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Prof. T. Koronowicz ; Assoc.Prof. K. Kosowski
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Editorial

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

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

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

Editorial Board

The current and future possibilities to adapt the R-JH3 method to ship power prediction practice

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ABSTRACT

In the paper the current and future possibilities of the R-JH3 method are discussed. The R--JH3 method (presented in detail in [27]) was elaborated for the ship power prediction on the basis of a new definition of the open-water propeller fully equivalent to the behind-the--hull propeller. The R-JH3 method can be used to determine the mean effective velocity and the mean effective pitch coefficient of the behind propeller. The R-JH3 method may be very useful for scale-effect investigations and scale-effect designation when results of model self-propulsion test have to be transformed to the ship scale.

Keywords : ship hydromechanics, theory of propellers, ship hull-propeller interaction

PECULIARITY OF THE ITTC-78 POWER PREDICTION METHOD AND NECESSITY OF CHANGE OF THE THIRD GROUP OF PROCEDURES

The ITTC-78 ship power prediction method can be divided into three groups of procedures :

- the 1st group of procedures dealing with the ship resistance R_{TS}
- the 2nd group of procedures concerning the propeller thrust T_{BS} necessary to ensure the ship velocity V_S
- the 3rd group of procedures for prediction of the propeller torque Q_{BS} and the propeller revolution number n_S.

Only the 3rd group of procedures is proposed to be changed.

In the 1st group of procedures the ship model resistance is determined and its experimentally obtained value is transformed to the full scale. The scale effect is the main problem of the transformation.

In the 2nd group of procedures the change of the ship model resistance due to the action of the behind propeller is determined in the form of the suction coefficient. The value of the model suction coefficient is further supposed to be the same as that in the full scale.

In the 3rd group of procedures being the subject of changes, the ITTC-78 method introduces the concept of the equivalent open screw propeller. The concept makes it possible to determine – in model scale – the effective mean velocity and the advance mean coefficient. The possibility can be realized when the model thrust coefficients of the behind propeller and the open propeller are equal to each other. Their identity gives the value of the mean advance coefficient J_{TM} and the effective mean velocity $V_{TM} = J_{TM} \cdot n_M \cdot D_M$.

The torque coefficients of the two propellers for $J = J_{TM}$ are different. The ratio of the two torque coefficients values :

$$\frac{K_{QOM}(J_{TM})}{K_{QBM}} = \eta_{RM}$$

is known as the relative rotative efficiency.

The effective mean velocity V_{TM} and the open propeller (SOM) characteristics $K_{TOM} = f(J)$ and $K_{QOM} = f(J)$ are transformed from model scale to full scale according to the ITTC--78 transformation formulae. This way the mean effective velocity V_{TS} is received. The hydrodynamic characteristics $K_{TOS} = f(J)$ and $K_{QOS} = f(J)$ are to be joined with a new open propeller in full scale. In model scale the identity of the thrust coefficients of the model behind propeller (SBM) and the equivalent model open propeller (SOM) : $K_{TBM} = K_{TOM}(J)$, is the criterion of equivalence of both model propellers. In full scale the identity of the thrust loading coefficients of the behind propeller (SBS) and the open propeller (SOS) is the equivalence criterion of these propellers :

$$\frac{T_{BS}}{\rho_S V_{TS}^2 D_S^2} = \frac{K_{TOS}(J)}{J^2}$$

where :

$\frac{T_{BS}}{\rho_S V_{TS}^2 D_S^2}$ - thrust loading coefficient of the SBS propeller

$\frac{K_{TOS}(J)}{J^2}$ - thrust loading coefficient of the SOS propeller.

From this equivalence criterion the advance coefficient $J = J_{TS}$ can be determined and then the number of revolutions of the propeller, n_S :

$$n_S = \frac{V_{TS}}{J_{TS} D_S}$$

In the ITTC-78 method it is further necessary to assume that the value of the relative rotation efficiency in full scale, η_{RS}, is equal to η_{RM}. From the relationship :

$$\eta_{RM} = \frac{K_{QOM}(J_{TM})}{K_{QBM}} = \frac{K_{QOS}(J_{TS})}{K_{QBS}}$$

the torque coefficient K_{QBS} can be determined :

$$K_{QBS} = \frac{K_{QOS}(J_{TS})}{\eta_{RM}}$$

and hence the torque Q_{BS} and the power P_{DS} :

$$Q_{BS} = K_{QBS} \cdot \rho_S \cdot n_S^2 \cdot D_S^5$$

$$P_{DS} = 2\pi \cdot Q_{BS} \cdot n_S$$

can be calculated.

One can note that the 3rd group of procedures being in use in ship power determination has been designed on the basis of the equivalent open propeller definitions different in model and full scale. The geometry of the SOM propeller and the SBM propeller are identical. The *a priori* defined geometry leads to the hydrodynamic characteristics of such open propeller, $K_{TOM} = f(J)$ and $K_{QOM} = f(J)$, and to the equivalence criterion limited to the thrust coefficients only :

$$K_{TBM} = K_{TOM}(J)$$

From the criterion the advance coefficient $J = J_{TM}$ results. Hence the effective mean velocity $V_{TM} = J_{TM} \cdot n_M \cdot D_M$ can be determined. In full scale the hydrodynamic characteristics $K_{TOS} = f(J)$ and $K_{QOS} = f(J)$ derived form the model relationships : K_{TOM} and K_{QOM} with the help of the ITTC-78 transformation formulae, define the open propeller (SOS).

The equivalence criterion of the SOS propeller and the SBS propeller is different from that in model scale and is given in the form of equality of the thrust loading coefficients of the SBS and SOS propeller.

Some controversial elements of the 3rd group of procedures of the ITTC-78 method can be indicated.

- The open screw propeller is given *a priori* being different in model and full scale :
  - **in model scale** : (SOM) ≡ (SBM)
  - **in full scale** : the SOS propeller is described in the form of the characteristics $K_{QOS}(J)$ and $K_{TOS}(J)$.
- The equivalence criterion of both propellers (the behind propeller and open one) is different in model scale (the equality of thrust coefficients) and in full scale (the equality of thrust loading coefficients).
- The requirement that only the thrust coefficients of both propellers are to be equal to each other in model scale, results in that the torque coefficients of the propellers are different. The ratio of the coefficients, i.e. the relative rotative efficiency, is the same – according to the assumption – in model and full scale. Due to such assumption only it is possible to determine the torque Q_{BS} and the power P_{DS} by means of in the ITTC-78 method.
- The condition of equality of the thrust loading coefficients of both full-scale propellers is based on the following assumptions :
  - the transformation of V_{TM} into V_{TS} is known
  - the transformations of K_{TOM} and K_{QOM} into K_{TOS} and K_{QOS} are known.

The transformations can be verified by using indirect methods only.

The controversial elements in question can be eliminated, in principle, in two ways :

- by introducing the generalization of the equivalent open propeller definition
- or by eliminating the equivalent open propeller definition. In this case the numerical determination of the ship power can be taken into account in the full scale directly.

The first way has been applied in the Jarzyna's papers [20, 21, 22, 27]. The second one is wholly connected with investigations into the computer hydrodynamics problems (numerical basin). It is a very promising direction of scientific activity though its practical application is not yet possible.

Even though this research direction will bring full success, the results of investigations into the equivalent open water propeller will remain of a great value in determining the effective mean values connected with the behind propeller problems.

As an example of such values can serve the effective mean velocity or the effective mean pitch coefficient of the behind propeller. The only way to determine the mean effective values connected with the behind propeller is to make use of the definition of the equivalent open water propeller.

So the generalization of the equivalent open propeller definition can satisfy the following different conditions :

- to make the realization of new concepts of the 3rd group of procedures of the ITTC-78 method possible
- to make it possible to form a compact criterion of determination of mean effective values which are connected with screw propeller's global characteristics
- to serve as a valuable tool in scale effect investigations especially when the propeller thrust full-scale measurements are possible in practice.

THE ESSENCE OF THE R-JH3 METHOD

Definition of the open water propeller equivalent to the behind propeller

For given values (K_{TB} , K_{QB}) of the behind propeller and for the one – parameter family of open – water screw propellers with the pitch coefficient (P/D) being the family parameter (of the same family geometry as that of the behind propeller, except of the propeller pitch) the equation set :

$$(1) \quad \Rightarrow \quad K_{TB} = K_{TOj}\left[J, \left(\frac{P}{D}\right)_j\right]$$

$$(2) \quad \Rightarrow \quad K_{QB} = K_{QOj}\left[J, \left(\frac{P}{D}\right)_j\right]$$

has one and only one solution [$J = J_{TQ}$; $P/D = (P/D)_e$]. The parameter $(P/D)_e$ identifies – out of the propeller family – one propeller SO[$(P/D)_e$] which is the wanted open propeller (SO) equivalent to the behind propeller (SB). The characteristics of the SO[$(P/D)_e$] are :

$$K_{TO} = f\left[J, \left(\frac{P}{D}\right)_e\right]$$

$$K_{QO} = f\left[J, \left(\frac{P}{D}\right)_e\right]$$

Identity of other global characteristics of both propellers

The global hydrodynamic parameters of both the propellers: SB and SO[$(P/D)_e$] satisfy the following equations obtained from the identity of thrust loading coefficients and the identity of torque loading coefficients :

$$(3) \quad \Rightarrow \quad \frac{T_B}{\rho V_{TQ}^2 D^2} = \frac{K_{TO}\left[J, \left(\frac{P}{D}\right)_e\right]}{J^2}$$

$$(4) \quad \Rightarrow \quad \frac{Q_B}{\rho V_{TQ}^2 D^3} = \frac{K_{QO}\left[J, \left(\frac{P}{D}\right)_e\right]}{J^2}$$

In order to obtain V_{TQ} from the equation (3) the value $J = J_{TQ}$ resulting from the set of equations (1) , (2) is to be used.

Thereafter the value of the number of propeller revolutions, n :

$$(5) \quad \Rightarrow \quad n = \frac{V_{TQ}}{J_{TQ} D}$$

can be received.

The wanted value of the torque Q_B results from the equation (4) for $J = J_{TQ}$ hence the delivered power P_D can be calculated as : $P_D = 2\pi \cdot Q_B \cdot n$.

Next the thrust and torque coefficients K_{TB} and K_{QB} can be verified.

THE PROBLEMS CONNECTED WITH THE USE OF THE R-JH3 METHOD IN FULL SCALE

The set of equations (1), (2), (3), (4) and (5), in presence of the feedback between the parameters K_{TBS} , K_{QBS} and the roots of the equation set, can be solved by means of the iteration method only. One can suggest two procedures to determine the first approximation values of the K_{TBS} and K_{QBS}.

✶ The first procedure is based on the preliminary design of the behind propeller. To this end the following input parameters should be given :

T_{BS}	-	the propeller thrust sufficient to maintain the velocity V_S
V_S	-	ship velocity
D_S	-	screw propeller diameter
$V_e/V_S(r/R)$	-	the design velocity field.

The thrust T_{BS} results from the first and second procedure group of the ITTC-78 method. The parameters V_S and D_S are given values. The design velocity field can be produced by using the Koronowicz's procedure (computer program).

✶ The second procedure of determination of K_{TBS} and K_{QBS} is the O-JH3 method of ship power prediction.

The O-JH3 method is below presented in detail because it can serve as an independent prognostic method parallel to its function in determining the first approximation values of K_{TBS} and K_{QBS} for the R-JH3 method. The O-JH3 method based on the new definition of the equivalent open water propeller is very strongly connection with the ITTC-78 method and it can serve as a very useful element of correlation of both power prediction methods.

It is worth mentioning that in the case when the paralell measurement results of thrust and torque, number of propeller revolutions and ship velocity are at one's disposal, the R-JH3 method will be a useful tool for verification of and for scale-effect investigations on selected parameters applied in ship hydrodynamics.

GENERAL REMARKS CONCERNING THE REALIZATION OF THE PROGNOSTIC METHODS : R-JH3 AND O-JH3

➲ The realization of the methods R-JH3 and O-JH3 does not entail any additional experimental activity in comparison with that related to the ITTC-78 method.

➲ In the 3rd group of procedures the results of experimental model tests are processed by means of computer programs quite different from those used in the ITTC-78 method. The proposed methods do not use the open-propeller hydrodynamic characteristics experimentally obtained as in the case of the ITTC-78 method. In the new method the hydrodynamic characteristics of the one – parameter open – propeller family, and thereafter of the equivalent open propeller, are determined by using a computer program. The propeller family's geometry is the same, except of the pitch coefficient equal to that of the behind propeller.

➲ The realization of the R-JH3 and O-JH3 methods, in the phase of their introduction to the prognostic practice, is carried out parallel to the calculation procedures of the ITTC-78 method.

➲ In the ITTC-78 method the value of the parameter :

$$G_{00} = \frac{\dfrac{V_{TS}}{V_S}}{\dfrac{V_{TM}}{V_M}}$$

is forejudged because the mean effective velocity V_{TS} is determined from the model value V_{TM} by using the ITTC-78 transformation.

In the O-JH3 method the variability interval of the parameter G is assumed to be $G_{min} \le G \le G_{max}$, in which the value G_{00} is contained. The results of the O-JH3 method are given in function of the parameter G ($G = G_{00}$ included).

In the R-JH3 method the effective mean velocity V_{TS} results directly from the equation (3) (due to the identity of thrust loading coefficients of the behind propeller and the equivalent open propeller) for $J = J_{TQS}$ and $P/D = (P/D)_{eS}$ being the roots of the set of equations (1) and (2).

In the R-JH3 method the mean effective velocity V_{TQS} is directly dependent upon the input values (K_{TBS},K_{QBS}) to this equation set.

THE REALIZATION OF THE R-JH3 METHOD FOR GIVEN (K_{TB} , K_{QB})VALUES OF THE BEHIND PROPELLER

One can distinguish three cases when the (K_{TB},K_{QB}) values can be determined univocally :

➲ Full-scale measuring results : T_{BS} , Q_{BS} , V_S , n_S .

In the case when the ship measuring results (T_{BS} , Q_{BS} , n_S , V_S) are available, then – for given K_{TBS}, K_{QBS} – the equivalent open propeller, SOS[$(P/D)_{es}$], can be received from the equations (1) , (2) with the roots [J_{TQS},$(P/D)_{es}$]. And, V_{TQS} results from equation (3). Then the number of revolutions

$$n_S = \frac{V_{TQS}}{J_{TQS} D_S}$$

and the torque Q_{BS} can be calculated from (4).

Both the values n_S and Q_{BS} should be comparable with those measured on ship. This way a valuable correctness criterion of the measured values was created. The iteration method can be used to bring the values to identity.

On the other side the full-scale values :

$$V_{TQS} , K_{TOS} = f ([J,(P/D)_{es}]$$
$$K_{QOS} = f [J,(P/D)_{es}] , (P/D)_{es}$$

can be compared wih those in model scale :

$$V_{TQM} , K_{TOM} = f ([J,(P/D)_{eM}]$$
$$K_{QOM} = f [J,(P/D)_{eM}] , (P/D)_{eM}$$

The method R-JH3, used in parallel for both model and full scale, can be a very efficient tool in studying scale-effect.

➲ Model-scale measuring results : T_{BM} , Q_{BM} , V_M , n_M .

The R-JH3 method can be used in model scale because the values measured in self propulsion tests make it possible to determine K_{TBM} and K_{QBM}. The set of equations (1), (2), (3), (4) and (5) gives, in model scale, the following results :

$$J_{TQM}, (P/D)_{eM} \rightarrow SOM[(P/D)_{eM}] \rightarrow \begin{cases} K_{TOM} [J,(P/D)_{eM}] \\ K_{QOM} [J,(P/D)_{eM}] \end{cases}$$
$$V_{TM} \rightarrow n_M \quad ; \quad Q_{BM}$$

Two of the values, n_M and Q_{BM} , if related to the measured values, can be a measurement correctness criterion for these values. The iteration method can be used to bring the values to identity.

➲ Calculated values : T_{BS} , Q_{BS} , V_S , n_S
for a given final propeller (SBS).

The R-JH3 method makes it possible to find – for a given behind propeller and given parameters (K_{TBS} , K_{QBS}) – the following mean effective values :

- the effective mean velocity V_{TQ}
- the effective mean pitch coefficient $(P/D)_e$.

THE R-JH3 METHOD USED TO POWER PREDICTION DIRECTLY IN FULL SCALE

The difficulties in use of the R-JH3 method directly in full scale have been already mentioned. The difficulties are due to the feedback of the behind propeller and open propeller parameters. Two possible ways of overcoming the difficulties were indicated.

1st – which concerns the preliminary behind propeller design used as a source of the first approximation values of the thrust and torque coefficients : K_{TBS} , K_{QBS}.
One can assume that the designed propeller is the optimum one from the efficiency point of view. For a given propeller diameter an optimum revolution number can be determined. The unknown design velocity field can be described by using different approximate methods, e.g. the results presented by T. Koronowicz [28]. In the referred papers the effective velocity field is numerically generated as the final result of interaction between the ship hull and the screw propeller. Although the investigations are still in progress their partial results can be used when only the first approximation values are needed to start using the R-JH3 method.

2nd – concerning the zero method O-JH3 used to produce the first approximation values of K_{TBS} , K_{QBS}.

Stage 1 : the R-JH3 method is used in model scale. The parameters J_{TQM} , $(P/D)_{eM}$ and the equivalent open propeller, SOM[$(P/D)_{eM}$] , of the hydrodynamic characteristics K_{TOM} and K_{QOM} , are received. The effective mean velocity :
$V_{TQM} = J_{TQM} \cdot n_M \cdot D_M$ can be determined.

Stage 2 : the parameters V_{TQM} , K_{TOM} and K_{QOM} are transformed to full scale by using the ITTC transformation formulae.

Stage 3 : from the equations (3) , (4) and (5) of the R-JH3 method applied to full scale the parameters J_{TQS} , n_S and Q_{BS} can be calculated.

It should be mentioned that V_{TQS} can be determined – according to Griegson and Holtrope [5 , 7] – as a variable parameter given by the relation :

$$V_{TQS} = G \cdot V_{TQM}$$

where G is defined as :

$$G = \frac{\frac{V_{TQS}}{V_S}}{\frac{V_{TQM}}{V_M}} \quad ; \quad G_{min} \le G \le G_{max}$$

In the case when the variable parameter G is introduced all results obtained in stage 3 are functions of this parameter.

Thereafter the thrust and torque coefficients K_{TBS} and K_{QBS} are calculated in order to be used in the R-JH3 method as the first approximation of the input data.
The values $Q_{BS} = f(G)$ and $n_S = f(G)$
can be used to predict the delivered power P_{DS} :

$$P_{DS} = 2\pi \cdot Q_{BS} \cdot n_S$$

NOMENCLATURE

G	-	the ratio of nondimensional effective mean velocity of ship and model
h_R	-	relative rotative "efficiency"
J	-	advance coefficient
K_Q	-	torque coefficient
K_T	-	thrust coefficient
n_M , n_S	-	number of revolutions of model and ship propeller, respectively
P	-	power (in general)
Q	-	propeller torque
SB	-	behind - the - hull (behind) screw propeller
SB_{mod}	-	modified behind screw propeller with constant effective pitch coefficient
SBM	-	behind propeller in model scale
SBS	-	behind propeller in full scale
SO	-	open-water propeller
SO[P/D)e]	-	open-water propeller with the given effectiv pitch coefficient
SOM	-	open-water propeller in model scale
SOS	-	open-water propeller in full scale
T	-	propeller thrust

V_S - ship velocity
V_{TQ} - mean effective velocity from thrust and torque identity

Indices

B - behind quantity
M - model quantity
0 - open water quantity
S - ship quantity

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Acronyms

IMP PAN - Institute of Fluid-Flow Machinery, Polish Academy of Science
ISP - International Shipbuilding Progress
ITTC - International Towing Tank Conference
JSR - Journal of Ship Research
RINA - Royal Institution of Naval Architects

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Miscellanea

TOP KORAB
awards in the academic year 2003/2004

It is already a tradition that the Society of Polish Naval Archtects and Marine Engineers KORAB has granted awards for the best M.Sc. theses from among those commended by the Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology.

In the academic year 2003/2004
the awardwinning was :

- Mr.Krzysztof Potulski, M.Sc. for the project on :

Structural analysis of a corrugated bulkhead/ bottom connection of a medium size tanker

The project was elaborated under supervision of Mr. Janusz Ziółkowski, M.Sc., the Department of Mechanics and Strength of Ship Structures.

An algorithm for preliminary estimating hull structure mass and mass centre height of inland navigation ships

Jan P. Michalski
Gdańsk University of Technology

ABSTRACT

The paper presents an algorithm for preliminary calculation of mass and mass centre height of hull structure of inland navigation ships. It was elaborated basing on requirements of the Rules for the Classification and Construction of Inland Navigation Ships of Polish Register of Shipping, with application of a simplified method of estimating mass of hull plating stiffeners. The algorithm deals with the dimensioning of scantlings of structural members of classical ships intended for shipping dry cargo, and dry and liquid bulk cargoes; however it does not cover ships of entirely different structural arrangement such as roll-on-roll-off ships fitted with heavy decks, as the dimensioning of their scantlings is based on different models.

Key words : preliminary ship design, ship structure, preliminary ship mass estimation

INTRODUCTION

In the ship design theory an important role is played by the methods intended for preliminary, approximate estimation of mass and mass centre coordinates of a designed ship.

Knowledge of the parameters is necessary already in early designing phases to balance ship floatability and stability before exact structural strength calculations are performed. In the subject-matter literature many similar methods dealing with sea-going ships are known e.g. [1, 2, 3, 4]. However only few methods concerning inland navigation ships have been published so far, e.g. [5, 6, 7].

Such methods are elaborated on the basis of statistical investigations of mass of built ships or the rules of classification institutions, which provide analytical and tabulated relationships or diagrams. Applicability range of a particular method is usually limited to ships of a given functional type and hull structural arrangement; moreover it concerns a given structural material and ship size range.

For elaboration of the algorithm in question the rules for dimensioning hull structural elements, contained in the Rules of Polish Register of Shipping [9], were applied, as well as a method of accounting for mass of hull plating stiffeners [5] was used. The simplyfing assumptions applied in the considered mathematical model have been deemed allowable for approximate estimating the total mass of ship hull structure.

The elaborated calculation algorithm has been made easy for computer programming as it is intended for preparation of a computer software to perform series of simulation calculations of ship hull structure mass to provide a basis for elaboration of a parametric method to predict mass characteristics of inland navigation ships.

An origin of the presented research are problems met during the design work on realization of the Eureka - INCOWATRANS (Inland and Costal Water Transport System) project E!3065. One of the goals of the project is to develop a modern fleet for ecological inland waterways navigation.

Formulation of the problem and its assumptions

The algorithm concerns inland navigation ships intended for shipping both solid and liquid cargoes, whose construction corresponds with the midship section shown in Fig.1.

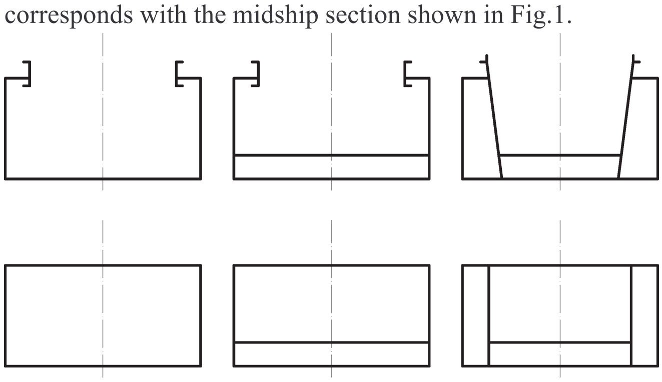

Fig. 1. *Schematic diagrams of standard midship sections of inland navigation ships*

The presented standard midship sections are special cases of the generalized midship section shown in Fig.2. By respective establishing the following dimensions : double bottom depth, double side breadth, hatch opening breadth and hatch coaming height as well as number of longitudinal bulkheads – – an appropriate standard midship section can be generated from the generalized one defined in the algorithm.

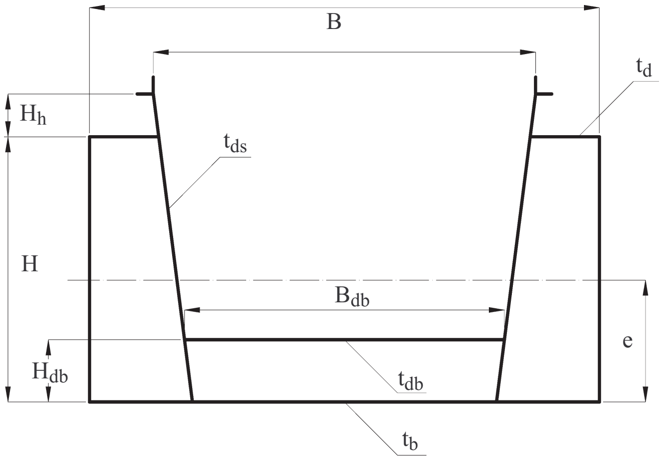

Fig. 2. *Schematic diagram of the generalized midship section of inland navigation ships*

The considered research problem consists in formulation of a mathematical model for analytical transforming the ship characteristics known at preliminary design stage, namely :

- ship numerical parameters $\overline{x} \equiv (L, B, H, T, C_B, ...)$ describing : ship main dimensions, hull form, structural material properties
- qualitative attributes $\overline{p} \equiv (K, R)$ identifying : ship functional type, its class and midship section topology
- into values of the hull structure mass *m* and mass centre height z_g , to be determined :

$$m(\overline{x}) = m(L, B, T, H, C_B, p_1, ..., p_n) \qquad (1)$$

$$z_g(\overline{x}) = z(L, B, T, H, C_B, p_1, ..., p_n) \qquad (2)$$

The hull structural elements should satisfy a number of strength requirements imposed by the classification institution rules. In the algorithm in question dimensioning the scantlings is carried out on the basis of overall and local strength criteria.

ALGORITHM FOR DETERMINING HULL STRUCTURAL SCANTLINGS

Qualitative identification attributes of hull structures

Two functional types of ships, identified by means of the symbolic index K, are considered :

- K = 1 dry cargo ships
- K = 0 liquid cargo ships.

Three classes of ships determined by restricted areas of navigation and identified by the symbolic index R [10], are distinguished :

- ★ R = 1 – navigation area on which waves of the height $h_{1/10} \leq 2.0$ [m] can be encountered (e.g. Bay of Gdańsk)
- ★ R = 2 – navigation area on which waves of the height $h_{1/10} \leq 1.2$ [m] can be encountered (e.g. Vistula Bay, Włocławek Bay)
- ★ R = 3 – navigation area on which waves of the height $h_{1/10} \leq 0.6$ [m] can be encountered - rivers, channels and lakes, which are rated, by the administration, among the inland waterways of Poland.

Numerical identifiers of hull structures

Determination of ship hull scantlings and its structural designing is associated with knowledge of ship hull form which – at early design phases – is identified by : ship dimensions, hull space subdivision, hull form coefficients etc. In the algorithm in question the following numerical parameters identifying the hull were assumed :

L - ship length [m]
B - ship breadth [m]
H - ship depth to upper deck [m]
T - ship design draught [m]
C_B - hull block coefficient [-].

Design loads

The longitudinal hull bending moment M_{FW} is the sum of the still water bending moment M_F and wave bending moment M_W. In accordance with the requirements [9] :

- ❋ the still water bending moments were determined as that not less than :

$$M_F = 0.07 B \cdot H \cdot L^2 \; [\text{kNm}] \qquad (3)$$

- ❋ for determination of the wave bending moment the following approximate formula was used :

$$M_W = k_1(R) \cdot k_2(T) \cdot A_1(R, L) \bullet$$
$$\bullet\, h_1(R) \cdot C_B \cdot B \cdot L^2 \; [\text{kNm}]$$

where : $h_1(R)$ - design wave height.

The correction coefficients $k_1(R)$, $k_2(T)$, $A_1(R,L)$ in [9] are defined by means of tabulated discrete values, rather inconvenient in coding computer algorithms. Therefore the relationships were approximated by using the following analytical functions :

$$h_1(R) \cong h(R) = 3 - 1.1R + 0.1R^2 \qquad (4)$$

$$k_1(R) \cong f_1(R) = 0.81 + 0.095R - 0.055R^2 \qquad (5)$$

$$k_2(T) \cong f_2(T) = 2.2 + 0.014T^2 - 1.85T^{0.2} \qquad (6)$$

$$A_1(R, L) \cong f_3(R, L) = 0.37 - 0.0115R + \\ - 0.0016L - 0.00063R \cdot L \qquad (7)$$

The design bending moment is the sum of the moments :

$$M_C \equiv M_{FW} = M_F + M_W \; [\text{kNm}] \qquad (8)$$

Permissible stresses

Ship hull structure exposed to external loads due to action of environment during building and operation of the ship, should provide an appropriate level of its resistance against :

- ➲ loss of integrity of structural elements
- ➲ excessive geometrical deformation of the structure
- ➲ permanent change of dimensions of the structure.

Scantlings of structural elements should be so selected as to fulfil the above mentioned criteria at given material properties, that can be realized by an appropriately low level of internal loads within the structure.

The level of permissible stresses in the structure was defined by means of the coefficient k < 1 related to the structural material yield stress σ_P [MPa] . Hence the required strength modulus of midship section can be described by the following formula :

$$W_1 = 10 \frac{M_C}{k \cdot \sigma_P} \; [\text{cm}^2\text{m}] \qquad (9)$$

As the approximating formulas usually are of a nonstructural character hence they are provided with appropriate units of measure.

Description of midship section geometry is identified by the set of parameters :

- e - distance from the neutral axis [m]
- B_h - breadth of cargo hatches [m]
- B_{db} - breadth of inner bottom [m]
- H_h - height of hatch coamings [m]
- H_{db} - depth of double bottom [m]
- t_b - thickness of outer bottom and side plating [cm]
- t_d - deck plating thickness [cm]
- t_{db} - thickness of inner bottom [cm]
- t_{ds} - thickness of inner side plating [cm]
- t_{lb} - thickness of longitudinal bulkhead plating [cm]
- S - midship cross-section area [m^2]
- W_C - minimum midship section modulus [cm^{2}m]

Minimum thickness values of the elements taken into account in the section modulus W_1 cannot be smaller than the minimum ones defined in [9], where a - frame spacing [m] :

minimum thickness of deck plating :

$$t_d = a(0.01L + 0.5) \ \text{[cm]} \tag{10}$$

minimum thickness of outer bottom and side plating :

$$t_b = a(0.01L + 0.65) \ \text{[cm]} \tag{11}$$

minimum thickness of inner bottom plating :

$$t_{db} = a(0.01L + 0.45) \ \text{[cm]} \tag{12}$$

thickness of inner side plating :

$$t_{ds} = 1.77a(0.082L + 2.5) \ \text{[cm]} \tag{13}$$

thickness of longitudinal bulkheads :

$$t_{lb} = 5a\sqrt{H} + 0.1 \ \text{[cm]} \tag{14}$$

The midship section modulus at minimum thickness of deck plating is determined as follows :

$$W_2 = t_d^{min} \frac{2.4H^2}{f_4(\cdot)\left(1 + 2.1\frac{H_h}{H}\right)} \ [\text{cm}^2\text{m}] \tag{15}$$

The midship section modulus at minimum thickness of bottom plating is determined by means of the following formula :

$$W_3 = t_d^{min} \frac{2.4H^2}{f_4(\cdot)\left(1 + 2.1\frac{H_h}{H}\right)} \cdot \frac{f_5(\cdot)}{1 - f_5(\cdot)} \cdot \frac{B}{B - B_h} \ [\text{cm}^2\text{m}] \tag{16}$$

The value of the dimensioning section modulus W_C is the largest out of the following ones :

$$\min(W_C) = \max(W_1, W_2, W_3) \tag{17}$$

Mass of hull structure stiffeners

The functions f_4 and f_5 which represent magnitude and distribution of mass of hull plating stiffeners, given in [5] by means of diagrams (rather inconvenient for coding computer programs) are described by the relationships :

$$f_4(\cdot) = f_4\left(\frac{B - B_h}{H}, \frac{B}{H}\right) \tag{18}$$

where : $\frac{B - B_h}{H}$ – independent variable

$\frac{B}{H}$ – discretely changeable parmeter.

$$f_5(\cdot) = f_5\left(\frac{B_{db}}{B - B_h} \cdot \frac{t_{db}}{t_d}, \frac{H_{db}}{H}\right) \tag{19}$$

where : $\frac{B_{db}}{B - B_h} \cdot \frac{t_{db}}{t_d}$ – independent variable

$\frac{H_{db}}{H}$ – discretely changeable parameter.

The relationships were approximated with the use of the analytical functions :

$$f_4(\cdot) = \left[ds\left(0.05 + \frac{0.95}{\frac{B - B_h}{H}} + \frac{0.82}{\frac{B}{H}}\right) + (1 - ds)\left(0.145 + \frac{1.68}{\frac{B - B_h}{H}}\right) \right] \cdot 10^{-2} \tag{20}$$

where : ds = {0,1}

If ds = 1 then inner side structures are taken into account in the hull section modulus.

$$f_5(\cdot) = db\left(0.25 + \frac{0.125}{\frac{B_{db}}{B - B_h} \cdot \frac{t_{db}}{t_d}} + 0.55\frac{H_{db}}{H}\right) + 0.5(1 - db) \tag{21}$$

where : db = {0,1}

If db = 1 then inner bottom structure is taken into account in the hull section modulus, and in the case of single bottom hull : $f_5(\cdot) = 0.5$.

Mass and mass centre height

The cross-section area corresponding with the midship section modulus W_C is determined by means of the formula consisting functions which account for stiffeners. In the case of a double bottom hull, the cross-section area of midship section is determined by means of the relationship :

$$S_{DB} = f_6\left(\frac{B - B_h}{H}, \frac{B}{H}\right)\frac{W_C}{H}\left(1 + 2.1\frac{H_h}{H}\right) + f_7\left(\frac{B}{H}, \frac{t_{ds} + t_d}{t_{db}}\right)B_{db} \cdot t_{db} \ [\text{m}^2] \tag{22}$$

In the case of a single bottom hull, the cross-section area of midship section is determined by the formula :

$$S_{NB} = f_6\left(\frac{B - B_h}{H}, \frac{B}{H}\right)\frac{W_C}{H}\left(1 + 2.1\frac{H_h}{H}\right) \cdot \left(1 + 1.34\frac{H}{B}\right) + 0.027H(t_{ds} + t_d) \ [\text{m}^2] \tag{23}$$

In [5] the functions f_6 and f_7 are defined by means of the diagrams expressing the relationship :

$$f_6(\cdot) = f_6\left(\frac{B - B_h}{H}, \frac{B}{H}\right) \quad (24)$$

where : $\frac{B - B_h}{H}$ - independent variable

$\frac{B}{H}$ - discretely changeable parameter.

$$f_7(\cdot) = f_7\left(\frac{B}{H}, \frac{t_{ds} + t_{lb}}{t_{db}}\right) \quad (25)$$

where : $\frac{B}{H}$ - independent variable

$\frac{t_{ds} + t_{lb}}{t_{db}}$ - discretely changeable parameter.

The relationships were approximated by using the following analytical functions :

$$f_6(\cdot) = \begin{bmatrix} ds\left(1.38 + 0.25\frac{B - B_h}{H} - 0.1\frac{B}{H}\right) + \\ +(1 - ds)\left(1.7 + 0.1\frac{B - B_h}{H}\right) \end{bmatrix} \cdot 10^{-4} \quad (26)$$

$$f_7(\cdot) = \begin{pmatrix} 1.22 - 0.125\frac{B}{H} + 3.14\frac{t_{ds} + t_{lb}}{t_{db}} + \\ -0.375\frac{B}{H} \cdot \frac{t_{ds} + t_{lb}}{t_{db}} \end{pmatrix} \cdot 10^{-2} \quad (27)$$

The total mass of hull structure is expressed by the formula :

$$m(\overline{x}, \overline{p}) = \rho_m \cdot S(\overline{x}, \overline{p}) L \cdot f_8(K, R) \, [t] \quad (28)$$

where :

$f_8(\cdot)$ - a function for correcting mass distribution in hull end regions

ρ_m [tm^{-3}] - structural material density.

The function $f_8(\cdot)$ defined in [5] by means of tabulated values, was approximated by using the analytical function :

$$f_8(K, R) = 1.92 + 0.167K - 0.225R \quad (29)$$

The height of hull mass centre $z_g(\cdot)$ is defined by the relationship :

$$e = z_g(\cdot) = H \cdot f_5(\cdot) = H \cdot \quad (30)$$

$$\cdot \left[db\left(0.25 + \frac{0.125}{\frac{B_{db}}{B - B_h} \cdot \frac{t_{db}}{t_d}} + 0.55\frac{H_{db}}{H}\right) + 0.5(1 - db) \right]$$

where : the function $f_5(\cdot)$ is defined by the formula (21).

PRELIMINARY VERIFICATION OF THE ALGORITHM

The preliminary verification of the algorithm was performed by using – as an example – SINE 207 ship designed in the frame of the Eureka E!3065 INCOWATRANS project. The selected ship is intended for carrying general cargo and containers : one tier in the hold and another on the deck hatch covers.

Main parameters of the SINE 207 ship

overal ship lenght	L_{OA} = 56.50 m
ship lenght between perpendiculars	L_{BP} = 55.30 m
ship breadth	B = 9.00 m
ship depth to upper deck	H = 3.00 m
design draught	T_D = 1.00 m
deadweight at T_D	P_D = 280 t
scantling draught	T_S = 1.60 m
deadweight at T_S	P_S = 580 t
container capacity	Cc = 18/36 TEU
power	N = 620 kW
speed	v = 15 km/h

In the ship's design documentation, the outfitted ship mass equal to 144.5 t was estimated on the basis of the longitudinal distribution curve of ship mass. The abscissa of mass centre was estimated equal to 27.66 m, and mass centre height equal to 1.95 m. Because of lack of information on a navigation area assumed in the design of the ship, the verifying calculations were performed for three areas of navigation : R=1, R=2 and R=3. The verified ship has been distinguished by lack of a large superstructure on the main deck. Only a small deckhouse accomodating ship's wheelhouse, has been provided on its bow. Calculation results obtained by means of the algorithm in question deals solely with hull structure mass, without outfitting, therefore the verification serves only as an estimation of prediction credibility.

Tab. 1. *Results of verifying calculations*

Verified quantities		Values estimated by using the algorithm					
Mass [t]	Mass centre (MC) height [m]	Navigation area R=1		Navigation area R=2		Navigation area R=3	
		Mass [t]	MC height [m]	Mass [t]	MC height [m]	Mass [t]	MC height [m]
144.5	**1.95**	**168.6**	**1.27**	**143.4**	**1.27**	**123.7**	**1.27**

The verifying calculations were performed by using Excel calculation sheet.

RECAPITULATION

- The presented algorithm for estimating hull structure mass of inland navigation ships can be useful in the preliminary design stage of inland navigation ships intended for carrying general cargo, dry and liquid bulk cargoes. In the case of ships of different construction, e.g. those for carrying heavy roll-on – roll-off cargo, fitted with heavy decks, results obtained by means of the algorithm may significantly differ from real masses as structural designing such ships is carried out in a different way.
- The performed verifying calculations showed convergent results of estimated masses, and, under the assumption that the SINE 207 ship construction satisfies the requirements for the navigation area R = 2 , the obtained estimation can be deemed very correct.
- The large difference of estimation of mass centre height results from not accounted for ship's outfitting.

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Miscellanea

The Academic Computer Centre in Gdańsk (CI TASK)

Founded in 1994 by the State Committee for Scientific Research (KBN), Academic Computer Centre (CI TASK) in Gdańsk is an inter-university unit that manages one of the biggest and most modern metropolitan area networks (MAN) in Poland.

The TASK network covers the territory of the whole so-called Tri-City, i.e. Gdańsk-Sopot-Gdynia. It connects 70 LAN networks of various research institutes in which over 6000 computers of PC class, workstations, and servers are installed. The network has about 16000 users (excluding students). The TASK network is connected to the national network thus enabling remote access to the Centre resources, the usage of all networks services, multimedial transmissions, interactive work, teaching and learning through the network. The Centre co-operates closely with four other supercomputer centres in Poland, and participates in creation of national metacomputing system. It also has a link to the TEN-155 panEuropean research network which provides extension of the service to the USA and other overseas regions.

A concise characteristics of the Centre main departments is as follows :

The Network Group serves as the design and supervising centre of the whole Tri-City network. Now the network operates in ATM and FDDI technologies. It consists of three FDDI rings, each with five to eight FDDI nodes connected with links operating in ATM technology. The group maintains all the network services and servers, monitors secure network operation, and gathers network statistics, both within TASK, cooperating MANs, and worldwide transmissions. Network supervising is highly automated with such tools as SUN Net Manager, Cisco Works, and Cisco Net--Flow Analyzer. Lately, the Network Group has offered creation of the Virtual LAN facility. A pilot VLAN configuration has been established for the University of Gdańsk.

The Supercomputers Group maintains and manages all the computers (IBM, SGI and SUN) installed in CI TASK equipped with a high-capacity archivisation system (ATL, HP, and EXABYTE). 34 profesional software packages make it possible to perform large scale calculations in various fields of science and technology. The group also delivers expert advice in software application and security of servers. Its latest attainment is the SGI cluster project that resulted in effective supercomputer workload.

The Inter-Disciplinary Group of Mathematical Modelling was established as a group for mathematical modelling of theoretical and applied research. Its main fields of interest include: Quantum Chemistry, Plasma Physics, Atomic Physics, Biomolecular Modelling, Mechanics, Medicine, and Electrodynamics. The Group publishes papers of its interests in the TASK Quarterly Scientific Bulletin. The journal makes it possible to present papers and exchange of views on applied numerical methods for solving a variety of problems in science and engineering by using high performance computers. Contributors are invited to send their proposals to :

quarterly@task.gda.pl

In 1999, in recognition of the Centre's achievements in metacomputing technologies based on SGI servers, a special prize was awarded to it by the KBN (The State Committee for Scientific Research).

Notation :

ATM - Asynchronous Transfer Mode
FDDI - Fiber Distributed Data Interface
LAN - Local Area Network
MAN - Metropolitan Area Network
VLAN - Virtual Local Area Network
ATL, EXABYTE, HP, IBM, SGI, SUN - firm names

ECODOCK

The Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology, having completed the Baltecologicalship project, realizes another project within the frame of EUREKA European projects. It is E!2968 project whose full name is :

Environment Friendly Floating Dock

The program's aim is to perform research and analyses to determine probable directions and intensity of future needs for environment friendly docks. The measurements and systematic investigations of pollution emitted during ship repair process will be carried out in ship repair yards. The obtained data and assessment of needs will allow to prepare technical requirements, initial design and assumptions for the construction process of environment -friendly and cost-efficient ecological floating dock of a new type.

The project will include the in-depth analytical work dedicated to the reduction of the level of hazard for the sea, coastal area, ports and adjacent estuaries of rivers, assuming application of modern technical solutions of proecological components and systems, in order to decrease the level of harmful emission and amount of waste materials. The expected result is to establish design requirements for technically-advanced, environment-friendly and cost-effective floating dock of a new type able to safely serve for all kinds of docking operations and ship repair work, particularly for the ECOLOGICAL VESSEL.

The realization of the project has been divided into several tasks to be realized by co-operating scientific centres and enterprises, namely :

Gdańsk University of Technology :

- ➲ Assessment of the present and future market demand for ship repair work in Europe versus the situation on the world market, determination of future demand for ship repair potential (overhauls and conversions) in the area of Baltic Sea.
- ➲ Analysis of quantitative state, condition and potential of ship repair industry in the Baltic Sea area.
- ➲ Design assumptions for the ecolgical floating dock and an adapted dock, and their developing study.
- ➲ Study and compendium of legal, organizational and technical regulations concerning floating docks, docking operations, as well as building the docks and their conversions with taking into account environment protection aspects.
- ➲ Development concepts of a new generation of classification rules for construction and operation of floating docks.
- ➲ Characteristics and classification of typical procedures and processes of repair , modernization and conversion of ships.
- ➲ Proecological operational concepts of service procedures for a floating dock and realized processes of ship repair and conversion.
- ➲ Analysis of technical solutions and development of a new generation of floating docks and their systems with taking into account operational reliability and safety as well as ecological cleanness (design, building, operation, overhauls).
- ➲ Analysis of ship docking-in-and-out processes with taking into account selected critical damage states of ships and ecological hazards resulting from them (hydromechanics, strength, engineering processes).
- ➲ Current and final verification of designs of a new ecological dock and a dock adapted to the CLEAN class.

Warsaw University of Technology :

Studies on and compendium of technical and physical environment protection problems associated with building and operation of floating docks :

- ⇨ ecological criteria for floating docks
- ⇨ identification and qualification of main ecological problems associated with building and operating a classical floating dock and CLEAN-class dock
- ⇨ identification of hazards to the environment from the side of ship repair processes
- ⇨ identification of qualitative and quantitative structure of production remains and waste materials from ship repair yards, and principles for their management
- ⇨ procedures for minimizing noxious emissions and waste materials in service conditions of ship repair yard
- ⇨ problems and methods of monitoring the engineering and operational processes during ship overhaul and conservation work as well as during building floating docks, in the environment protection aspect
- ⇨ proecological concepts of procedures and methods for floating dock operation, realized during ship repair and conservation work
- ⇨ recommendations concerning devices, systems and engineering processes for proecological floating docks.

SINUS Design Office Co :

- ➢ Analysis of technical and ecological shortcomings of a selected existing dock, choice of the dock as well as a scope and structure of proecological modernization work.
- ➢ Elaboration of design assumptions for proecological modernization of the selected floating dock.
- ➢ Taking part in elaborating design assumptions for a new generation of proecological floating docks of CLEAN class.
- ➢ Elaboration of preliminary design of a proecological floating dock.
- ➢ Elaboration of preliminary project of proecological modernization of the selected existing floating dock.
- ➢ Prediction of building cost at selected technological conditions.

Maritim Shipyard, Gdańsk :

- ★ Taking part in elaborating proecological design assumptions of a floating dock intended for own needs.
- ★ Taking part in monitoring the environment (water, land, air) in the area of its own hull building centre.
- ★ Taking part in monitoring the environment in the area of docking centre of NAUTA Shipyard or Gdynia Naval Shipyard.

Schiffsmakler GmbH, Rostock :

❋ Assessment of German ship repair market.

INFERT GmbH, Rostock :

❖ A study of industrial environment protection systems in the range of ship docking processes.

On application of some artificial intelligence methods in ship design

Maria Meler - Kapcia
Stefan Zieliński
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ABSTRACT

In the paper were presented examples of use of some intelligence tools such as a neural network, expert system and relational database to ship design. The neural network of back-propagation of errors was applied to select required power of ship main propulsion system on the basis of ship main parameters. Results obtained by using the network were compared with resulting values for similar ships found in Access database application. To aid design of the main propulsion system and ship power plant automation fuzzy logic was applied as an element of Case Based Reasoning (CBR) method in Exsys expert system as well as a few methods for selection of similar ships, elaborated by the authors.

Keywords : artificial intelligence, expert systems, neural networks, relational database, Case-Based Reasoning method, aided ship design

INTRODUCTION

In the subject-matter literature examples of application of artificial intelligence to ship design can be found, especially to its preliminary phase when ship's main parameters are selected on the basis of shipowner's design assumptions. In oder to ensure optimum main dimensions for a ship during its designing the approach which consists in finding ships of similar characteristics and modyfing the selected design solutions, is often applied.

To use information dealing with earlier elaborated similar designs is possible both by means of expert systems on the basis of the Case Based Reasoning method which makes designing a ship of high effectiveness faster and easier, or by means of neural networks which can be teached on the basis of representative examples and results achieved from other sources (e.g. from ship service). Thus the information processing characteristic for traditional expert systems can be deemed complementary to the dispersed parallel processing typical for neural networks.

In this work both the classical artificial intelligence tools together with a relational database were used to exemplify their application to aiding ship design in the following range :

★ Selection of power output of ship's main engine (ME) on the basis of overall ship's parameters (mainly its dimensions) by using similarity calculation methods applied in database, based on the Case Based Reasoning (CBR) approach, and then possible verification of results with the use of a neural network.
★ Aiding design of ship main propulsion system (MPS) by means of selection of similar ships with the use of fuzzy logic method on the basis of such main propulsion system parameters as ME power output and speed, as well as similarity of the remaining parameters achieved from database application.
★ Aiding selection of some parameters of ship power plant automation system on the basis of a domain model or existing designs of similar ships, with the use of fuzzy logic method embedded in an expert system, in cooperation with database application.

In order to select ME power output on the basis of general ship parameters by using similarity calculation methods in database application it was made it possible to verify obtained results by means of a neural network which uses the error back-propagation method. And, for similarity calculations and selection of similar ships within the scope of design of ship main propulsion and power plant automation systems the fuzzy logic method [9] embedded in Exsys system was used, apart from the methods provided in database application [6].

SELECTION OF MAIN ENGINE POWER OUTPUT BY USING CBR METHOD AND NEURAL NETWORK

To select main engine power output a neural network was used apart from the ship similarity calculation methods provided in database application. The below presented research results are based on the set of 222 ships built by Polish shipyards. The set is composed of very different ships. Each of them is characterized by the following parameters :

Dwt – deadweight, L – overall length, B – breadth,
D – draught, V – speed, as well as ME rated output.

In the performed investigations a relationship between ME power output and the remaining parameters was searched for. A fragment of the ship database containing values of the parameters in question is examplified in Tab.1.

The ME power output values searched out of the ship database, most similar to the power chosen according to particular similarity calculation methods, were compared with the values obtained from the neural network.

In neural network applications the one-directional networks and the teaching method with teacher's assistance are used most often. They were also applied to the investigations on aiding ship design. A classical method of teaching the multi-layer, one-directional network is the back-propagation algorithm, the

__Tab. 1.__ An example fragment of the ship database

Ship No.	Dwt [t]	L [m]	B [m]	D [m]	V [kn]	ME rated output [kW]
2.	15 300	148.9	23	8.5	14	6 800
4.	15 300	148.9	23	8.5	14	6 800
6.	7 200	169.9	28	12.3	20.5	8 600
8.	41 600	206.5	30	11.5	14.3	11 330
9.	41 450	205	30	11.48	14.6	8 338
10.	16 500	149	23	8.5	18	7 230
11.	550	60.21	10.5	3.15	11	1 200
12.	210	30.25	10.2	4.72	5	600
13.	1 480	90.63	15.02	5.4	15	3 600
15.	1 564	88.88	15.22	5.4	16	3 600
17.	18 500	141.35	22.5	9.47	13	6 650
18.	2 209	102.6	17.07	5.7	16.5	5 200

most often used in technical applications aimed at modelling unknown processes. It is the most widely known and used algorithm for the neural networks of nonlinear output function. The algorithm's essence consists in the reverse direction of correcting the weights (teaching the network) : beginning from the initial layer to the first hidden layer preceding it, and further up to the first layer. Error measure is a function of network's weights. Teaching the network consists in an adaptive correction of all weights in such a way as to obtain a minimum of the measure.

The error back-propagation method has contributed to broadening the application range of neural networks a.o. to ship design.

In the calculations in question a two-layer network of a continuous unipolar activation function and the classical algorithm of back-propagation of errors of weight changes was used. The set of ships was split in two subsets : teaching and testing. To the testing set 25% of ships were chosen in random. A fragment of the teaching set is shown in Tab.1. The ships of numbers not shown in col.1 of the table were assigned to the testing set.

All values of the ship parameters were normalized in advance to obtain the values within the interval [0,1]. In this case one calculation cycle comprised putting-in parameters of all ships of the teaching set to the network, successively. The network teaching was completed when the cycle rms error E_m achieved a value smaller than the assumed one. The error concerned the difference between the real ship ME power output and that calculated by the network for the same ship.

Convergence of the teaching process with the use of the teaching set of ships, expressed by the relationship between E_m and the number of calculation cycles L_{cc}, is presented in Fig.1. The results were obtained for the network of the following features : six inputs, 25 neurons in the hidden layer, one output neuron, the teaching factor $\eta = 0.5$, the activation function parameter $\beta = 1.5$.

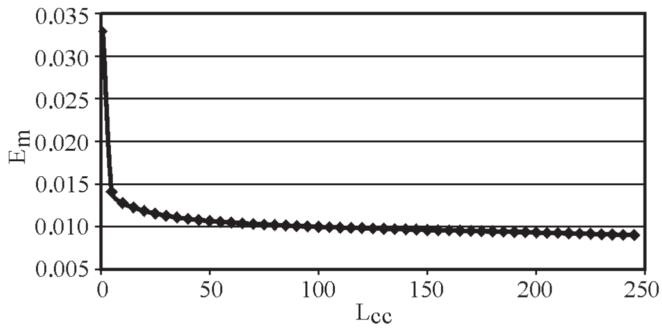

__Fig. 1.__ Teaching process convergence of the applied neural network

On completion of teaching the network by means of the teaching set of ships the calulations based on the obtained network's weights were performed with the use of parameters of ships from the testing set. The calculated ME power outputs are compared with the real ones in Fig.2. In the figure are also presented the results obtained by means of the multi-dimensional regression method with the use of 3^{rd} order polynomial model [4].

In the testing calculations the relative rms error *e* expressed by the formula (1) was also calculated :

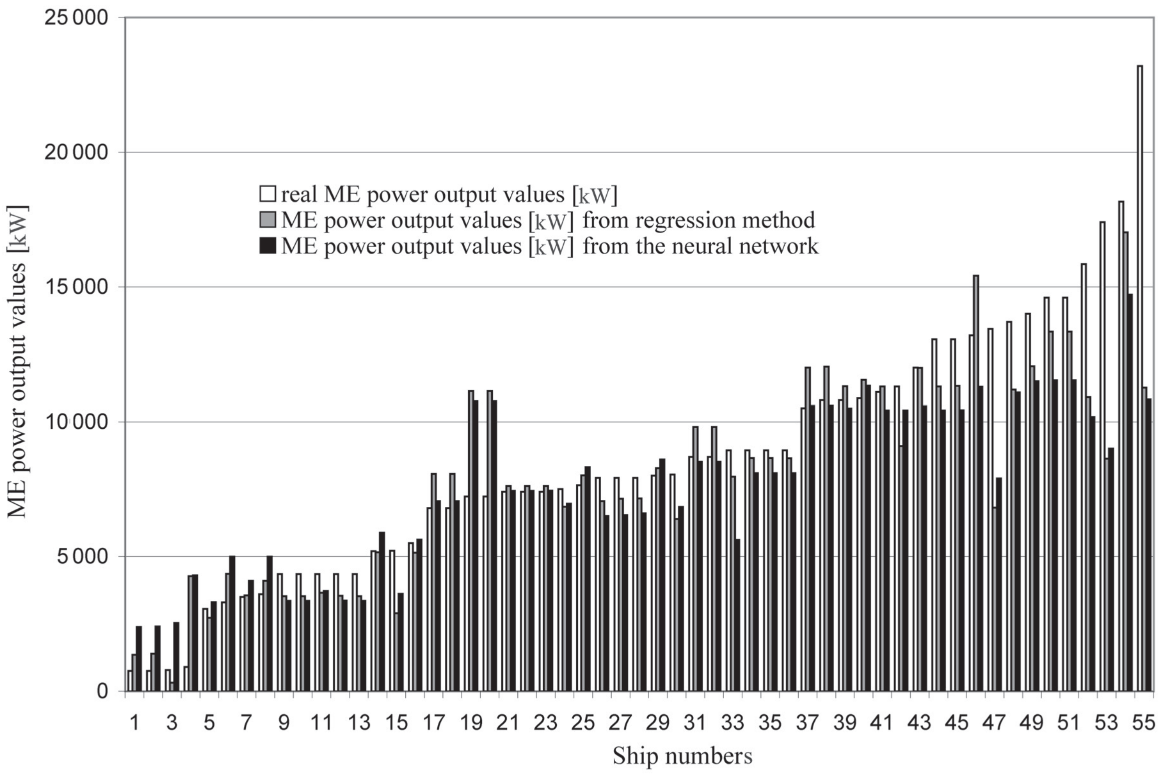

__Fig. 2.__ Comparison of the results obtained for the testing set of ships

$$e = \sum_{i=1}^{n} \frac{|M_{ri} - M_{oi}|}{M_{ri}\, n} 100 \qquad (1)$$

where :

M_r , M_o – real ME power output
and that calculated by the network, respectively.

The following values of the error were obtained :

e_1 = 25.61% for the regression method
and e_2 = 23.13% for the network.

On the basis of the results of the presented research on the neural network use the following conclusions can be offerred :

- the results obtained from the neural network and those from the regression method are similar, respectively
- the large discrepancy of the results for the testing set (large values of the error *e*) results from the large variety of the considered ships
- for the gradient teaching algorithm of the constant teaching factor η a fast convergence in the initial phase and a very slow one in the further phase of calculations, is characteristic (Fig.1).

For practical applications more effective teaching algorithms and methods of changing the factor η in the course of calculations, should be used.

In the literature sources various modifications of the error back-propagation algorithm, aimed at acceleration of the algorithm's convergence, have been proposed [10].

On the basis of the following ship design parameters : displacement, overall length, breadth, draught, and speed the similar ships were selected and their ME power output values were compared with those designed as well as with relevant values obtained from the neural network [4]. Calculation results dealing with the ships built by Polish shipyards, both obtained by applying the similarity calculation methods embedded in the database and those achieved from the neural network are presented in Tab. 2.

Tab. 2. *ME power output values of similar ships, obtained by using different similarity calculation methods, and by applying the neural network*

ME power output of designed ship [kW]	ME power output of similar ship				
	calculated by using the following similarity methods				from the neural network
	based on function with lower limit	based on Gaussian function	based on trapezoidal function	based on triangular function	
3 057	7 000	7 000	7 000	7 000	1 991
4 350	4 350	4 350	4 350	4 350	2 503
5 500	5 500	5 500	5 500	5 500	5 043
7 400	7 400	7 400	7 400	7 400	7 250
8 043	4 800	8 048	8 048	8 048	6 537
11 100	13 050	13 050	12 960	13 050	11 191
12 000	10 800	10 800	10 800	10 800	11 153
13 050	13 050	13 050	12 960	13 050	12 900
13 700	13 700	13 700	13 700	12 960	13 500

In many cases the designer can assume values of certain parameters with a lower or higher tolerance; it specially concerns the upper limit for a required parameter value as another criteria may be decisive, e.g. preference for a given supplier, or more favourable terms of delivery. For other parameters to lower a limiting value of a designed parameter may be recommended e.g. with a view of price criterion. From this point of view choice of a form of similarity function is important. In the subject-matter literature mainly symmetrical similarity functions are presented, e.g. the symmetrical similarity based on the theory of sets, or the above mentioned symmetrical similarity with a lower limit [9].

These authors have enlarged the set of similarity functions by the following ones : triangular, trapezoidal and Gaussian [6]. On the one hand it offers, for designer, a greater flexibility in determining similarity, on the other hand it forces him to select a function complying with requirements and type of an analyzed design parameter of automation system.

The similarity function with lower limit makes it possible to limit the range of calculations of similarity of an investigated parameter by establishing its lower limit. In this case the similarity value linearly increases from zero, at the lower limit, up to one.

The trapezoidal and triangular similarity functions make it possible to limit the range of similarity calculations by establishing two limits : lower and upper, and in the case of trapezoidal function – also the deviations : lower and upper, allowing to apply some tolerance margin for a given parameter.

The Gaussian similarity function treats the similar cases with the highest tolerance, not rejecting even the least similar ones.

From the presented example it results that the ME power output values for the most similar ships, obtained by using particular similarity calculation methods, are not always close to those of the designed ships. It results from that the similar ships were selected on the basis of total similarities of all input parameters. Therefore it is very important to appropriately establish values of weights of the parameters as well as limiting values for investigated ranges and their deviations.

Summing up the performed investigations one can state that the best results are obtained by using the calculation method based on the Gaussian function, and in some cases – the method based on the trapezoidal function.

The differences of the similarities obtained by means of particular methods may result from the following causes :

- a too low number of input parameters of crucial importance for choice of ME power output, has been taken for the comparisons
- a very diverse structure of the investigated set of ships in the database (different types, tasks, gabarites)
- a too low number of the analyzed ships
- a too small set of ships in the database, which is especially important for teaching the neural network.

FUZZY LOGIC AIDED DESIGN OF SHIP MAIN PROPULSION SYSTEM

The aiding of design of ship main propulsion system (MPS) is carried out by using the probability calculation methods contained in database application, as well as fuzzy logic method embedded in expert system.

Fuzzy logic can be applied simultaneously to any number of parameters. It makes it possible to pass from numerical quantities to linguistic ones by which logical reasoning can be easily performed [8]. The process this way becomes independent of a scale of numerical values of considered parameters. A general schematic diagram of the discussed method of probability determination is presented in Fig.3.

Resulting value of numerical measure of the similarity M_p depends on :

- assumed attribution functions by which input values are fuzzified

- ✦ definition of rules by which reasoning is carried out
- ✦ mode of sharpening realization.

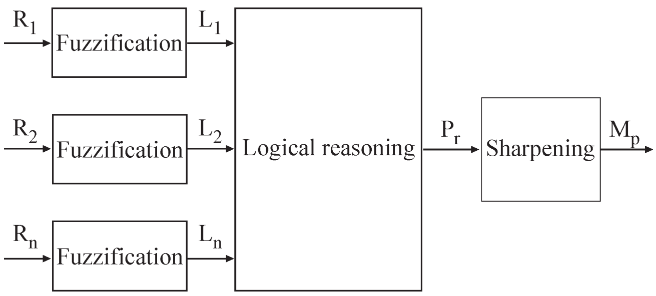

Fig. 3. Schematic diagram of the fuzzy set method

where :

$R_1, R_2, .. R_n$ – differences of numerical values of considered input parameters
$L_1, L_2, .. L_n$ – sets of linguistic definitions associated with assumed fuzzy sets defined on the basis of the values $R_1, R_2, .. R_n$, respectively
P_r – fuzzified probability
M_p – resulting numerical similarity measure.

The MPS design aiding is carried out in the following way :

- ➲ in the database application similarity of MPS is calculated on the basis of such non-numerical parameters as : type of ME and its producer, as well as the following numerical parameters : number of MEs and number of propellers.
- ➲ the weighted sum of similarities of the parameters is introduced, together with the data on ME power output and speed, into the expert system *Exsys* in which the parameters are fuzzified.

Assessment of the calculation methods of MPS similarity of ships in the expert system was carried out on the basis of the data on the selected MPS parameters given in Tab.3, as well as the values of the MPS design parameters contained in Tab.4.

In Tab.4 values of all MPS parameters of a designed ship are recorded.

As far as the MPS is concerned, determination of similarity of ships by using fuzzy logic is based on the following numerical parameters :

- ♦ the absolute value of difference of ME power output of a designed ship and built one, R_m
- ♦ the absolute value of difference of ME speed of a designed ship and built one, R_n
- ♦ the non-numerical similarity of MPS (calculated in the database) attributed to built ship, P_o , taking values from the interval [0,1].

Calculation of fuzzified similarities is realized in two phases :

- ➢ the fuzzified similarity of MPS, P_l, is calculated on the basis of the fuzzified parameters R_m and R_n
- ➢ the resulting fuzzified similarity of MPS, P_w , is calculated on the basis of the numerical similarity of MPS and fuzzified similarity P_o .

Ships similar to a designed ship are selected on the basis of the maximum resulting fuzzy similarity. To realize the fuzzification process of values of the parameters the attribution functions shown in Fig.4, were applied.

The example logical relationships assumed in rules of the expert system, dealing with the numerical fuzzified similarity of MPS are given in Tab.5, and the resulting fuzzified similarity of MPS – in Tab.6.

Tab. 3. *Data of MPS parameters selected out of the database*

Ship's No.	Main engine (ME)					Number of propellers
	Type	Producer	Output	Speed	Number of units	
1.	6ZB40/48	SULZER	3 600	500	1	1
2.	12V40/54A	M.A.N.	7 500	450	1	1
3.	16V32D	WÄRTSILÄ	8 043	800	1	1
4.	7S35MC	B & W	6 650	154	1	1
5.	5S60MC	B & W	10 869	102	1	1
6.	6RTA58	SULZER	12 960	127	1	1
7.	5L35MC	B & W	3 800	200	1	1
8.	6RLB66	SULZER	11 100	124	1	1
9.	6RTA58	SULZER	12 960	127	1	1
10.	7RND76	SULZER	14 000	122	1	1
11.	5RD68	SULZER	5 500	140	1	1
12.	6ZL40/48	SULZER	4 350	480	1	1
13.	5K62EF	B & W	7 400	155	1	1
14.	6ZL40/48	SULZER	4 350	480	1	1
15.	5RD68	SULZER	5 500	140	1	1
16.	6ZL40/48	SULZER	4 350	480	1	1
17.	K6Z70/120E	M.A.N.	8 400	140	1	1
18.	6RND90	SULZER	17 400	122	1	1
19.	SBA8M528	DEUTZ	600	620	1	1
20.	SBA8M528	DEUTZ	600	620	1	1
21.	SBA8M528	DEUTZ	600	620	1	1
22.	K9Z60/105E	M.A.N.	9 000	165	1	1
23.	6K62EF	B & W	8 940	155	1	1

Tab. 4. *Values of MPS design parameters*

Ship's symbol	Main engine (ME)					Number of propellers
	Number of units	Type	Producer	Output [kW]	Speed [rpm]	
Designed ship	1	6RTA 76	HCP SULZER	17 220	104	1

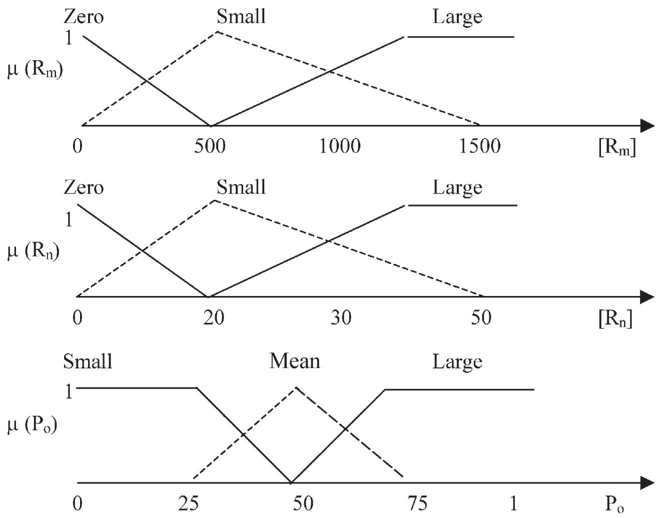

Fig. 4. *Attribution functions for MPS parameters*
μ – value of membership function determining fuzzy set

The example rules resulting from Tab.5 and 6 can be written as follows :

If (R_m = zero) and (R_n = zero) then (P_1 = large)
If (P_1 = large) and (P_o = small) then (P_w = mean).

In the Exsys system the degrees of attribution to fuzzy sets are taken as the so called confidence values. Reasoning by using

Tab. 5. Logical relationships in 1st phase of reasoning

R_m	Zero			Small			Large		
R_n	zero	small	large	zero	small	large	zero	small	large
P_1	large	large	mean	large	mean	small	mean	small	small

Tab. 6. Logical relationships in 2nd phase of reasoning

P_1	Large			Mean			Small		
P_0	small	mean	large	small	mean	large	small	mean	large
P_w	mean	large	large	small	mean	large	small	small	mean

appropriate rules triggers calculating confidence values concerning conclusions from the rules, on the basis of confidence values for premises. On their basis the sharpening is realized in result of which a single numerical value is achieved :

$$Z = W_1C_1 + W_2C_2 + W_3C_3$$

where :

Z – numerical value of MPS similarity (contained within the interval [0, 1]

C_1, C_2, C_3 – confidence values for the values : „large", „mean", „small", respectively, achieved from the reasoning process

W_1, W_2, W_3 – weights having non-negative values, $W_1 + W_2 + W_3 = 1$.

The application of fuzzy logic was tested on several examples (the design data P1÷ P5), and their results dealing with similarity calculations and selection of similar ships are shown in Tab.7.

Tab. 7. Calulation results of MPS similarity obtained by using Exsys system

Example symbol	MPS power as designed	MPS speed as designed	Number of similar ships	Similarity value	MPS power of similar ship	MPS speed of similar ship	Nos. of similar ships
	[kW]	[rpm]	-	-	[kW]	[rpm]	
P1	16 200	107	3	0.6286	18 160	110	41,60, 124
P2	11 400	110	20	0.6286	10 800	118	71
P3	6 600	150	1	0.8	6 650	154	4
P4	11 000	120	38	0.6286	13 050	124	63
P5	17 000	500	3	0.45	17 400	530	84

In Tab.8 the calculated similarities and selected similar ships (examples P1÷ P3) selected with the use of fuzzy logic are compared with those obtained by using the database application.

Tab. 8. Results of selection of similar ships from database and by using Exsys software

Example symbol	MPS power as designed	MPS speed as designed	Similar ship number	MPS power of similar ship	MPS speed of similar ship	Applied method
	[kW]	[rpm]	-	[kW]	[rpm]	-
P1	6 600	150	4	6 650	154	All applied methods
P2	11 000	120	93,117, 114	10 800	119	Database methods
			63	13 050	124	Exsys
P3	17 000	500	38,104, 84	17 200	530	Database methods
			84	17 400	530	Exsys

AIDING SELECTION PROCESS OF SOME PARAMETERS OF SHIP POWER PLANT AUTOMATION SYSTEM

Apart from aiding MPS design, fuzzy logic was also applied to aiding selection process of some parameters of ship power plant automation system.In the case in question the similarity was related to the features characterizing power plants of built ships as it was assumed that solutions dealing with automation system depended on some features of ship power plant. Because of the large number of accounted-for features, similarity of ships was determined by using some groups of the features. Full set of the considered features was split into subsets related to :

- the whole ship (general similarity determined by a type and size of ship) - Tab.9
- the main propulsion system (MPS) (similarity determined by a type of propulsion system and its main parameters) Tab.10
- selected ship systems (similarity determined by system's function and its design features) - Tab.10
- the electric power plant (similarity determined by a type of generating sets and their main parameters) - Tab.11

Tab. 9. Values of general ship parameters

Ship symbol	Ship type	Ship deadweight	Number of refrigerated containers	Number of cars	Classification society	Automation class
		[t]	[pieces]	[pieces]		
B191	container ship	1 504	200	100	DNV	AUT
B222	bulk carrier	14 800	0	0	LRS	UMS
B369	refrigerated ship	9 860	60	0	DNV	UMS
B500	container ship	29 600	150	120	BV	AUT
B501	roll on–roll off ship	9 760	0	80	DNV	E0
B683	bulk carrier	49 000	0	0	DNV	E0
B684	bulk carrier	48 000	0	0	DNV	E0
Designed ship	container ship	25 000	200	100	BV	AUT

Results of similarity calculations within the range of the subsets are considered as partial similarities whose sum represents the total weighted similarity.

The similarities calculated in the database application were transferred to the Exsys expert system to be fuzzified together with the parameters whose similarities were determined by using fuzzy logic directly.

In Exsys system the B191 ship was selected as the most similar. The partial similarities of this ship calculated with taking into account the weights, are given in Tab.12.

The maximum partial similarities together with symbols of relevant ships as well as maximum total similarity of ship, i.e. sum of partial similarities were transferred from Exsys system to the database.

Partial similarities of the similar ship (B 191), calculated with accounting for appropriate weights, are presented in Tab.13. The maximum partial similarities dealing with different ships are presented in Tab.14.

It should be added that when the entire automation design project of the selected ship does not satisfy assumed requirements its particular elements may be taken from other built ships, selected on the basis of maximum similarities of the systems.

The results concerning investigation of similar ships, carried out by using Exsys system, were obtained on the basis of Tab.14 which contains weighted partial similarities of all ships stored in the database.

Aiding the design of ship main propulsion and automation systems, based on CBR methods consists in automatic searching out of ship database the ships most similar to a designed one [5]. In this scope, the designer selects, out of database fields, the parameters on the basis of which at first the partial similarity of MPS, electric power plant and particular ship's systems, then the similarity of the whole ship, together with their relevant weights, will be calculated. The designer may pass over

Tab. 10. *Values of parameters of MPS and selected ship's systems*

Ship symbol	Number of MEs	ME type	ME output	ME speed	Number of transmission gears	Number of propellers	Type of propeller(s)	Number of valves in fuel system	Number of valves in bilge system
	[pieces]	-	[kW]	[rpm]	[pieces]	[pieces]	-	[pieces]	[pieces]
B191	1	6L70 MC	16 200	107	1	1	of fixed pitch	22	25
B222	2	6L46	6 300	500		1	of controllable pitch	24	30
B369	1	6RTA 62-R1	11 400	102	2	1	of fixed pitch	35	27
B500	1	6RTA 76	1 7220	104	0	1	of fixed pitch	40	23
B501	4	8ZAL 40 S	23 040	510	2	2	of controllable pitch	28	33
B683	1	5RTA 62 U	8 670	102	0	1	of fixed pitch	25	28
B684	1	5S 60 MC	10 200	105	2	1	of fixed pitch	25	30
Designed ship	1	6RTA 76	16 500	110	1	1	of fixed pitch	28	30

Tab. 11. *Values of electric power plant parameters*

Ship symbol	GS1 generating set				GS2 generating set				shaft generator
	number	type	output	speed	number	type	output	speed	type
	[pieces]	-	[kW]	[rpm]	[pieces]	-	[kW]	[rpm]	-
B191	3	8S20 H	1 160	1 440	1	6S20 H	950	1 000	none
B222	3	6L20C	1 080	1 000	2		1200	920	
B369	3	6ATL 25H	1 000	920	1		850	1 440	
B500	4	HCP	1 200	920	1	6S20 H	1700	800	
B501	2	GR 22 HF	1 170	1 000	2	8R 22 HF	1750	920	
B683	3	GR 20	920	920	2		1300	1 000	
B684	3	KRG-6	1 010	2 880	1		1260	800	
Designed ship	3	6ATL25H	1 500	1 000	2		1850	800	

Tab. 12. *Results of investigation of similarities of ships*

Ship symbol	Similarity				
	Total	General	of MPS system	of electric power plant	of ship's systems
B191	0.3051	0.3432	0.4208	0.1680	0.2601
B369	0.2713	0.3164	0.3338	0.1512	0.3040
B501	0.1790	0.2634	0.1479	0.1414	0.2552
B683	0.2047	0.2424	0.2044	0.1550	0.2609
B684	0.2644	0.2448	0.2971	0.1680	0.3534
B500	0.2783	0.4124	0.3165	0.1680	0.3002
B222	0.1992	0.2755	0.910	0.2400	0.3160

Tab. 13. *Partial similarities of similar ship (B 191)*

Kind of similarity	Weight of parameter	Value of similarity
General similarity	0,1	0.343
Similarity of MPS system	0,4	0.420
Similarity of electric power plant	0,3	0.168
Similarity of ship's systems	0,2	0.260

Tab. 14. *Maximum partial similarities dealing with different ships*

Kind of similarity	Ship	Value of similarity
General similarity	B500	0.41
Similarity of MPS system	B191	0.42
Similarity of electric power plant	B222	0.24
Similarity of ship systems	B684	0.35
Total similarity	B191	0.30

the phase of calculation of weights, in this case they will be assumed equal.

The weighted similarity both partial and total one of the whole ship can be calculated by means of a database application program with the use of any of the proposed method basing on one of the applied similarity functions (that with lower limit, trapezoidal, triangular or Gaussian one). In the expert system using fuzzy logic, for the searched-out ships most similar to the designed one, are specified values of its partial similarities and of the maximum partial similarities found for a given parameter.

If ambiguous results are obtained from the above mentioned methods the designer is able to verify them by using an error back-propagating neural network, assuming selected database fields as input and output parameters.

RECAPITULATION

- In recent years intensive research aimed at making use of developments of artificial intelligence, a.o. expert systems and neural networks, in solving different tasks of ship design, have been carried out [2], [3], [4].
- These authors attempted to apply an error back-propagating neural network for selection of power of ship main propulsion system (MPS) on the basis of general ship parameters, a.o. ship main dimensions, in order to find a new solution in the case when already found one is not satisfactory for ship designer.
- The results obtained from the neural network were compared with the values resulting for the similar ships found in the database. It may be concluded that the neural network teached in advance on the basis of solutions of existing ships, with the use of general ship design data, can be applied to verify results achieved by means of the similarity determination methods embedded in the database or expert system.
- Moreover, a fuzzy logic method was used in Exsys expert system to aid design of MPS and ship power plant automation, as a tool supplementary for CBR method ; this approach is an original and novel solution in the area of ship designing.
- The elaborated system together with the database application can serve as an intelligent designer-friendly tool to aid design process of ship automation systems.

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FOREIGN *conference*

PRADS 2004

The Technical University of Hamburg - Harburg and Schiffbautechnische Gesellschaft organized the successive, 9th International Symposium on:

Practical Design of Ships and other Floating Structures

This widely recognized scientific conference was held on 12÷17 September 2004 in Lűbeck - Traveműnde (Germany). It gave an opportunity for international contacts and cooperation of experts to stimulate development of design and production technology with a view of effectiveness and economy as well as safety improvment of ships and other floating objects.

To realize the idea 142 qualified papers were presented during 36 topical sessions on :

- Design methods
- Resistance
- Design loads
- Ultimate load
- Manoeuvring
- Methodology
- Trimarans
- Operation
- Dynamic response
- Design for safety
- Fatigue
- Seakeeping
- Cavitation
- Stability
- Catamarans and pentamarans
- Floating production systems
- Experimental techniques
- Production management
- Slamming and sloshing
- Extreme wave loads
- Hull girder strength
- Springing and torsion
- Novel ship concepts
- Lightweight structures
- Grounding and collision
- Production technology
- High speed monohulls
- Reliability analysis
- Padded drives
- Propellers
- Steel sandwich
- Propulsion
- Design for operation
- Vibration and noise
- Marine engineering
- Computer Integrated Design and manufacturing

In the broad spectrum of topics the greatest number of papers dealt with the following : Fatigue (9 papers), Ultimate load, and Methodology (7 papers each), Design methods, Resistance, Operation, Design for safety, Stability, and Padded drives (6 papers each), Propellers, and Novel ship concepts (5 and 4 papers, respectively).

The presented papers were prepared by experts from 16 European countries as well as Australia, Brazil, People Republic of China, Egypt, India, Japan, Korea, Taiwan and USA.

Among them were also four Polish authors who presented the following problems :

★ *Strength test of steel sandwich panel* – by J. Kozak (Gdańsk University of Technology)

★ *Numerical simulation of crash and grounding of inland waterway transportation barges* – by T. Jastrzębski, M. Taczała (Technical University of Szczecin), and K. Grabowiecki (CIM-MES Project Co., Warsaw)

★ *Efficient freight transport on shallow inland waterways* – – Results of the INBAT R&D project – by T. Jastrzębski (Technical University of Szczecin), as a co-author.

T. Borzęcki from Gdańsk University of Technology, representing Polish scientific circles, took part in the work of the International Standing Committee of the conference.

A valuable monography on electric power quality in isolated systems

The book recently published reflects technical development in the area of application of semiconductor power elements and computer techniques to electric power generation, transmission and usage onboard sea-going ships.

The book titled :

Assessment of electric power quality in ship systems fitted with converter subsystems

was elaborated by Prof. Janusz Mindykowski of Gdynia Maritime University, who adressed it to ship designers, builders, shipyard technical supervision service and ship repair teams, as well as to ship crews. Obviously it may serve as an important aid for scientific and teaching personnel and students of technical university faculties dealing with electric engineering.

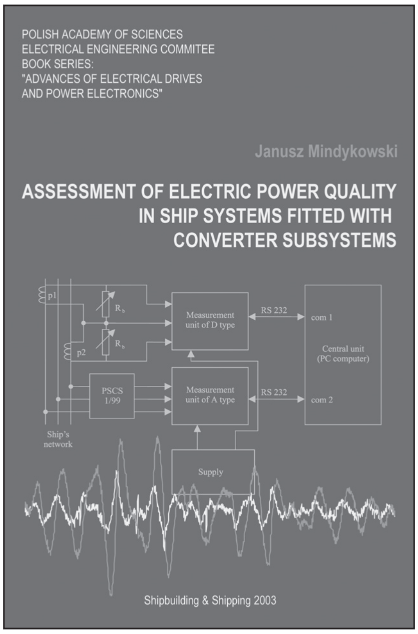
Ample content of the book based on the author's knowledge and experience gained from his multi-year research and development acitvity, is divided into 10 chapters containing the following topics :

I. Electric power quality in ship electric power systems. Character of disturbances occurring in them.
II. Specific character of their exploitation. Electric power sources, switchboards and consumers onboard ships. Requirements dealing with ship electric networks and their service conditions. Role of measurements for diagnostics and control of electric energy processes.
III. Characteristics of converter systems with accounting for electronic power converters, and their influence on electric power quality on ships. Disturbance sources and mechanisms within such systems.
IV. Measurement and control instruments applicable to ships, requirements for them and assessment of their accuracy.
V. Ship electric power plant as an object of measurements. Main relationships describing ship three-phase electric power network. Analysis of measurement instruments applied to assessment of electric power quality on ships.
VI. An attempt to defining : what is quality of electric power in ship electric networks, by which factors is it determined, and on the basis of which premises should it be assessed ?
VII. Analysis of influence of electric power quality on operation safety and economy of ship systems at ship service conditions.
VIII. Electric power quality indices with the view of ship electric engineering features. A synthetic index proposed as a simple multi-criterial assessment method of electric power quality.
IX. Methods of determination of electric power quality indices by means of measuring instruments, both analogous and microprocessor ones. Basic mathematical tools applied to such instruments. Multi-functional microprocessor instruments and analyzers of electric power quality, elaborated and manufactured by a team of Department of Ship Electric Engineering, Gdynia Maritime University, under supervision of the author. Description of a special analyzer based on own original measurement algorithms and software, as well as on positive results of own experimental investigations.
X. Possible ways to improving quality of electric power in the systems in question by means of usage of passive, active and hybrid harmonic filters, or damping transformers and selected solutions of the systems. Considerations on possible control of electric power quality in accordance with work safety and exploitation economy criteria for ship engineering systems.

In that broad presentation of the matters the author took into account the up-to-date state of knowledge in the area of measurement technique, informatics, electronics and electric engineering. The applied mathematical tools seem to be appropriate and the way of presentation of many complex problems, accessible. The author also indicated directions of further research in the area of „intelligent" analyzers and computerized control systems.

The book of Prof. Mindykowski is translated from the revised Polish edition: „Ocena jakości energii elektrycznej w systemach okrętowych z układami przekształtnikowymi", published in 2001 by the same publisher, i.e. Shipbuilding & Shipping Ltd, and shortly presented in Polish Maritime Research No.1/2002.

In comparison with the original, Polish version, to the English edition were introduced only small changes consisting in removal of editorial errors found in the Polish edition as well as some subject-matter changes to Ch. 3, 4 and 8, associated with amendments (of 2002) of the rules of Polish Register of Shipping and other international classification institutions. Some cuts resulting from a specifity of translation were also made, however the scope of the book remained unchanged.



A simulation model of energy distribution in ship combustion engine

Nguyen Hoang
Gdynia Maritime University

ABSTRACT

In the paper a model of energy flow and distribution in ship diesel engine cylinder was presented. This is a model of discrete parameters, being a continuation of the author's research on simulation of energy processing within ship main propulsion engine [2,3]. The model in question makes it possible to calculate energy flow values delivered and transferred during every working cycle of the engine cylinder. Results of application of the model for 6ZA40S Sulzer engine installed on B672 ship were also attached. The results were compared with results of test-stand measurements of the engine, performed at different load levels.

Keywords : computer simulation, ship diesel engines, dynamic processes, energy balance and distribution

INTRODUCTORY NOTE

Knowledge of energy flow and distribution within combustion engine is important for many reasons. It makes it possible to estimate and optimize engine's efficiency as well as to diagnostically model its technical state. In the case of analyses concerning ship propulsion system it enables to determine values of input energy flows to auxiliary systems and waste heat utilization devices.

PHYSICAL MODEL

Cylinder's space together with a working medium contained in it forms a thermodynamical system which can be described by means of the equation of the 1st principle of thermodynamics. The energy delivered to the cylinder in the form of fuel flow is processed, due to combustion process, into flows of internal work, thermal energy transferred by combustion chamber walls, as well as of engine exhaust gas energy.

The energy transferred by combustion chamber walls is absorbed by media contained in the systems of fresh water cooling cylinders, heads and injectors, as well as of lubricating and piston cooling oil. Whereas the exhaust gas energy is transferred to drive turbine of supercharging system, to produce steam in waste heat boiler, as well as to the atmosphere and the water cooling head and turbine. Internal work of cylinder's working cycle is transferred through the crankshaft-piston system to the propeller, to drive auxiliary devices of the engine, as well as consumed during overcoming friction drag within the engine. A small part of the combustion energy is transferred through the engine's elements to the environment.

The following assumptions have been made :

- Instantaneous temperature, pressure and gas constant of a medium take the same values in every point of the space over piston.
- Heat exchange surface area in the cylinder contains only three crucial ones : that of lower plate of cylinder head together with valves, of piston head, as well as changeable surface area of cylinder liner which is in contact working medium. Average thickness of walls of cylinder liner, lower plate of cylinder head, piston head and cylinder block, as well as material parameters of the walls, are assumed constant.
- Heat flows only in perpendicular direction to particular surfaces.

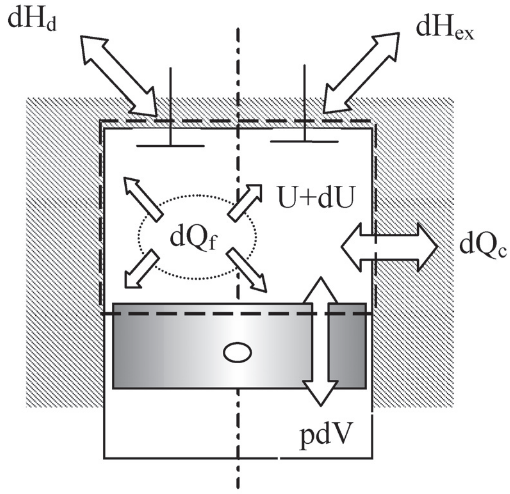


Fig. 1. *Schematic diagram of energy balance for the thermodynamical system "cylinder".* ***Notation*** *: dH_d - input enthalpy increment, dH_{ex} - output enhtalpy increment, dQ_f - heat emitted during fuel combustion, dQ_c - heat exchanged with walls, pdV - work done by working medium, U - working medium internal energy*

MATHEMATICAL MODEL

Energy balance in the cylinder can be written in the following form :

$$\frac{dU}{d\tau} = \frac{dQ_f}{d\tau} - \frac{dQ_c}{d\tau} - p\frac{dV}{d\tau} + \frac{dH_d}{d\tau} - \frac{dH_{ex}}{d\tau} \quad (1)$$

The derivatives appearing in (1) express rate of change of the following flows :

$\frac{dU}{d\tau} = \dot{U}(\tau)$ - of internal energy in the cylinder

$\frac{dQ_f}{d\tau} = \dot{Q}_f(\tau)$ - of heat of combustion of fuel injected to the cylinder

$\frac{dQ_c}{d\tau} = \dot{Q}_c(\tau)$ - of heat exchanged with the walls

$p(\tau)\frac{dV}{d\tau} = N_i(\tau)$ - of internal work performed by working medium

$\frac{dH_d}{d\tau} = \dot{H}_d(\tau)$ - of input enthalpy

$\frac{dH_{ex}}{d\tau} = \dot{H}_{ex}(\tau)$ - of output enthalpy.

By taking into account that the internal energy of working medium is a function of the temperature T and its chemical content F

$$\frac{du}{d\tau} = \frac{\partial u}{\partial T}\frac{dT}{d\tau} + \frac{\partial u}{\partial F}\frac{dF}{d\tau} \tag{2}$$

where :

F - ratio of the combusted fuel mass m_{fb} and the mass, m_a, of the air contained in the mass of working medium m :

$$F = \frac{m_{fb}}{m_a} \;,\; m = m_{fb} + m_a \tag{3}$$

and

$$c_v = \frac{\partial u}{\partial T}$$

the equation (1) can be transformed to the form :

$$\frac{dT}{d\tau} = \frac{1}{mc_v}\left(\begin{aligned} &\frac{dQ_f}{d\tau} - \frac{dQ_c}{d\tau} - p\frac{dV}{d\tau} + h_d\frac{dm_d}{d\tau} + \\ &- h_{ex}\frac{dm_{ex}}{d\tau} - u\frac{dm}{d\tau} - m\frac{\partial u}{dF}\frac{dF}{d\tau} \end{aligned} \right) \tag{4}$$

On its basis the instantaneous temperature of working medium in the cylinder after the calculation step $\Delta\tau$, can be determined :

$$T(\tau + \Delta\tau) = T(\tau) + \frac{dT}{d\tau}\Delta\tau \tag{5}$$

The equation (5) together with the state equation of working medium make it possible to determine course of temperature and pressure in the cylinder in function of time or crank angle [3]. To this end is also necessary to determine rates of the above mentioned energy flows as well as energy parameters of working medium in every calculation step of cycle. The working medium flows through inlet and outlet valves in normal and reverse direction depending on pressure values in the cylinder and manifolds as well as instantaneous state of opening of the valves, were determined. Different cases of the equation (1) regarding cycle phases realized in the cylinder, were taken into account.

For instance, in the charging phase only inlet valves through which fresh air flows in the cylinder, are opened. Depending on difference of the pressure p in the cylinder and p_d in inlet manifold, the following flows through inlet valves take place :

- at $p_d > p$ - **normal flow** - in which the mass flow $\left(\frac{dm_d}{d\tau}\right)$ enters the cylinder - of the specific enthalpy of air, h_d, in the manifold, resulting in changing the chemical content of working medium F and its internal energy u in the cylinder :

$$\frac{dQ_f}{d\tau} = 0 \;;\; \frac{dm_{ex}}{d\tau} = 0 \;;\; \frac{dF}{d\tau} = -\frac{(F+1)F}{m}\frac{dm}{d\tau}$$

The equation (4) is of the following form :

$$\frac{dT}{d\tau} = \frac{1}{mc_v}\cdot$$

$$\cdot\left(-\frac{dQ_c}{d\tau} - p\frac{dV}{d\tau} + h_d\frac{dm_d}{d\tau} - u\frac{dm}{d\tau} - m\frac{\partial u}{dF}\frac{dF}{d\tau}\right)$$

- at $p_d < p$ - **reverse flow** - during which the mass flow $\left(-\frac{dm_d}{d\tau}\right)$ flows out - of the specific enthalpy of working medium, h, in the cylinder. The chemical content of working medium, F, and the internal energy u of the medium in the cylinder remain unchanged :

$$\frac{dQ_f}{d\tau} = 0 \;;\; \frac{dm_{ex}}{d\tau} = 0 \;;\; \frac{dF}{d\tau} = 0$$

The equation (4) takes the following form :

$$\frac{dT}{d\tau} = \frac{1}{mc_v}\left(-\frac{dQ_c}{d\tau} - p\frac{dV}{d\tau} - h\frac{dm_d}{d\tau} - u\frac{dm}{d\tau}\right)$$

When an engine is in a steady working state, it is possible to determine an approximate state of working medium after each of the working cycle in a given cycle point. Energy balance in the engine can be expressed as follows :

$$\dot{Q}_d = N_i + \dot{Q}_c + \dot{Q}_{ex} + \dot{U} \tag{6}$$

where :

$\dot{Q}_d$ - rate of the energy delivered to cylinder

N_i - internal work power (indicated power) of cylinder

$\dot{Q}_c$ - flow rate of heat exchanged with walls

$\dot{Q}_{ex}$ - energy rate of gas exhausted from cylinder

$\dot{U}$ - internal energy rate of working medium in cylinder.

In the steady state the internal energy rate $\dot{U}$ should be equal to zero in an ideal model which is practically unavailable. However a model in which the value is relatively small, can be deemed correct.

The instantaneous flow rate of heat transferred through walls, $\dot{Q}_c$, as well as temperature distribution in walls can be determined by means of a model of heat exchange process in the cylinder – environment system [3]. Heat exchange – through walls – between working medium and cooling medium is an unsteady process. All the parameters : heat exchange surface area, temperature of walls, as well as the convective heat-transfer coefficient α are changeable in function of crank angle. The heat exchange with cylinder walls was modelled by means of the finite element method. The cylinder liner was divided into n finite elements in which concentrated temperatures of cylinder liner wall and of cooling water, are determined.

Changeable temperature in the direction of wall thickness *x* are derived from the Fourier equation :

$$\frac{\partial T_s}{\partial \tau} = a_s \frac{\partial^2 T_s}{\partial x^2} \tag{7}$$

with the 3rd kind boundary conditions on the side of working medium and cooling water :

$$\alpha(T - T_{s1}) = -\lambda_s \left(\frac{\partial T_s}{\partial x} \right)_{x=0}$$
$$\alpha_w (T_{s2} - T_w) = -\lambda_s \left(\frac{\partial T_s}{\partial x} \right)_{x=\delta_s} \tag{8}$$

where :

- α , α_w – convective heat-transfer coefficient on the side of working medium and on the side of cooling water, respectively
- a_s , λ_s – temperature equalization coefficient and thermal conductance of wall, respectively
- T_{s1} , T_{s2} – internal and external cylinder liner wall temperature, respectively
- δ_s – wall thickness.

Due to significant temperature and velocity changes of working medium contained in cylinder in different phases of working cycle, a value of the coefficient α is calculated in function of crank angle, in accordance with the empirical formula [5] :

$$\alpha(\varphi) = 130\, d^{-0.2} p(\varphi)^{0.8} T(\varphi)^{-0.53} \cdot \left[C_1 C_m + C_2 \frac{V_s T_1}{p_1 V_1} (p(\varphi) - p_{ob}) \right]^{0.8} \tag{9}$$

where :

- φ [deg] - crank angle
- d [m] - cylinder diameter
- p [bar] - working medium pressure in cylinder
- C_m [m/s] - mean piston speed
- $C_1 = 6.18$ - for compression, combustion and expansion phase, 2.28 - for charge exchange phase
- $C_2 = 0.00324$ [m/s K] - for integral combustion chamber, 0.00622 [m/s K] - for divided combustion chamber
- V_s [m^3] - piston displacement
- T_1 , p_1 ,V_1 - working medium parameters at the beginning of compression
- p_{ob} [bar] - pressure in cylinder in case of external drive.

The delivered energy flow is determined as the total energy contained in the flow of fuel oil delivered to the cylinder, and in the flow of charging air :

$$\dot{Q}_d = \dot{m}_f W_f + \dot{m}_d h_d \tag{10}$$

The cylinder internal work flow N_i is transferred through crankshaft – piston system to drive engine shaft . Its magnitude depends on working medium pressure in the cylinder and on engine rotation speed. The latter is determined by means of the ship motion model [2] considered as an energy consumer model.

The flow of energy contained in cylinder exhaust gas, $\dot{Q}_{ex}$, is determined as the enthalpy of cylinder exhaust gas flows :

$$\dot{Q}_{ex} = h_{ex} \dot{m}_{ex} \tag{11}$$

The flow of heat exchanged with cylinder walls by working medium can be considered as a source of the particular heat flows absorbed by cooling media: that of cylinder liner cooling water, $\dot{Q}_{wc}$, of cylinder head cooling water, $\dot{Q}_g$, of piston cooling water, $\dot{Q}_{wt}$, of lubricating oil, $\dot{Q}_{ol}$, as well as of the environment, $\dot{Q}_{ot}$:

$$\dot{Q}_c = \dot{Q}_{wc} + \dot{Q}_g + \dot{Q}_{wt} + \dot{Q}_{ol} + \dot{Q}_{ot} \tag{12}$$

If water temperature distribution within cylinder cooling jacket is known [3], the flow of heat absorbed by cylinder cooling water, $\dot{Q}_{wc}$, can be determined from the equation :

$$\dot{Q}_{wc} = \dot{m}_w c_w \left[T_{w,1} - T_{w,n} \right] \tag{13}$$

where :

- $T_{w,1}$, $T_{w,n}$ – cooling water temperature at inlet to and outlet from cylinder jacket, respectively.

The flow of heat transferred through cylinder head walls in steady state can be determined by using the following equation :

$$\dot{Q}_g = \alpha A_g (T - T_{og}) \tag{14}$$

where :

- A_g , T_{og} – area and temperature of head's lower plate surface, repectively.

The flow of heat absorbed by cylinder head cooling water, $\dot{Q}_{wg}$, is determined as the sum of the heat transferred by head's lower plate, $\dot{Q}_g$, and the heat flow given up by exhaust gas to valve and exhaust channel walls. The latter is estimated to be 10% of the total heat transferred to cooling medium [1,4].

Similarly the flow of heat transferred by piston head walls in steady state can be determined by using the equation :

$$\dot{Q}_{wt} = \alpha A_t (T - T_t) \tag{15}$$

where :

- α - working medium convective-heat coefficient
- A_t , T_t - area and temperature of piston head surface, respectively.

The flow of heat given up to the environment is determined as the heat absorbed by the engine outer surface having the temperature equal to that of cylinder block. This is expressed by the equation :

$$\dot{Q}_{ot} = \alpha_{ot} A_b (T_b - T_{ot}) \tag{16}$$

where :

- α_{ot} - ambient air convective-heat coefficient
- A_b ,T_b - area and temperature of outer surface of cylinder block, respectively.

The flow of heat absorbed by lubricating oil, $\dot{Q}_{ol}$, can be determined as the rest of thermal energy balance (12) :

$$\dot{Q}_{ol} = \dot{Q}_c - \dot{Q}_{wc} - \dot{Q}_{wt} - \dot{Q}_g - \dot{Q}_{ot} \tag{17}$$

Fig.2 and 3, show the information on distribution of the energy flowing through 6ZA40S engine, given by its manufacturer [6]. In Fig.2. is presented the percentage distribution of the energy of fuel oil delivered to the engine, into : mechanical energy, heat transferred to cooling media and environment, as well as energy contained in exhaust gases. The balance was elaborated for nominal conditions of the engine's operation in the tropical zone and it did not account for the heat recovered in waste heat boiler. However in Fig.3 shares of the enumerated flows are shown in function of the engine's load, related to its rated load. Information on the conditions in which the characteristics have been obtained, is lacking.

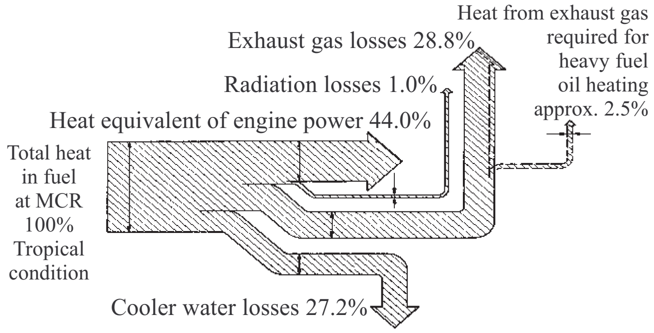


***Fig. 2.** Sankey's energy flow diagram for 6ZA40S engine in nominal conditions acc. [6]*

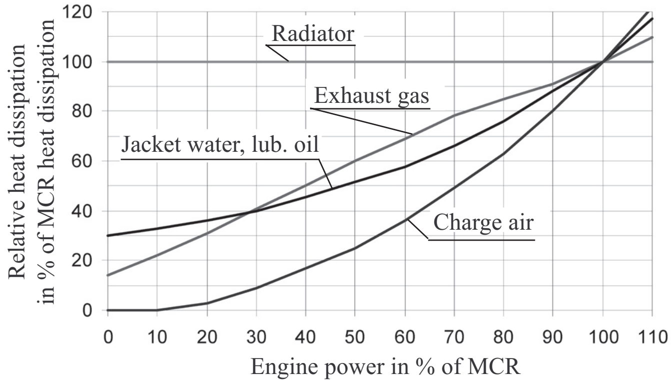


***Fig. 3.** Percentage shares of thermal energy flows through 6ZA40S engine in function of its load acc. [6]*

CALCULATION RESULTS

The model of energy flow and distribution, together with that of heat exchange process in the cylinder-environment system, was programmed in Borland Pascal code of Delphi package. Calculations of the model were carried out by applying changeable integration step with the use of a PC fitted with Pentium II processor [2].

In Table the results of calculation of the model of energy distribution within cylinder are presented for different engine load levels. The following flows and their percentage shares in the balance : mechanical energy (cylinder indicated power) (item. 3), heat transferred from working medium through combustion chamber walls (item. 4), heat contained in gas exhausted from cylinder (item. 5), heat transferred through cylinder liner and head walls (item. 7 and 8), heat absorbed by lubricating oil (item. 10) , as well as heat given up to the environment (item. 11).

The difference between percentage share of the energy delivered to and that transferred from cylinder in steady state was shown in item 13 as the relative error of calculations. In reality the value has not necessarily to be equal to zero; it equals internal energy increment of working medium. This is due to a nonlinearity of cyclic work of the engine.

The calculated share of mechanical energy appeared higher by abt. 2% than that presented in the Sankey's diagram (Fig. 2) for 100% engine load. Hence it results that the value given in the diagram is the percentage share of the effective power measured at the engine shaft.

From Fig. 2. it results that the value given by the engine's manufacturer as the share in the energy balance of the heat absorbed by cooling water, is understood as the heat absorbed by all cooling systems (that of fresh water, lubricating oil, and charging air); and the share of the exhaust gas energy is calculated for exhaust gas behind the turbine.

On the basis of the calculation results the energy balance and its diagram for the modelled engine was elaborated, as shown in Fig.4.

Calculation results of energy distribution in 6ZA40S engine

No	Item			Desired load factor			
1	Engine load		%	100	90	75	50
2	Flow of energy contained in fuel oil	Q_f	kW	1187	1061	915	634.6
			%	100	100	100	100
3	Cylinder indicated power	N_i	kW	555	494	421	284
			%	46.8	46.6	46.0	44.8
4	Flow of heat transferred through walls from working medium	Q_c	kW	207	188	166	121
			%	17.4	17.7	18.1	19.1
5	Flow of heat contained in cylinder exhaust gas	Q_{ex}	kW	412	359	312	215
			%	34.7	33.8	34.1	33.9
6	Flow of heat contained in exhaust gas behind turbine	Q_{ex}	kW	362.0	309.8	269.9	190.4
			%	30.5	29.2	29.5	30
7	Flow of heat absorbed by cylinder liner cooling water	Q_{wc}	kW	76.5	70.8	61.2	42.5
			%	6.4	6.7	6.7	6.7
8	Flow of heat transferred through walls of cylinder head's lower plate	Q_g	kW	46.2	43.4	43	41.9
			%	3.9	4.1	4.7	6.6
9	Flow of heat absorbed by cylinder head cooling water	Q_{wg}	kW	66.9	62.2	59.6	54
			%	5.6	5.9	6.5	8.5
10	Flow of heat transferred to lubricating oil	Q_{ol}	kW	79.4	69.1	57.6	32.8
			%	6.7	6.5	6.3	5.2
11	Flow of heat given up to environment	Q_{ot}	kW	4.9	4.7	4.2	3.8
			%	0.4	0.4	0.5	0.6
12	Cylinder energy balance (3 + 4 + 5)		%	98.9	98.1	98.3	97.7
13	Relative error		%	-1.09	-1.86	-1.74	-2.30

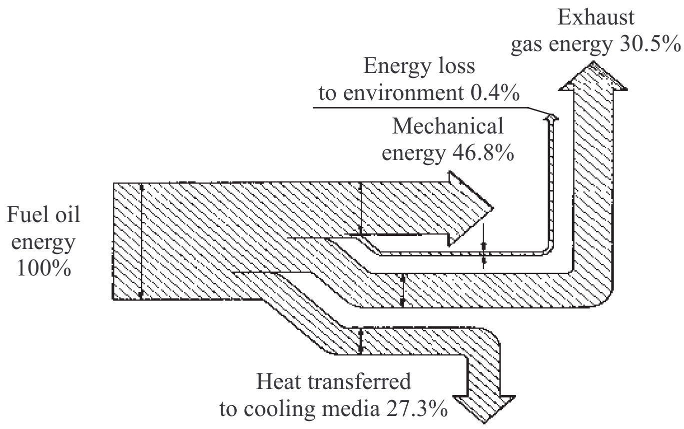


***Fig. 4.** Sankey's energy flow diagram for 6ZA40S engine model at 100% load in nominal conditions*

On the basis of comparison of the results presented in Fig.3 and the results of calculation of the model it can be stated that the first has been elaborated for constant rated speed of the engine.

The research on dynamics of the processes was presented in the author's paper [3].

CONCLUSIONS

- In the paper the physical and mathematical model of energy flow and distribution within the engine were presented, which makes it possible to determine instantaneous values of particular energy flows in function of engine load.
- The model was exemplified by means of 6Z40S engine (installed as the main engine on B672 ship) and the research results were compared with the test-bed measurements of the engine.

- The condition of formal correctness of the modelled process, i.e. energy balance of the engine, was satisfied in every step of calculations at an acceptably low error.
- The model can be used as a tool for investigation of the engine operation process, influence of engine parameters on the process, as well as for diagnosing the engine technical state.
- Investigating the model makes it possible to provide new recommendations important for laboratory experiments and conditions of carrying out empirical tests.
- The model was used for programming a ship propulsion system simulator.

NOMENCLATURE

a_s - temperature equalization coefficient [m^2/s]
A - surface area [m^2],
c_v - specific heat [J/kgK]
F - ratio of combusted fuel mass and charging air mass
h - specific enthalpy [J/kg]
H - enthalpy [J]
m - mass [kg],
$\dot{m}$ - mass flow [kg/s]
N - power [W]
p - pressure [Pa]
Q - heat, energy [J]
Q_c - heat exchanged with walls of combustion chamber [J]
$\dot{Q}$ - heat flow, power [W]
T - temperature [K]
u - specific internal energy [J/kg]
U - internal energy [J]
V - volume [m^3]
W - calorific value [J/kg]

α - convective heat-transfer coefficient [$W/m^2 K$]
δ - thickness [m]
φ - crank angle [deg]
λ - thermal conductance [W/m K]
τ - time [s]

Indices

a - of air
b - of cylinder block
d - of inlet
ex - of exhaust gas
f - of fuel
fb - of combusted fuel oil
i - indicated
ot - of environment
s - of wall
w - of cooling water

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Conference

Jubilee Symposium

On 18÷19 November 2004

25th Symposium on Ship Power Plants

was held in Gdańsk.

As it can be observed the scientific conference gathering experts working in the domain of ship powering systems has had long tradition. It has been born from the idea of yearly meetings to exchange knowledge and experience in this area among scientific workers of five Polish Baltic Sea coast universities, namely :

- Gdańsk University of Technology
- Technical University of Szczecin
- Gdynia Maritime University
- Maritime University of Szczecin
- Naval University of Gdynia.

Every 5th year each of them serves as the organizer of a sucessive symposium in question. The 25th Symposium was organized by the Department of Ship Power Plants, Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology. Its scope was very wide and covered the following topics :

Design, manufacturing and operation processes of ship power plants and their equipment with accounting for reliability, safety and environment protection.

It was reflected in 43 presented papers in preparation of which the representatives of Gdynia Maritime University with 11 papers, and Naval University with 10 papers contributed to the largest extent. The remaining universities took part in the Symposium: Gdańsk University of Technology - with 9 papers, Maritime University of Szczecin - with 8 papers, and Technical University of Szczecin - with 5 papers.

The Symposium program was enriched by presentation of products of several firms :

- ENAMOR - measurement instruments
- UNITEST - educational simulators
- MAN - B&W - ship engines
- Alfa Laval - fuel oil purifying devices
- INTERMASZ - filtration membrane devices.

UniSail

This is an educational program
of Gdańsk University of Technology under the heading :

Enhancement of the University Education Process through Sailing Activities

The basic idea of the project is to make use of the attractive features of sailing for social, organizational and scientific activation of the University's students and employees.

The activation having to start the process of stimulating students to be interested in sailing will be realized within 5 activity groups :

- The **1st** group of activities consists of a cycle of optional lectures widening education of students. The three-semester cycle of lectures will contain such topics as : yacht building, physical background of sailing as well as marketing problems in small undertakings such as sailing activities. This scope of themes has to provide not only technical knowledge but also to demonstrate a role of organizational and managing activities in work of high qualified personnel. The lectures in question will be prepared and given by experienced didactic workers. Didactic materials will be elaborated and published in a multimedial form.
- The **2nd** group of activities covers preparation of production and building process of two sailing yachts (of about 5 m in length) intended for navigation both in sheltered waters and inland waterways. The building of the yachts has to be an attempt to create an equipment base for training, sport competition and turism of students. It has also to give an opportunity to do student's professional practice as well as to gain skill in collective activity, coordination of activities, planning, in order to be aware of the role of organization and management in engineer's work.
- The **3rd** group of activities covers sailing instruction camps (courses). Their aim is to propagate sport and to give an opportunity of gaining yachtsman's degrees. One of the courses has to prepare a group of about 20 sailing instructors – from among students and the University's employees – who would be able to propagate sailing by taking care of students intersted in getting the skill of sailing. The remaining sailing courses have to give an occasion to be acquainted with sailing sport for the group of 60 students. A term for participation in the sailing camps will be obligatory participation in educational part of the project ("No free lunch" policy).
- The **4th** form will be organizational activities associated with servicing the "small-business" undertakings. The problems will be included into the production process of yachts as well as medial servicing process of the project. Students and teachers will realize the activities as it will give them an opportunity to be acquainted with practical aspects of realization of engineering undertakings in which not only technology itself but also economy, organization and management plays an important role.
- The **5th** activity form will be preparation of an internet page and multimedial materials about the project. This task will be realized by the University's employees and students within the frame of practical classes. The students will have an opportunity to apply the newest informatic technologies to the realized project.

An additional activity will be a sociological research on perception of sailing among students. **It is important to know** : how much sailing is attractive for students, which sailing activities are the most popular, whether sailing is considered to be important for their future work and studies. **The research has to answer the questions** : which way sailing should be included into educational processes of the university, and to which extent it would be possible.

During realization of the project a set of materials, designs, analyses and lectures making it possible to adapt the project to the conditions of another high school, will be elaborated.

The project's results will be discussed during
a seminar finalizing the project's realization.

Teachers from the universities of the North-coast region, sailing instructors, sport organizers and representatives of local authorities will take part in the seminar. It will make it possible to disseminate the educational ideas on which the project is based , to transfer the gained experience to other circles, to initiate cooperation of the circles as well as to assess potential of technical sports for the widening of educational offer of the high schools.

The Gdańsk University of Technology serves as a leader of the project, and its didactic and scientific workers as well as those of the State High Professional School in Elbląg and activists of Academic Maritime Club realize the program.

Photo: C. Spigarski

Some aspects of vibration control

Part II: An optimal active controller

Part I : „Active and passive correction” was published in No. 4/2004 of this journal

Krzysztof Kosowski
Gdańsk University of Technology

ABSTRACT

The paper presents a theoretical method for determining the optimal correction to be introduced in a mechanical system. The active control of harmonic vibrations may be achieved by applying a control unit which ought to reduce the vibration amplitude of the selected elements of the system. The proposed method makes it possible to determine the controller parameters which provide an optimum value of the chosen quality index. This criterion includes the reduction of weighted amplitudes of the elements on the one hand, and minimizes energy of the control signal on the other. The described method is suitable for determination of an optimum controller of turbine rotor vibrations caused by bearing oil whip, bearing oil whirl or aerodynamic forces. For the case of rotor self-excited vibrations of aerodynamic type the linear model of excitations was compared with the neural network method.

Key words : active control, mechanical vibrations, optimal controller

PROBLEM DESCRIPTION

Let us consider a mechanical system with *n* degrees of freedom. Its steady harmonic vibrations at the frequency ω_i are described by the following equation:

$$\mathbf{q}_i = \mathbf{G}_i \cdot \mathbf{f}_i \qquad (28)$$

where :

- $\mathbf{f}_i$ - vector of amplitudes of harmonic forces (or moments) acting upon the inertial elements of system
- $\mathbf{q}_i$ - vector of displacement amplitude of the inertial elements of system
- $\mathbf{G}_i$ - system dynamic flexibility matrix.

When taking into consideration vibrations with *k* different values of the frequency ω one can write :

$$\mathbf{q} = \mathbf{G} \cdot \mathbf{f} \qquad (29)$$

The active control of the system vibrations is achieved by applying a controller. The block diagram of the system and its control unit are presented in Fig.11 (for details see Part I of the paper).

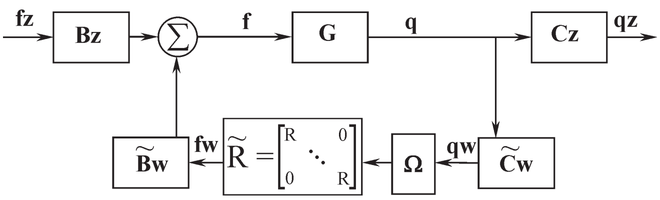

Fig 11. Block diagram of active control of mechanical vibrations

All of the vectors **f**, **fz**, **fw**, **q**, **qz**, **qw** in the generalized scheme are column vectors composed of the corresponding vectors for a single frequency:

$$\mathbf{f} = \begin{bmatrix} \mathbf{f}_1 \\ \mathbf{f}_2 \\ \vdots \\ \mathbf{f}_k \end{bmatrix} \quad \mathbf{fz} = \begin{bmatrix} \mathbf{fz}_1 \\ \mathbf{fz}_2 \\ \vdots \\ \mathbf{fz}_k \end{bmatrix} \quad \mathbf{fw} = \begin{bmatrix} \mathbf{fw}_1 \\ \mathbf{fw}_2 \\ \vdots \\ \mathbf{fw}_k \end{bmatrix}$$

$$\mathbf{q} = \begin{bmatrix} \mathbf{q}_1 \\ \mathbf{q}_2 \\ \vdots \\ \mathbf{q}_k \end{bmatrix} \quad \mathbf{qz} = \begin{bmatrix} \mathbf{qz}_1 \\ \mathbf{qz}_2 \\ \vdots \\ \mathbf{qz}_k \end{bmatrix} \quad \mathbf{qw} = \begin{bmatrix} \mathbf{qw}_1 \\ \mathbf{qw}_2 \\ \vdots \\ \mathbf{qw}_k \end{bmatrix}$$

The matrices **G**, Ω, $\widetilde{\mathbf{R}}$, $\widetilde{\mathbf{Bw}}$, $\widetilde{\mathbf{Cw}}$, **Bz**, **Cz** are described by the following formulas :

$$\mathbf{G} = \begin{bmatrix} \mathbf{G}_1 & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \mathbf{G}_2 & \cdots & \mathbf{0} \\ \vdots & \vdots & \ddots & \vdots \\ \mathbf{0} & \mathbf{0} & \cdots & \mathbf{G}_k \end{bmatrix} \quad \Omega = \begin{bmatrix} \Omega_1 & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \Omega_2 & \cdots & \mathbf{0} \\ \vdots & \vdots & \ddots & \vdots \\ \mathbf{0} & \mathbf{0} & \cdots & \Omega_k \end{bmatrix}$$

$$\widetilde{\mathbf{R}} = \begin{bmatrix} \mathbf{R} & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \mathbf{R} & \cdots & \mathbf{0} \\ \vdots & \vdots & \ddots & \vdots \\ \mathbf{0} & \mathbf{0} & \cdots & \mathbf{R} \end{bmatrix} \quad \widetilde{\mathbf{Bw}} = \begin{bmatrix} \mathbf{Bw} & \mathbf{0} & \cdots & \mathbf{0} \\ \mathbf{0} & \mathbf{Bw} & \cdots & \mathbf{0} \\ \vdots & \vdots & \ddots & \vdots \\ \mathbf{0} & \mathbf{0} & \cdots & \mathbf{Bw} \end{bmatrix}$$

$$\widetilde{\mathbf{Cw}}=\begin{bmatrix}\mathbf{Cw} & \mathbf{0} & \cdots & \mathbf{0}\\ \mathbf{0} & \mathbf{Cw} & \cdots & \mathbf{0}\\ \vdots & \vdots & \ddots & \vdots\\ \mathbf{0} & \mathbf{0} & \cdots & \mathbf{Cw}\end{bmatrix} \quad \mathbf{Bz}=\begin{bmatrix}\mathbf{Bz}_1 & \mathbf{0} & \cdots & \mathbf{0}\\ \mathbf{0} & \mathbf{Bz}_2 & \cdots & \mathbf{0}\\ \vdots & \vdots & \ddots & \vdots\\ \mathbf{0} & \mathbf{0} & \cdots & \mathbf{Bz}_k\end{bmatrix}$$

$$\mathbf{Cz}=\begin{bmatrix}\mathbf{Cz}_1 & \mathbf{0} & \cdots & \mathbf{0}\\ \mathbf{0} & \mathbf{Cz}_2 & \cdots & \mathbf{0}\\ \vdots & \vdots & \ddots & \vdots\\ \mathbf{0} & \mathbf{0} & \cdots & \mathbf{Cz}_k\end{bmatrix}$$

It is assumed that external forces **fz** act on selected inertial elements in a way which can be described by the binary matrix **Bz**. The vector **qz** represents the vibration amplitudes whose value has to be minimized. The selection of these amplitudes from the vector **q** is performed by means of the binary matrix **Cz**. In real mechanical systems it is possible to measure the amplitudes **qw** of vibrations of certain elements only. The vector **qw** can be obtained from vector **q** by multiplying it by the binary matrix **Cw**. Based on the vector **qw**, the controller (characterized by the parameters matrix **R**) creates the output signal **fw**, by using the transfer matrix $\tilde{\mathbf{R}}\cdot\mathbf{\Omega}$. This signal is introduced to some possible elements of the system, selected by the binary matrix **Bw**.

The system presented in Fig.11
can be described by the following set of equations :

$$\begin{cases}\mathbf{q}=\mathbf{G}\cdot\mathbf{f} & (30)\\ \mathbf{f}=\mathbf{Bz}\cdot\mathbf{fz}+\mathbf{Bw}\cdot\mathbf{fw} & (31)\\ \mathbf{qz}=\mathbf{Cz}\cdot\mathbf{q} & (32)\\ \mathbf{qw}=\mathbf{Cw}\cdot\mathbf{q} & (33)\\ \mathbf{fw}=\tilde{\mathbf{R}}\cdot\mathbf{\Omega}\cdot\mathbf{qw} & (34)\\ \tilde{\mathbf{R}}=\sum_{i=1}^{k}\left[(e_i\times\mathbf{I}_s)\mathbf{R}(e_i^T\times\mathbf{I}_{3r})\right] & (35)\end{cases}$$

In the case of a PID controller,
the matrix **R** takes the following form :

$$\mathbf{R}=[\mathbf{K}_P\,|\,\mathbf{K}_I\,|\,\mathbf{K}_D] \qquad (36)$$

where :

$\mathbf{K}_P$, $\mathbf{K}_I$ and $\mathbf{K}_D$ are matrices of the proportional (P), integrating (I) and differentiating (D) action, respectively.

The optimization problem is to determine a controller matrix **R** which minimizes the amplitudes **qz** of chosen elements, taking into account the role of particular elements. This is obtained by assigning, to each of them, an appropriate coefficient from the weight matrix **Φ**. On the other hand, the energy of the control signals **fw** must be taken into account by applying the weight matrix **Λ**. Thus, the optimization index E may be written as follows :

$$E=\mathrm{tr}\,(\mathbf{qz}^*\cdot\mathbf{\Phi}\cdot\mathbf{qz}+\mathbf{fw}^*\cdot\mathbf{\Lambda}\cdot\mathbf{fw}) \qquad (37)$$

where :

tr denotes the trace of a matrix, while the superscript asterisk of a matrix symbol stands for a transposed conjugate matrix.

From the set of equations (30÷35)
the following relation can be derived :

$$\mathbf{fw}=\mathbf{Z}\cdot\mathbf{Cw}\cdot\mathbf{G}\cdot\mathbf{Bz}\cdot\mathbf{fz} \qquad (38)$$

where the matrix **Z** is defined by the formula :

$$\mathbf{Z}:=\tilde{\mathbf{R}}\cdot\mathbf{\Omega}(\mathbf{I}-\mathbf{Cw}\cdot\mathbf{G}\cdot\mathbf{Bw}\cdot\tilde{\mathbf{R}}\cdot\mathbf{\Omega})^{-1}= =\sum_{i=1}^{k}\left[(e_i\times\mathbf{I}_s)\mathbf{R}(e_i^T\times\mathbf{I}_{3r})\right]\mathbf{\Omega}\cdot \cdot\left\{\mathbf{I}-\mathbf{Cw}\cdot\mathbf{G}\cdot\mathbf{Bw}\sum_{i=1}^{k}\left[(e_i\times\mathbf{I}_s)\mathbf{R}\cdot\mathbf{\Omega}(e_i^T\times\mathbf{I}_{3r})\right]\right\}^{-1} \qquad (39)$$

Furthermore, from the equations (30÷35,38)
it is possible to show that :

$$\mathbf{q}=\mathbf{G}(\mathbf{I}+\mathbf{Bw}\cdot\mathbf{Z}\cdot\mathbf{Cw}\cdot\mathbf{G})\mathbf{Bz}\cdot\mathbf{fz} \qquad (40)$$

The equations (32) and (39) lead directly to the relation :

$$\mathbf{qz}=\mathbf{Cz}\cdot\mathbf{G}(\mathbf{I}+\mathbf{Bw}\cdot\mathbf{Z}\cdot\mathbf{Cw}\cdot\mathbf{G})\mathbf{Bz}\cdot\mathbf{fz} \qquad (41)$$

By substituting **fw** and **qz** in the equation (37) for the right sides of the equations (38) and (41) respectively, the quality index E may be expressed directly as a function of the parameters of the mechanical system and the controller, the system input **fz** and the weight matrices **Φ** and **Λ** :

$$E=\mathrm{tr}\Big\{\mathbf{fz}^*\cdot\mathbf{Bz}^*\,[(\mathbf{I}+\mathbf{G}^*\cdot\mathbf{Cw}^*\cdot\mathbf{Z}^*\cdot\mathbf{Bw}^*)\mathbf{G}^*\cdot\mathbf{Cz}^*\cdot\mathbf{\Phi}\cdot\mathbf{Cz}\cdot\mathbf{G}\cdot \cdot(\mathbf{I}+\mathbf{Bw}\cdot\mathbf{Z}\cdot\mathbf{Cw}\cdot\mathbf{G})+\mathbf{G}^*\cdot\mathbf{Cw}^*\cdot\mathbf{Z}^*\cdot\mathbf{\Lambda}\cdot\mathbf{Z}\cdot\mathbf{Cw}\cdot\mathbf{G}]\mathbf{Bz}\cdot\mathbf{fz}\Big\} \qquad (42)$$

The optimal value of the index E for a controller matrix **R** must occur in a stationary point of the function E(**R**), i.e. for a matrix **R**, which fulfills the relation :

$$\frac{\partial E}{\partial\mathbf{R}}=0 \qquad (43)$$

The condition for all partial derivatives with respect to single coefficients $\partial\mathbf{R}[i,j]$ of the matrix **R** may be written in the form :

$$\frac{\partial E}{\partial\mathbf{R}[i,j]}=\sum_k\sum_l\frac{\partial E}{\partial\mathbf{Z}[k,l]}\cdot\frac{\partial\mathbf{Z}[k,l]}{\partial\mathbf{R}[i,j]}=0 \qquad (44)$$

It is possible to conclude from the relations (39) and (44), that the condition (43) will be always fulfilled for k=1 (which refers to the case in which a single frequency is considered) as long as :

$$\frac{\partial E}{\partial\mathbf{Z}}=0 \qquad (45)$$

The formula of the derivative $\frac{\partial E}{\partial\mathbf{Z}}$ can be obtained from the equation (42), with the aid of formulas given in [1,3]. The result may be written in the following form :

$$\frac{\partial E}{\partial\mathbf{Z}}=2\left[\begin{array}{c}\mathbf{Bw}^*\cdot\mathbf{G}^*\cdot\mathbf{Cz}^*\cdot\mathbf{\Phi}\cdot\mathbf{Cz}+\\ +(\mathbf{Bw}^*\cdot\mathbf{G}^*\cdot\mathbf{Cz}^*\cdot\mathbf{\Phi}\cdot\mathbf{Cz}\cdot\mathbf{G}\cdot\mathbf{Bw}+\mathbf{\Lambda})\mathbf{Z}\cdot\mathbf{Cw}\end{array}\right]\cdot \cdot\mathbf{G}\cdot\mathbf{Bz}\cdot\mathbf{fz}\cdot\mathbf{fz}^*\cdot\mathbf{Bz}^*\cdot\mathbf{G}^*\cdot\mathbf{Cw}^* \qquad (46)$$

From the condition (45) for the extremum of function E(**Z**), the matrix $\mathbf{Z}_{opt}$ matching the optimal controller matrix $\mathbf{R}_{opt}$, can be determined :

$$\mathbf{Z}_{opt}=-(\mathbf{Bw}^*\cdot\mathbf{G}^*\cdot\mathbf{Cz}^*\cdot\mathbf{\Phi}\cdot\mathbf{Cz}\cdot\mathbf{G}\cdot\mathbf{Bw}+\mathbf{\Lambda})^{-1}\cdot \cdot\mathbf{Bw}^*\cdot\mathbf{G}^*\cdot\mathbf{Cz}^*\cdot\mathbf{\Phi}\cdot\mathbf{Cz}\cdot\mathbf{G}\cdot\mathbf{Bz}\cdot\mathbf{fz}\cdot\mathbf{fz}^*\cdot\mathbf{G}^*\cdot\mathbf{Cw}^*\cdot \cdot(\mathbf{Cw}\cdot\mathbf{G}\cdot\mathbf{Bz}\cdot\mathbf{fz}\cdot\mathbf{fz}^*\cdot\mathbf{Bz}^*\cdot\mathbf{G}^*\cdot\mathbf{Cw}^*)^{-1} \qquad (47)$$

The matrix $\mathbf{Z}_{opt}$ exists if the two matrices inverted in the above given equation, i.e.: $(\mathbf{Bw}^*\cdot\mathbf{G}^*\cdot\mathbf{Cz}^*\cdot\mathbf{\Phi}\cdot\mathbf{Cz}\cdot\mathbf{G}\cdot\mathbf{Bw}+\mathbf{\Lambda})$

and $(\mathbf{Cw}\cdot\mathbf{G}\cdot\mathbf{Bz}\cdot\mathbf{fz}\cdot\mathbf{fz}^{*}\cdot\mathbf{Bz}^{*}\cdot\mathbf{G}^{*}\cdot\mathbf{Cw}^{*})$ are nonsingular. If the conditions are fulfilled, the optimal matrix $\mathbf{R}_{opt}$ exists and may be easily calculated from the relations (39) and (47) :

$$\mathbf{R}_{opt} = (\mathbf{I} + \mathbf{Z}_{opt}\cdot\mathbf{Cw}\cdot\mathbf{G}\cdot\mathbf{Bw})^{-1}\mathbf{Z}_{opt}\cdot\mathbf{\Omega}^{-1} \quad (48)$$

The final values of the controller matrix **R** should be determined by taking into account some additional limits or criteria, for example stability requirements.

EXAMPLES OF APPLICATION

A ship propulsion unit

An example of a ship propulsion unit equipped with flexible couplings, shown in Fig.12, was described in Part I of this paper. It was shown that the controller using the angular velocity of the generator as its correction signal offers the largest possibilities of reducing torsional vibration amplitude in the main coupling, Fig.13.

The linear dynamic model of this power plant was compared with experimental data, followed by an analysis of the vibrations due to operation of the engine with one misfiring cylinder [2]. By using the method presented in this paper, the optimal matrix **R** of the parameters of the additional controller was estimated, under the assumption that the values of parameters should be contained within a range of practical applications. The optimum parameters of the matrix **R** were determined as those fulfilling the criterion (43) for minimizing the quality index E in the range of vibration frequencies 12÷26 Hz.

The comparison of the amplitude of the fundamental harmonic frequency of the torque acting on the main flexible coupling is presented in function of engine speed in Fig.13 for the cases with and without correction. By using the method described in Part I and II of the paper, it turned out possible to change the structure and parameters of the main engine control system and to remarkably decrease torque amplitudes (much below the hazardous limit).

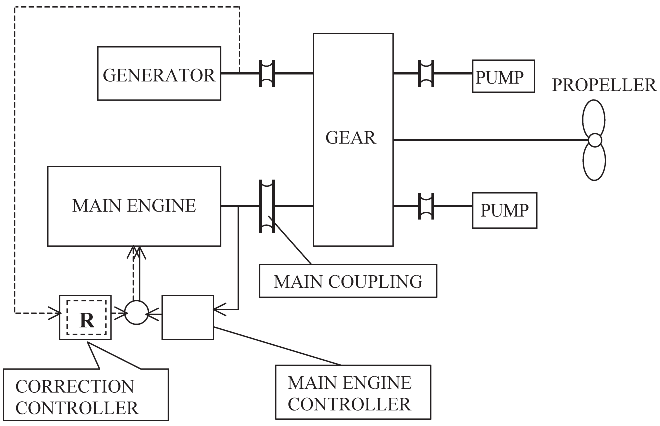

Fig 12. *Ship propulsion unit with the correction loop for reduction of main coupling vibrations*

Turbine rotor self-excited vibrations

Currently conducted work (see Part I of this paper) concentrates on active control of rotor vibrations of a 200 MW steam turbine by means of pressurized bearings. The aim is to investigate the possibilities of applying the pressurized bearings to the large output steam turbine rotor to reduce self-excited vibration caused by oil whip or oil whirl, as well as aerodynamic excitations.

The proposed method is used for :

- determining the optimum construction of the applied pressurized bearings (i.e. location of the external oil supply, Fig.14a)

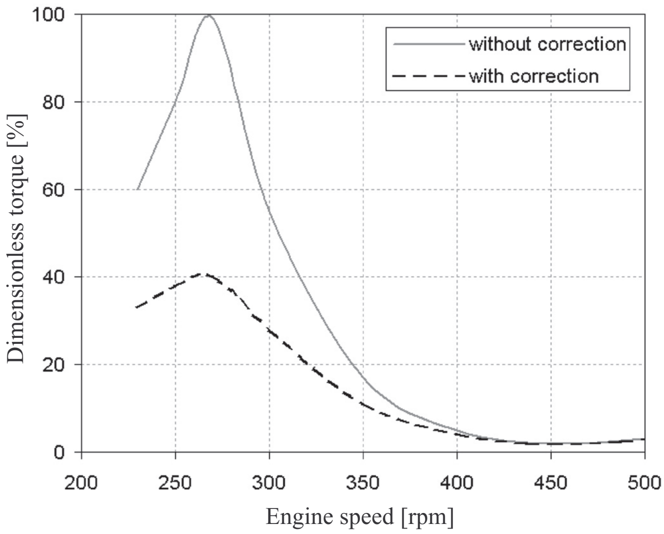

Fig 13. *Comparison of the main coupling torque (current torque related to the maximum torque for the case without correction)*

- finding the optimal controller for active control of the construction parameters of the bearing supports, Fig.14b
- determining the optimum structure of the active control system and the optimum controller parameters for governing the rotor vibrations by means of pressurized bearings, Fig.15.

After research is completed, the results will be presented in a separate paper.

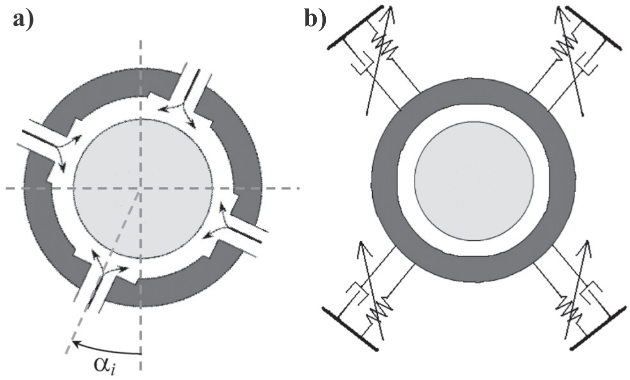

Fig 14. *Diagrams for determining optimum location of external oil supplying (**a**) and for determining optimum controller for active control of bearing supports (**b**)*

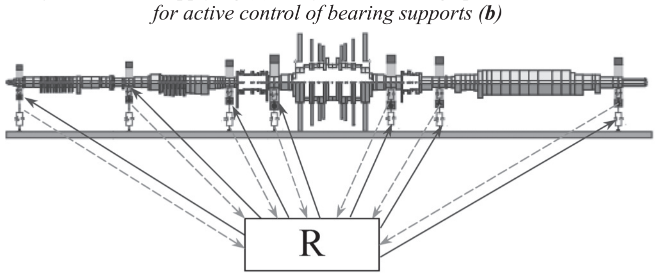

Fig 15. *Diagram for determining the optimum structure of the active control system for governing the turbine rotor vibrations*

CONCLUSIONS

- A theoretical approach for determining the optimal correction loop for improving the dynamic behaviour of a linear mechanical system was described in the paper. The active control of harmonic vibrations was achieved by applying a control unit which reduces the vibration amplitude of selected elements of the system. This method was applied to a ship propulsion unit in order to change the structure and parameters of the main engine control system, and by use of it a remarkable decrease of torque amplitudes was achieved.

- The presented method makes it possible to determine the controller parameters which provide the optimum value of the chosen quality index. This criterion includes the reduction of weighted amplitudes of the elements on the one hand, and minimizes energy of the control signal on the other.
- The proposed method is suitable for the determination of optimum controller of turbine rotor vibrations due to bearing oil whip, bearing oil whirl or aerodynamic forces.

Acknowledgements

The author wishes to express his gratitude to professor Włodzimierz Gawroński (NASA - National Aeronautics & Space Agency). Part I and II of this paper constitute a generalized approach to the problem elaborated together with Prof. Gawroński [4].

NOMENCTLATURE

Bw, **Bz**, **Cw**, **Cz** - binary matrices
e_i - i-th versor of dimension k
E - optimization index
f - vector of force amplitudes
fw - controller output vector
fz - vector of external forces
G - dynamic flexibility matrix
I - unitary matrix
j, k, l, n, r, s - dimensions of matrices and vectors
K - matrix of stiffness coefficients
q - vector of displacement amplitudes
qw - vector of measured amplitudes
qz - vector of amplitudes of controlled elements
R - controller matrix

Φ, **Λ** - weight matrices
ω - frequency
Ω - frequency multiplier matrix

Indices

P - proportional controller
I - integrating controller
D - differentiating controller

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Conference

21st Scientific Session on Marine Technology

On 27-28 September 2004 Polish shipbuilders met in Gdańsk to held their 21st scientific conference which was organized by the Polish Society of Naval Architects and Marine Engineers KORAB, Faculty of Ocean Engineering and Ship Technology of Gdańsk University of Technology (GUT), (which hosted the conference), and North Shipyard Co.

The Conference started with the session on *History of Shipbuilding* which contained six papers prepared by scientific workers of the Faculty of Ocean Engineering and Ship Technology, GUT, namely :

- *Main Directions of the Development of Ships over the Centuries* – by K. Rosochowicz, D. Duda
- *A short Pictorial History of Orthodox and Unorthodox Hydrodynamic Marine Propulsors* – by J.A. Szantyr
- *Outline of Marine Piston Engines' Technology Development* – by J. Girtler
- *Steam Turbine as the Main Ship Propulsion. More than a Hundred Years at Sea* – by K. Kosowski
- *Gas Turbine : Advanced Marine Propulsion* by K. Kosowski
- *Marine Turbines : Cogeneration and Combined Propulsion Systems* – by K. Kosowski

The interesting essence of the papers was enriched by presentation of many illustrations and drawings.

39 papers were read during the technical part of the Conference, which was divided into 7 topical groups :

- Shipbuilding Techniques (7 papers)
- Ship Design and Hydromechanics (6 papers)
- Ship Hydromechanics and Structural Mechanics (7 papers)
- Safety of Ships and Shipping (4 papers)
- Ship Power Plants (6 papers)
- Ship Power Plants and Ship Equipment (4 papers)
- Ship Equipment and Production Management (5 papers).

Representatives of Gdańsk University of Technology and Technical University of Szczecin prepared the greatest number of papers (13 and 11, respectively). Authors and co-authors of the15 remaining papers came from Gdynia Maritime University, Polish Register of Shipping, Naval University of Gdynia, Ship Design and Research Centre – Gdańsk, Wrocław University of Technology, Foundation for Safety of Navigation and Environment Protection, Gdańsk Shiprepair Yard.

Photo: C. Spigarski

Conference

KONES 2004

On 12-15 September 2004 it was 30th time when the international scientific conference on :

Internal Combustion Engines

was held.

This time the conference venue was Zakopane, a Polish health & recreation resort . It was organized by Institute of Aeronautics, Warsaw, Research & Development Enterprise „PRO-MO" Cracow, and Polish Academy of Sciences.

The Conference program contained 65 papers including 9 ones presented during the plenary session on :

- *Perspectives and trends in „Diesel emissions and control"* – by Thimoty V. Johnson (Corning Incorporated, USA)
- *Experimental study on the urea SCR system for medium Duty Track* – by Tazuki Watanabe, Jin Kusaka (Graduate School of Science and Engineering, Waseda University, Tokyo, Japan)
- *TARDEC – National Automotive Center (NAC) Research and Development Overview* – by Alexander Sandel (TARDEC The National Automotive Center – US Army, Waren, USA)
- *Novel pistons for diesel engines* – by Jerzy Nykiel (Federal Mogul S.A., Gorzyce, Poland)
- *Space vehicles propulsion – some concept in reality and fiction* – by Cezary Szczepaniak (Łódź University of Technology, Poland)
- *Facilitation of HCCI combustion of biogas at moderate compression ratios by application of fuel reforming and inlet air heating* – by M. L. Wyszyński, T. Megaritis, J. Karlovsky, D. Yap, S. Peucheret, R. S. Lehrle (The University of Birmingham, UK), H. Xu (Yaguar Cars, Coventry, UK), S. Golunski (Johnson Matthey Technology Centre, Reading, UK)
- *Recent advances in engine air cleaners design and evaluation* – by T. Jaroszczyk, Ch. E. Holm (Fleetguard Inc. Stoughton, US) and Byron A. Pardue (Fleetguard Inc. Cookeville, US)
- *Effect of fuel temperature and ambient pressure on a common rail rape-seed oil spray* – by M. T. Bialkowski, T. Pekdemir, D. P. Towers, R. Reuben (Heriot-Watt University, Edinburgh, Scotland), M. Brautsch (University of Applied Sciences, Amberg-Weiden, Germany) and G. Elsbett (Elsbett Technologies GmbH, Thalmassing, Germany)
- *Effect of composite material on performance and emission characteristics of S.I. engine* - by P. Mahendra Kumar (National Engineering College, Tamilnadu, India) and C.G. Saravanan (Annamalai University, India).

The remaining papers were read during 3 panel sessions devoted to the following themes :

- Ecology, Combustion, Thermodynamic processes (19 papers)
- Fuelling, Measurement, Control (20 papers)
- Construction, Technology, Operating (17 papers).

Moreover, had place a poster session divided into the above enumerated themes, during which 33 elaboration were presented (14, 12, 7, respectively).

Apart from Polish participants 33 representatives of foreign scientific centres of USA (7), Lithuania (5), Japan (5), Great Britain (4), Slovenia (3), Czech Republic (2), Slovakia (2), India (2), Germany (1), Canada (1) and Switzerland (1) took part in the Conference.

Photo: C. Spigarski

*Mount **Giewont** (1909 m), a characteristic peak towering over Zakopane*

Miscellanea

IFFM PUBLISHERS

IFFM Publishers in Gdańsk is a unit of Szewalski Institute of Fluid-Flow Machinery (IFFM), Polish Academy of Sciences (PAS).

The publishing house issues the Institute's periodicals as well as publications of other bodies, for instance Thermodynamics and Combustion Committee and Committee on Energy Problems, PAS, as well as monographies and conference proceedings ordered by universities of Gdańsk sea-coast region.

Among the Institute's peridicals the following should be enumerated :

- Transactions of IFFM – the periodical which has been issued two times a year since 1960, since 1997 only in English
- Scientific Installments of IFFM – (Studia i Materiały, in Polish) – publishing series which presents selected articles in the form of separate dissertations
- Annual Report (Przegląd Prac) – (both in Polish and English) – the yearly report on achievements of IFFM scientific workers, together with comprehensive bibliography

- Archives of Thermodynamics, a quarterly in English
- Archives of Energetics, a periodical mainly in Polish are permanent publications of the above mentioned Committees of Polish Academy of Sciences.