiveness index for all sea-keeping qualities
E_{Tslam} – operational effectiveness index for slamming
E_{Tprop} – operational effectiveness index for propeller emerging
E_{Troll} – operational effectiveness index for rolling
E_{Tgreen} – operational effectiveness index for green-water shipping
GM_0 – initial transverse metacentric height
H – ship depth
L – ship length
MLP – multi-layer perceptron
NSRDC – Naval Ship Research and Development Center Bethesda
R – correlation coefficient
RMS – error of learning or testing the artificial neural network
T – ship draught
V – ship speed
$X_1, X_2 \dots X_n$ – ship design parameters
Y_m – transfer function of sea-keeping qualities
β_w – wave heading angle relative to ship

BIBLIOGRAPHY

1. Bales N.K. : *Optimizing the Sea-keeping Performance of Destroyer – Type Hulls*, David W. Taylor NSRDC, Maryland, USA
2. Karppinen T. : *Criteria for Sea-keeping Performance Predictions*, ESPOO 1987
3. Lloyd A.R.J.M. : *Seakeeping : ship behaviour in rough weather*, Ellis Horwood Limited, England, 1989
4. Pinkster J.A. : *Low Frequency Second-Order Wave Exciting Forces on Floating Structures*, Publication No. 650, Netherlands Ship Model Basin, Wageningen – Netherlands
5. Szelangiewicz T. : *Ship's Operational Effectiveness Factor as Criterion for Cargo Ship Design Estimation*, Marine Technology Transaction, Polish Academy of Sciences, Branch in Gdańsk, Vol. 11, 2000
6. Szelangiewicz T., Cepowski T. : *Application of artificial neural networks to investigation of ship sea-keeping ability*, Part 1, Polish Maritime Research, Vol 8, no 3, 2001
7. Szelangiewicz T., Cepowski T. : *Application of artificial neural networks to investigation of ship sea-keeping ability*, Part 2, Polish Maritime Research, Vol 8, no 4, 2001
8. Szelangiewicz T., Cepowski T. : *An approach to optimization of ship design parameters with accounting for sea-keeping ability,* Polish Maritime Research, Vol.9, No 4, 2002

CONTACT WITH THE AUTHOR

Tomasz Cepowski, D.Sc.,Eng.
Institute of Marine Navigation,
Maritime University of Szczecin
Wały Chrobrego 1/2
70-500 Szczecin, POLAND
e-mail : cepowski@am.szczecin.pl

Miscellanea

Jubilee of 36-th anniversary of CTO activity

Ship Design and Research Centre Co. (CTO S.A.) is a research, design and development institution working for Polish shipbuilding industry, which, due to its multi-year activity, has contributed in maintaining high level of novelty of ships built in Poland. Directions of research and development projects are first of all determined by needs of shipbuilding industry, that guarantees independence of technical ideas in the very important domain of national economy. Implementation of the obtained results of research projects is not limited solely to shipbuilding but also covers areas of other maritime industry and municipal economy, as well as makes it possible to export the gained knowledge and advanced technologies.

CTO takes part in activities of many international societies and organizations, initiated establishment of Polish Forum of Maritime Industries and co-operates in running Polish Technological Platform of Waterborne Transport.

R&D projects are realized by CTO in co-operation with many domestic economy institutions such as shipbuilding and ship-repair yards, ship equipment manufacturing firms, design offices as well as technical universities and other research centres.

Since 1998 CTO has taken part in the EU Outline Programs and realized projects both on one's own and in cooperation with foreign and domestic partners.

Modelling the characteristics of axial compressor of variable flow passage geometry, working in the gas turbine engine system

Paweł Wirkowski
Polish Naval University

ABSTRACT

This paper concerns application of mathematical modelling methods to analyzing gas-dynamic processes in marine gas turbines. Influence of geometry changes in axial compressor flow passage on kinematical air flow characteristics, are presented. The elaborated mathematical model will make it possible to realize – in the future – simulative investigations of gas-dynamic processes taking place in a compressor fitted with controllable guide vanes.

Keywords : marine gas turbine engine, compressor, modelling of characteristics.

INTRODUCTION

An important problem of operation of ship gas turbine engines[1] is to know influence of changeable technical state of engine on its working parameters. Compressor is a unit of gas turbine engine especially sensitive on changes in its technical state during operation [5,6]. The more-or-less-contaminated atmospheric air flowing into the compressor causes a.o. continuously changing form of blade passages, increased roughness of blade surfaces as well as change of mass of compressor rotor. It seriously impacts compressor's operation stability, changes its characteristics as well as performance and efficiency of the entire engine. If the compressor design contains a control system for setting controllable blades of guide vanes (of initial swirl guide vane or/and first stage guide vanes) to make co-operation of all engine's units optimal by continuous correcting the compressor characteristics then the disturbances occurring in work of the system will result in changes in operation of the compressor and engine, of a character similar to those due to rotational speed changes or contaminated blade passages of compressor.

Individual parametric features of every engine in service are identified by using expensive experimental tests conditioned by many constructional and operational limitations. Dynamic development of computer technique applied already to the design stage of engine's units has made it possible to use it also for numerical simulation of changeable technical state processes. Such approach greatly shortens the time necessary for diagnostic tests aimed at determining a set of „defect-symptom" relations, as compared with the time-consuming and expensive investigations carried out on real objects. Having at one's disposal an appropriate computer software one is able to elaborate the models simulating operation of engine's units, verified within an allowable range of static and dynamic loads. The computer software based on mathematical models makes it possible to realize numerical experiments consisting in putting-in real variables and hypothetical technical states of engine.

CHARACTERISTICS OF AXIAL COMPRESSORS WORKING IN ENGINE SYSTEMS

The universal characteristics of compressor (Fig.1), showing the dependence of the compression π_s and efficiency η_s of compressor on the air mass flowing through it, $\dot{m}$, and the rotational speed n, make it possible to determine the most favourable conditions of co-operation of the compressor with other units of the engine under assumption that parameters of the sucked -in air are of the values complying with the so-called ISA standard atmosphere (p_{ot} = 1.013 bar, T_{ot} = 273.15 K, $\varphi = 0\%$). The characteristics serve for the selection of optimum conditions for air flow control and assessment of influence of operational factors on parameters of the compressor.

Fig.2 highlights occurrence of unstable work phenomenon, on which a schematic diagram is presented of flow round a blade of axial stage rotor under motion with the constant rotational speed n, for which a change in the air flow rate $\dot{m}$ is effected. Fig.2a shows the schematic diagram of the flow in the conditions for which the air flow rate corresponds with optimum efficiency of the stage.

The relative velocity vectors w_1, w_2 are parallel to camber line of blade profile, that is conductive to laminar flow through blade passages. The lowering of air flow (Fig.2b) as compared with calculation conditions at the circumferential velocity **u** maintained constant, makes the axial component of absolute

[1] Called also further : „engines"

velocity, c_a , smaller, that results in the increasing of the inlet angle **i** of air flow onto rotor blades. It is conductive to boundary layer separation out of concave surfaces of blades and to generation of whirl zones.

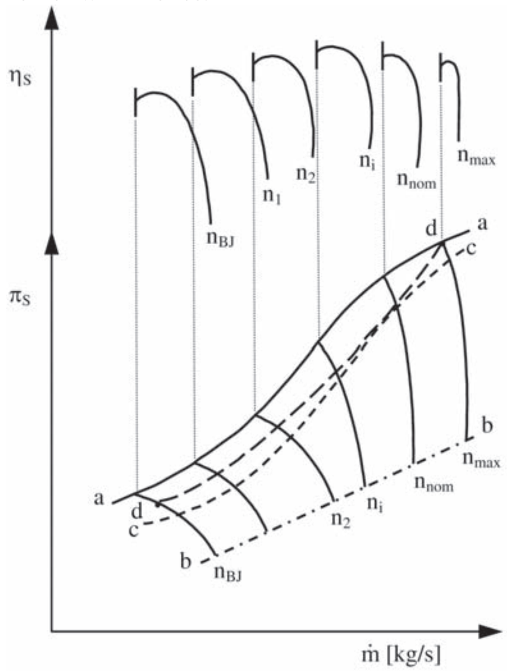


Fig. 1. *Characteristics of axial compressor of gas turbine engine;*
a-a *– stable operation limit,* ***b-b*** *– limit of maximum air flow rates,*
c-c *– line of optimum values of compressor efficiency,*
d-d *– line of compressor-turbine co-operation .*

A similar phenomenon occurs on convex surface of blade (Fig.2c) when air flow rate becomes greater at the circumferential velocity maintained constant. Then the air flow rate $\dot{m}$ takes its maximum values.

At critical values of angles of flow through the produced whirl zones which constitute circumferentially displacing low-pressure zones, a sudden air flow reversal (pumping) towards compressor's inlet can happen, that results in violent flow pulsations transferred onto engine's structure. The phenomenon is detrimental and dangerous because of thermal and mechanical overload of engine's structure. The occurring vibrations of significant amplitudes may cause fatigue cracks in blades [1, 2, 4].

In this connection the compressor should be – within the range of service rotational speeds – so adjusted as to place the line of compressor-network co-operation with certain stable work margin [5, 6, 9].

During engine's operation, rotational speed of rotor, air flow rate and optimum form of kinematics of air flow through stage blade passages, determined by the air flow inlet angle **i** onto the blades, have the greatest impact on compressor's performance and efficiency. The main principle of compressor control during changing its rotational speed or flow rate is to maintain values of the air flow inlet angles **i** close to zero. One of the used methods for control of axial compressor is to change geometry of its flow passage by applying a controllable inlet guide vane or controllable guide vanes of a few first stages of compression [4].

CONTROLLABLE INLET GUIDE VANE

The application of controllable blades of inlet guide vane and guide vanes of particular compressor's stages makes it possible to simultaneously change inlet angles of flow onto blades of rotor rings of the stages by changing the setting angles of the blades of the guide vanes during compressor's rotational speed changing. Fig.3 shows the essence of control of blades of controllable guide vanes by using single compression stage as an example. The situation shown in Fig.3b where velocity directions and values are indexed "1", corresponds with average values of rotational speed service range of compressor's rotor. In this case takes place an average angular setting of blades of guide vane ring, at which the inlet angle of flow onto blades of rotor ring does not generate disturbances in the flow through blade passages. In the case of smaller values of rotational speed of compressor, i.e. appearance of a smaller value of the axial component of absolute velocity, c_{1a}' , it is necessary to decrease the outlet angle of flow from the controllable blades of the guide vane ring (Fig.3a) to such an extent as to maintain the same value of inlet angle of flow onto rotor blades. The analogous situation takes place during operation of compressor at greater values of rotational speed of rotor, for which the axial

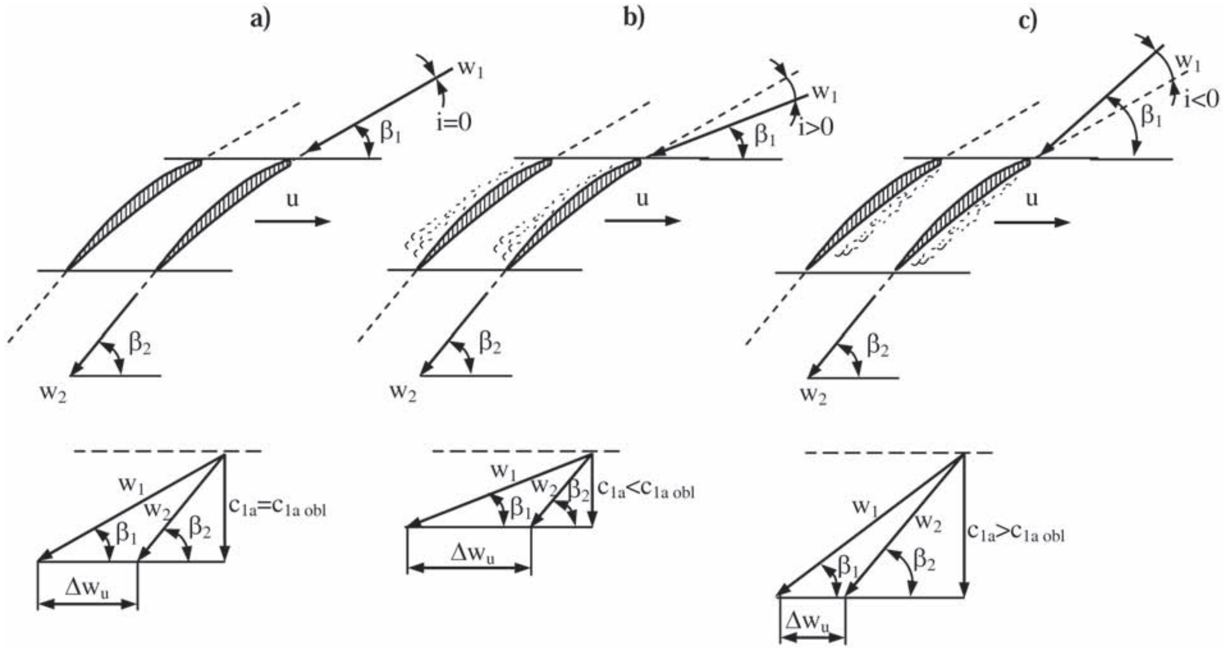


Fig. 2. *Schematic diagram of flow round blades of compressor axial stage ring at constant rotational speed of the rotor and changeable angles of air flow inlet.*
a) *calculation angle of air flow inlet,* ***b)*** *positive angle of air flow inlet of,*
c) *negative angle of air flow inlet .*

component value of absolute velocity, c_{1a}" , increases. Then it is necessary to increase the outlet angle of flow from guide vane blades, Fig.3c, to maintain stable work of compressor hence a constant value of inlet angle of flow onto rotor blades.

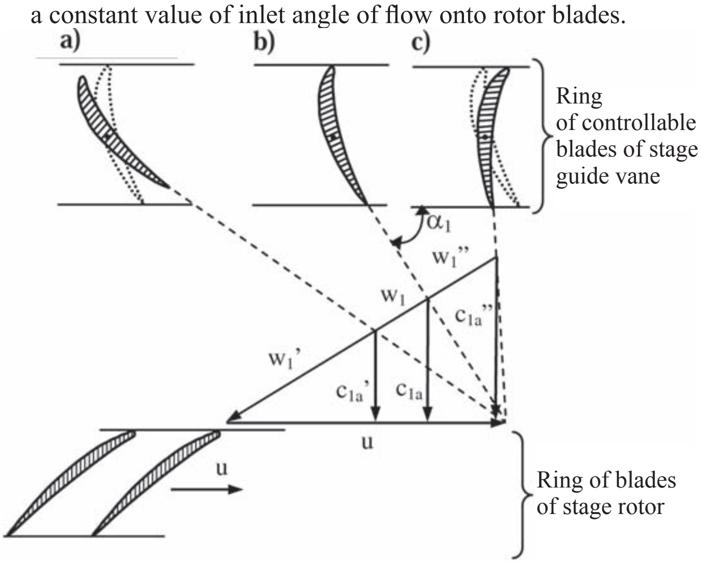


Fig. 3. *Essence of control of compressor's axial stage by changing the setting angle of blades of guide vane rings at changeable air flow velocity;* ***a)*** *decreased axial velocity,* ***b)*** *calculation axial velocity,* ***c)*** *increased axial velocity .*

Application of a control system of geometry of flow passages to gas turbine engine of a given design type significantly influences unsteady processes. In multi-stage axial compressors of compression value exceeding 8 ÷ 10 the design solution which ensures stable operation, is to apply a.o. controllable blades of inlet guide vane [4]. In Fig.4 are presented compressor characteristics of the engine fitted with controllable inlet guide vane, on which the lines of rotor acceleration and deceleration have been plotted.

The lines depicted on the characteristics represent the compressor's operation at three angular settings of inlet guide vane blades, α_{KW} , and the compressor rotational speed, n_1 , maintained constant.

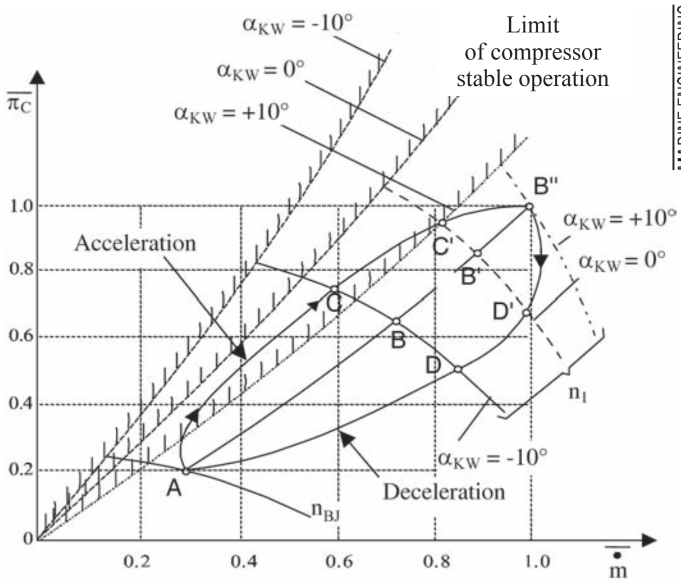


Fig. 4. *Compressor characteristics plotted in the system of relative-value parameters – test stand measurements. Change of compressor operation range resulting from interaction of controllable inlet guide vane [5];*
ABB'B" – points of co-operation of the compressor with network in stable states;
ACC'B" – points of co-operation of the compressor with network during engine acceleration;
B"D'DA – points of co-operation of the compressor with network during engine deceleration .

From the run of the co-operation lines shown on the characteristics it can be concluded that in the engine fitted with controllable inlet guide vane, to control output parameters of its compressor at constant rotational speed of rotor unit, is possible.

The DR76 and DR77 triple-shaft engines operating within COGAG propulsion system installed on naval ships of „Tarantula" class, are equipped with such design solution of inlet guide vane. In Fig.5 is presented the schematic diagram of such engine with indicated control cuts in its flow part.

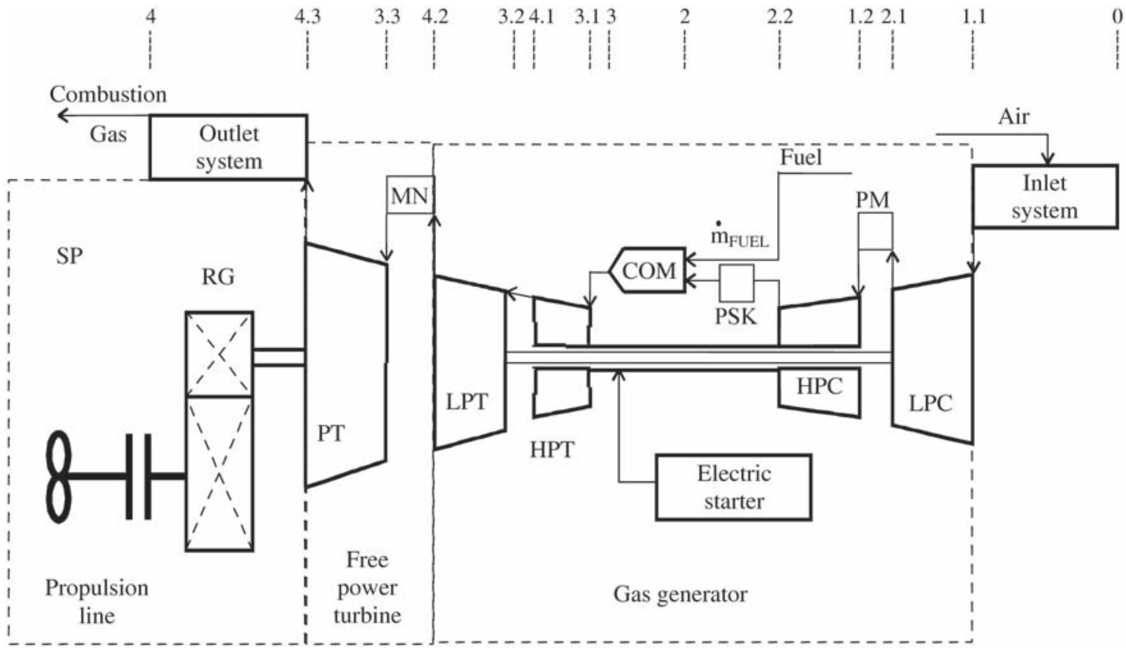


Fig. 5. *Schematic diagram of ship gas turbine engine fitted with separate reversible propulsion turbine [6] .*

CONTROLLABLE GUIDE VANES OF FIRST STAGES OF COMPRESSOR

In modern solutions of currently manufactured marine engines their compressors are fitted with controllable inlet guide vanes and controllable guide vanes of their first stages in order to ensure sufficient stable work margins. Such compressors are characterized by high compression values exceeding 20. Of that type are the objects being a part of e.g. LM 2500 engines used for propelling the frigates of Oliver Hazard Perry class as well as TW-3 aircraft engines applied on Mi helicopters.

A characteristic feature of the sixteen-stage axial compressor of LM 2500 engine is the possibility of changing of setting of the outlet flow angles α_1 of the blades of inlet guide vane and those of its first six stages in function of engine load. The solution prevents from unstable work of the compressor during fast realization of transient processes from one stable state to another. In the case of aircraft version of the engine such solution makes it possible to transit from "low gas" state to "full load" within only five seconds not going beyond stable operation zone.

In Fig.6 are shown elements of the control system of angular setting of blades of particular guide vanes of the compressor of LM 2500 engine.

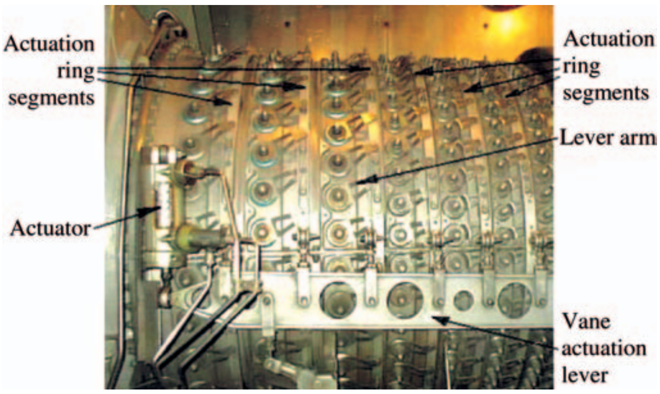


Fig. 6. *Elements of the angular setting control system of blades of particular guide vanes of the LM 2500 engine's compressor*

Fig.7 presents the relation between the outlet flow angle, α_1, of the controllable blade rings of guide vanes and the engine load represented by the rotational speed of gas generator's rotor, n_{zr}, reduced to the normal conditions (acc. ISA) at the inlet to the engine. The load control lever installed on the engine changes its position within the angular range $\lambda = 39° \div -3°$, along with changing the gas generator's rotational speed within the range $n_{zr} = 5000 \div 10000$ rpm, which results in changes of the angular setting of air outlet from blades of inlet guide vane and guide vanes of particular compressor's stages within the range shown in Fig.7.

MATHEMATICAL MODEL OF CHARACTERISTICS OF AXIAL COMPRESSOR WITH CONTROLLABLE BLADES OF INLET GUIDE VANE

The problems of elaboration of sufficiently accurate mathematical models of axial compressors are associated with range of simplifying assumptions which determine accuracy of the numerical modelling of real object [8]. Simulating investigations require a.o. to convert the usual graphical form of compressor characteristics (Fig. 4) into functional one suitable for numerical calculations, i.e. to the following form :

$$\pi_s = f\left(\dot{m}_{zr}, n_{zr}, \alpha_{KW}\right) \tag{1}$$

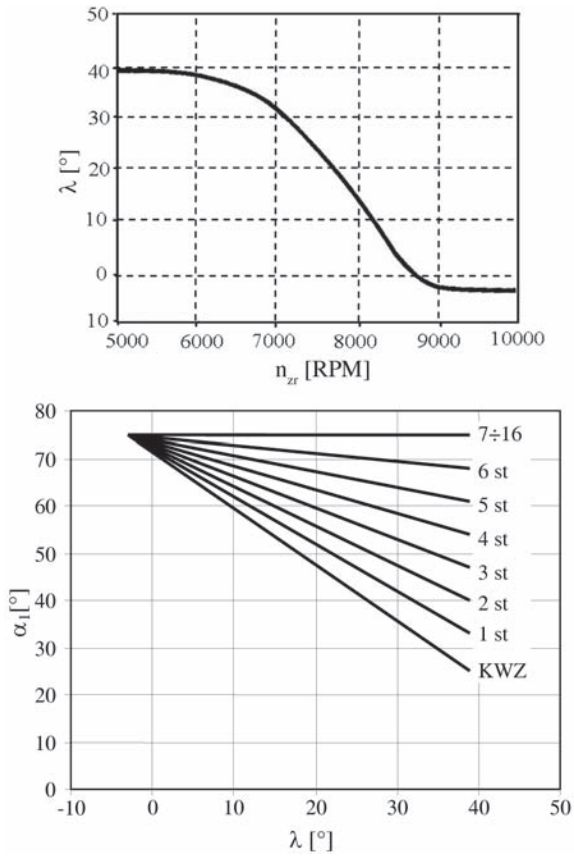


Fig. 7. *Range of changes of the angle of air outlet from blades of rings of controllable guide vanes of compressor, α_1, in function of the rotational speed n_{zr} and the angular setting of load control lever, λ , of the LM 2500 engine [7,8] .*

$$\eta_s = f\left(\dot{m}_{zr}, n_{zr}, \alpha_{KW}\right) \tag{2}$$

The obtaining of an analytical form of the functional relations (1) and (2) which model real characteristics of compressor, at maintained minimum approximation error, is associated with the difficulties resulting from complex form of the functions. In the range of low rotational speeds, isodroms of the characteristics exhibit moderate slopes which correspond with $\pi_s \approx$ idem, whereas in the range of high rotational speeds they show steep sections corresponding with $\dot{m}_{zr} \approx$ idem.

Hence it seems that the least squares method and multidimensional polynomial-based regression [3] can be an effective way for determining an analytical description of axial compressor operation, which guarantees that deviations of the model from reality would be contained within limits of measurement error.

The overall model of compressor is searched for by means of the set of regression equations which approximate its universal characteristics :

$$\pi_s = a_0 + a_1\dot{m}_{zr} + a_2\left(\dot{m}_{zr}\right)^2 + a_3 n_{zr} + a_4\left(n_{zr}\right)^2 + a_5\dot{m}_{zr}n_{zr} + a_6\alpha_{KW} + a_7\left(\alpha_{KW}\right)^2 + a_8\dot{m}_{zr}\alpha_{KW} + a_9 n_{zr}\alpha_{KW} \tag{3}$$

$$\pi_s = b_0 + b_1\dot{m}_{zr} + b_2\left(\dot{m}_{zr}\right)^2 + b_3 n_{zr} + b_4\left(n_{zr}\right)^2 + b_5\dot{m}_{zr}n_{zr} + b_6\alpha_{KW} + b_7\left(\alpha_{KW}\right)^2 + b_8\dot{m}_{zr}\alpha_{KW} + b_9 n_{zr}\alpha_{KW} \tag{4}$$

Values of the regression coefficients a_i , b_i are determined on the basis of the Gauss-Markov theorem when searching for minimum values of the functionals:

$$J_{\pi_s}(a_0,a_1,a_2,a_3,a_4,a_5,a_6,a_7,a_8,a_9)= =\sum_{k=1}^{n}\left(\pi_{sk}-\overline{\pi}_{sk}\right)^2 \tag{5}$$

$$J_{\pi_s}(b_0,b_1,b_2,b_3,b_4,b_5,b_6,b_7,b_8,b_9)= =\sum_{k=1}^{n}\left(\eta_{sk}-\overline{\eta}_{sk}\right)^2 \tag{6}$$

They constitute sums of squares of deviations of the values obtained by using the model π_s , η_s from real values $\overline{\pi}_s$, $\overline{\eta}_s$. Adequacy of matching the mathematical description of characteristics of a given compressor and its real run can be assessed on the basis of the following factors :

a) value of the multi-dimensional correlation coefficient R expressed as follows :

$$R=\frac{\sum_{n=1}^{N}\left[(Y_n-\overline{Y}_r)(\hat{Y}_n-\overline{Y}_m)\right]}{\sqrt{\sum_{n=1}^{N}(Y_n-\overline{Y}_r)^2\sum_{n=1}^{N}(\hat{Y}_n-\overline{Y}_m)^2}} \tag{7}$$

b) value of the remainder variance $\hat{\sigma}^2$ expressed as follows :

$$\hat{\sigma}^2=\frac{1}{N-K-1}\sum_{n=1}^{N}(Y_n-\hat{Y}_n)^2 \tag{8}$$

Value of R should be as large as possible, and value of $\hat{\sigma}^2$ - as small as possible. It then speaks for an insignificant deviation of the model from reality.

The mathematical model of compressor can be improved by adding new terms or replacing the existing ones in the equations (3) and (4). Solution of the mathematical model of compressor consists in determining values of the coefficients in the elaborated regression equations.

The LP compressor of DR76 engine was used to confirm usefulness of the least squares method and multi-dimensional polynomial-based regression method for building the mathematical model of characteristics of the compressor with changeable flow passage geometry. In Fig.8 a part of the compressor model characteristics is presented for three angular settings of controllable inlet guide vane : α_{KW} = -10°, α_{KW} = 0°, α_{KW} = +10°. Generally, for each angular position of guide vane blades the compressor characteristics take the form described by three values : of the rotational speed of compressor rotor, n_{zr}, of the mass air flow rate $\dot{m}_{zr}$, and the compressor compression π_s. In Fig.8 can be observed changes of the compressor operation range for two values of the reduced rotational speed of compressor, n_{zr} = 8000 and 9000 rpm, in function of the angle of setting of controllable blades of initial whirl guide vane.

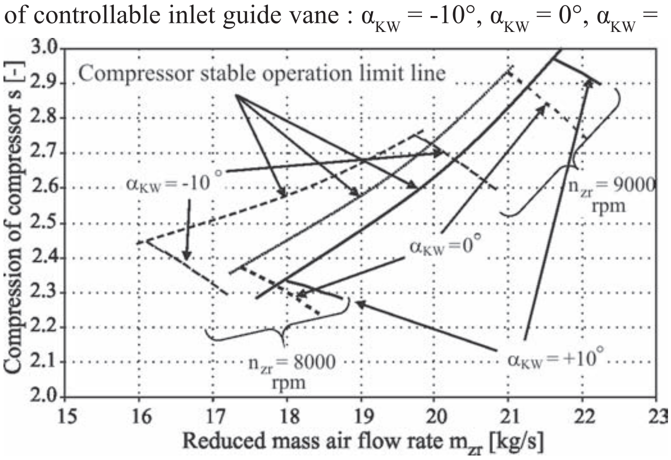


Fig. 8. Compressor model characteristics for changeable angles of setting of controllable blades of initial whirl guide vane; stable operation limit line for the angle of setting of controllable guide vane, α_{KW} = -10°, stable operation limit line for the angle of setting of controllable guide vane , α_{KW} = 0°, stable operation limit line for the angle of setting of controllable guide vane , α_{KW} = +10° .

Values of the regression coefficients of the equations which approximate characteristics of the compressor in question are given in Tab. 1, and in Tab. 2 – values of statistical parameters showing degree of adequacy of values obtained from model investigations against experimental ones.

Tab. 1. *Values of regression coefficients for particular equations.*

No.	Regression coefficient	Value	Regression coefficient	Value
1	a_0	2.099955	b_0	-0.583216
2	a_1	-0.312917	b_1	-0.272626
3	a_2	-0.022847	b_2	-0.017450
4	a_3	0.000047	b_3	0.000976
5	a_4	0.000000	b_4	0.000000
6	a_5	0.000115	b_5	0.000113
7	a_6	0.059954	b_6	0.068760
8	a_7	-0.003556	b_7	0.000929
9	a_8	0.016743	b_8	0.009397
10	a_9	-0.000038	b_9	-0.000032

Tab. 2. *Degree of adequacy between the model and real object*

No.	Characteristics	R	$\hat{\sigma}^2$
1	$\pi_s=f\left(\dot{m}_{zr},n_{zr},\alpha_{KW}\right)$	0.99939	0.00628
2	$\eta_s=f\left(\dot{m}_{zr},n_{zr},\alpha_{KW}\right)$	0.99782	0.00298

CONCLUSIONS

- The presented mathematical model of axial compressor working in the gas turbine engine system makes it possible to numerically investigate the gas-dynamic processes taking place in its flow passages.
- The changeable setting angles of blades of controllable guide vanes of particular compression stages, taken into account in the model, significantly widens possible identification of unserviceability states in the system of flow passage geometry control.
- Simulative investigations of gas-dynamic processes taking place in the compressor in the conditions of introduced changes in set-up values of the input quantities of the presented model, make it possible to determine the diagnostic relations „defect-symptom" used for assessing technical states of gas-turbine engines.

NOMENCLATURE

α_1 - angle of air flow off blades of guide vane ring
α_{KW} - setting angle of controllable blades of guide vanes
β_1, β_2 - angle of air inlet to and outlet off rotor blades, respectively
c_a - axial component of absolute flow velocity

c_{1a} - axial component of absolute flow velocity at inlet to rotor blade ring,
$c_{1a\,obl}$ - computational value of axial component of absolute flow velocity at inlet to rotor blade ring,
η_s - compressor efficiency
i - angle of air flow onto rotor blades
φ - relative air humidity
λ - angular setting of engine load control handle
$\dot{m}$ - mass air flow rate
$\dot{m}_{pal}$ - mass fuel flow rate
n - rotational speed of compressor rotor
n_{BJ} - rotational speed of rotor of compressor during engine idle running
n_1, n_2, n_i, - rotational speeds of compressor rotor at partial load states
n_{nom} , n_{max} - rotational speeds of compressor rotor at rated and maximum load of engine, respectively
n_{zr} - reduced rotational speeds of compressor rotor
p_{ot} - ambient pressure
π_s - compression of compressor
R - multi-dimensional correlation coefficient
T_{ot} - ambient temperature
$\hat{\sigma}^2$ - remainder variance
u - circumferential velocity
w_1, w_2 - relative air flow velocity at inlet to and outlet from rotor blade ring, respectively
Δw_u - flow swirl in rotor

Abbreviations and indices:

a_i, b_i - coefficients of regression equation ($i = 0…9$)
COGAG - Combination Gas-turbine and Gas-turbine propulsion system
k - number of regression function coefficient
KS - combustion chamber
MN - space between propulsion turbine and LP turbine of engine
n - number of measurement ($n = 1 … N$)
N - set of measurement points
PM - space between compressors
PR - reduction transmission gear
PSK - space between HP compressor and combustion chamber
S - compressor
SN - screw propeller
SNC - low pressure (LP) compressor
SWC - high pressure (HP) compressor
TN - propulsion turbine
TNC - low pressure (LP) turbine
TWC - high pressure (HP) turbine
Y_n - real value at the instant of n-th measurement
$\hat{Y}_n$ - value calculated from model at the instant of n-th measurement
$\bar{Y}_r$ - real arithmetic mean value
$\bar{Y}_m$ - arithmetic mean value calculated from model

BIBLIOGRAPHY

1. Balicki W., Szczeciński St.: *Diagnostics of aircraft gas turbine engines* (in Polish). Published by Library of Institute of Aeronautics, Warszawa, 1997
2. Charchalis A.: *Diagnostics of marine gas turbine engines* (in Polish). Published by Polish Naval University, Gdynia 1991
3. Ćwik B., Szczeciński St.: *Modelling the characteristics of radial compressor supercharging gas turbine engine* (in Polish). Bulletin of Military Technical Academy (WAT), no. 3-4, vol. XLIV, Warszawa, 1995
4. Dżygadło Z. *et al*.: *Rotor units of gas turbine engines* (in Polish). Transport and Telecommunication Publishing House (WKiŁ), Warszawa, 1982
5. Korczewski Z.: *A method for diagnosing flow part of marine gas turbine engine in service* (in Polish). Polish Naval University. Doctoral thesis, Gdynia 1992
6. Korczewski Z.: *Identification of gas-dynamic processes in compressor unit of marine gas turbine engine for diagnostics purposes* (in Polish). Polish Naval University. Ass.Prof. qualifying dissertation, Gdynia, 1998
7. Korczewski Z., Wirkowski P.: *Modelling gas-dynamic processes within turbine engines' compressors with variable geometry of flow duct*. 4th International Scientifical-Technical Conference "Explo-Diesel & Gas Turbine '05", Gdańsk-Międzyzdroje-Kopenhaga, Publ. by Gdańsk University of Technology, Gdańsk 2005
8. Marschal D.J., Muir D.E., Saravanamuttoo H.I.H.: *Health Monitoring of Variable Geometry Gas Turbines for the Canadian Navy*. The American Society of Mechanical Engineers 345 E.47 St., New York, N.Y.10017
9. Muszyński M., Orkisz M.: *Modelling of gas turbine jet engines* (in Polish). Published by Library of Institute of Aeronautics, Warszawa, 1997
10. Orkisz M., Stawarz S.: *Modelling of characteristics of compressors of gas turbine engines* (in Polish). Reports of Institute of Aeronautics, no.146, Warszawa, 1996
11. Pawlak W., Wiklik K., Morawski J.M.: *Synthesis and analysiss of control systems of aircraft gas turbine engines by using computer simulation methods* (in Polish). Published by Library of Institute of Aeronautics, Warszawa, 1996
12. Pieregudow W.: *Least squares method and its application* (in Polish). State Economical Publishers (PWE), Warszawa, 1967
13. Wirkowski P.: *Double-rotor marine gas-turbine engine as an object of modeling* (in Polish). 32nd National Symposium „Diagnostyka Maszyn" (Diagnostics of Machines), Węgierska Górka, 2005.

CONTACT WITH THE AUTHOR

Paweł Wirkowski, Ms.C.
Mechanic-Electric Faculty,
Polish Naval University
Śmidowicza 69
81-103 Gdynia, POLAND
e-mail : p.wirkowski@amw.gdynia.pl

Miscellanea

Days of Engineering

On 23-25 November 2006, 37th Days of Engineering was held on the occasion of 60th Anniversary of Technical University of Szczecin, the oldest technical university of West Pomerania, as well as of many scientific technical societies of the town, namely : that of electricians, geodesists, mechanical engineers and technicians, water engineers and technicians as well as the Federation of Scientific Technical Societies NOT (Naczelna Organizacja Techniczna).

The jubilee was celebrated under the slogan :

Youth and Engineering – a chance to developing the Town and Region

which has had a very distinct meaning.

It was arranged by the following organizations :

- the Federation of Scientific Technical Societies NOT, Szczecin
- Szczecin Division of the Polish Society of Electricians
- Technical University of Szczecin
- The Society of Graduates from Technical University of Szczecin.