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*Prof. Z. Domachowski, D.Sc. M. Pawłowski, Prof. J. K. Włodarski, Prof. L. Kobyliński, Z. Korczewski, Assoc.Prof.*



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GDAŃSK UNIVERSITY
OF TECHNOLOGY
Faculty
of Ocean Engineering
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ul. Narutowicza 11/12
80-952 Gdańsk, POLAND
tel.: +48 58 347 17 93
fax : +48 58 341 47 12
e-mail : sekoce@pg.gda.pl

**Account number :**
**BANK ZACHODNI WBK S.A.**
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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.
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# Location of ship rolling axis

**Ludwik Balcer**
Gdańsk University of Technology

ABSTRACT

*In the paper is presented a definition of ship rolling axis and a method for determination of location of the axis. Location of rolling axis depends not only on ship mass distribution but also on the so-called „added masses" which are equivalents of the hydrodynamic forces acting on ship hull during its motion. It has been assumed that the rolling axis is fixed with respect to ship's hull, and its location is determined with accounting for the added masses accompanying ship rolling, whereas the influence of damping forces is neglected. For a representative group of ships appropriate coefficients of hydrodynamic forces were calculated on the basis of the experimental test results available in the subject-matter literature. It has been assumed that the rolling axis passes through a common centre of ship's mass and relevant added masses. As a result proposed are simple approximate formulas which make it possible to calculate location of ship rolling axis on the basis of the typical ship data available at the preliminary design stage.*

**Key words** : ship seaworthiness, ship safety, ship hydromechanics, stability of floating units

## INTRODUCTION

The ship rolling axis is a conventional notion which serves mainly to simplified representation of ship motions in the case when ship rolling prevails. Under the assumption that the location of ship rolling axis is known it is easy to determine translations and accelerations in any ship's point, as then they are functions of distance of a given point from the axis in question. In some of the points the placing of some cargoes, measuring instruments, passenger cabins should be avoided or – in reverse – they may appear most favourable for installing given devices. In order to estimate rolling axis location a ship physical model of two degrees of freedom (dof ) was assumed. However if the location is assumed permanent the model can be taken as that of one dof only. In this model it is assumed that the considered ship rolls in calm water due to an external excitation free moment of appropriately selected parameters; it can also roll freely. In this paper the method of determining location of the rolling axis is described and – on this basis – submitted are simplified formulas for calculation of its location, applicable to typical merchant ships.

## DEFINITION OF SHIP ROLLING AXIS

It is usually assumed that the ship rolling axis coincides with X-axis of the coordinate system whose origin is fixed with respect to the ship in its mass centre [3,5]. It is also assumed that the axis which lies on the ship plane of symmetry and initially is parallel to ship waterline, can be considered as the ship's main axis of inertia. With respect to it, calculated is the ship mass moment of inertia $I_x$, an element appearing in equations of motion. The axis is assumed motionless with respect to both the ship and space. The approximate character of the calculations makes it possible to use values of the inertia moments of the assumed form, i.e. $(I_x + I_{xx})$, in spite of the fact, that the inertia moments are not corrected with taking into account a real location of rolling axis. It was this way assumed because the magnitudes of the added mass inertia moments of water, $I_{xx}$, are often determined on the basis of the measured free-rolling period of ship, hence the correct location of rolling axis has been already accounted for. This also concerns the approximate calculation formulas for both $I_x$ and $I_{xx}$ values.

## LOCATION OF SHIP ROLLING AXIS

In order to determine location of ship rolling axis in accordance with the above specified assumptions it is postulated that the ship oscillates in calm water, having two degrees of freedom : roll and sway. In this case the line on the ship's plane of symmetry, where lateral translations in the motionless, earth-fixed coordinate system are equal to zero, can be considered as the rolling axis. The similar approach is presented in [2 , 7].

The following equation of ship's rolling is taken for considerations :

$$m\,\ddot{y}_G + m_{22}\,\ddot{y} + 2\mu_{22}\,\dot{y} + m_{24}\,\ddot{\Phi} + 2\mu_{24}\,\dot{\Phi} = 0$$

and after changing notation : (1)

$$m\,\ddot{y}_G + m_{yy}\,\ddot{y} + 2\mu_{yy}\,\dot{y} + m_{y\Phi}\,\ddot{\Phi} + 2\mu_{y\Phi}\,\dot{\Phi} = 0$$

where :

$y_G$ - lateral translation of ship mass centre
$y$ - lateral translation of the point O (Fig.1)
$\Phi$ - ship rolling angle
$m$ - ship mass
$m_{yy}$ - added mass of water for ship sway
$m_{y\Phi}$ - added mass of water for ship sway in result of rolling with respect to the point $O_0$
$\mu_{yy}$ - damping coefficient for ship sway
$\mu_{y\Phi}$ - damping coefficient for ship sway due to rolling with respect to the point $O_0$

This is the equation which represents the sum of projections – on Y-axis – of the forces acting upon the ship, due to sway. In the equation (1) there are no expressions for restoring forces which really do not appear during sway.

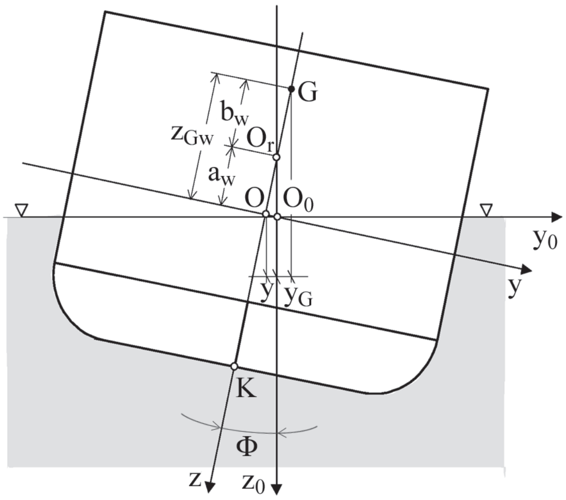


***Fig. 1.*** *The coordinate systems and location of rolling axis*

In Fig.1 are presented : the ship in a heeling position, the motionless, earth-fixed coordinate system $O_0,Y_0,Z_0$, and the movable, ship-fixed coordinate system O,X,Y. In the initial position of the ship both the coordinate systems coincide.

In Fig.1 the point $O_r$ represents the trace of the rolling axis and in accordance with the assumed definition it always remains on $Z_0$- axis of the motionless coordinate system. If in the ship-fixed reference system the quantity $z_{Gw}$ stands for the distance of the mass centre G from the point O, i.e. from the initial ship waterline, then the trace of the rolling axis, $O_r$, splits the segment OG into two parts :

$$z_{Gw} = a_w + b_w$$

For small heeling angles it can be written as follows :

$$y_G = - b_w\Phi \quad \text{and} \quad y = a_w\Phi$$

The quantities y determine shifts of the origin O of the movable coordinate system. The added mass $m_{y\Phi}$ and the damping coefficients $\mu_{yy}$ , $\mu_{y\Phi}$ are determined with respect to the origin O. The here considered damping coefficients $\mu_{yy}$ and $\mu_{y\Phi}$ take into account only dissipation of the energy generated by the radiation forces with neglecting the viscosity forces.

As the segment $a_w = OO_r$, and $b_w = GO_r$, hence $a_w < 0$ when the trace of the rolling axis, $O_r$ , lies over the initial waterline, i.e over the point O, and $b_w < 0$ when $O_r$ lies below the ship mass centre G (Fig.1). Now the equation (1) can be written as follows :

$$\begin{aligned} & - m\, b_w \ddot{\Phi} + m_{yy} a_w \ddot{\Phi} + \\ & + 2\mu_{yy} a_w\, \dot{\Phi} + m_{y\Phi}\ddot{\Phi} + 2\mu_{y\Phi}\dot{\Phi} = 0 \end{aligned} \tag{2}$$

Knowing that $a_w = z_{Gw} - b_w$ ,
and introducing appropriate transformations one obtains :

$$\begin{aligned} & - b_w \left[ \left( m + m_{yy} \right) \ddot{\Phi} + 2\mu_{yy}\ddot{\Phi} \right] = \\ & = - \left( m_{yy} z_{Gw} + m_{y\Phi} \right) \ddot{\Phi} - 2 \left( \mu_{yy} z_{Gw} + \mu_{y\Phi} \right) \dot{\Phi} \end{aligned} \tag{3}$$

If the influence of damping is now neglected, i.e. $\mu_{yy} = 0$ and $\mu_{y\Phi} = 0$, then the following expressions for the distance $b_w$ of the rolling axis point $O_r$ from the ship mass centre G, and for the distance $a_w$ of the point $O_r$ from the initial waterline (i.e. from the point O, Fig.1) are obtained:

$$\begin{aligned} b_w &= \frac{m_{yy} z_{Gw} + m_{y\Phi}}{m + m_{yy}} \\ a_w &= \frac{z_{Gw} m - m_{y\Phi}}{m + m_{yy}} \end{aligned} \tag{4}$$

Therefore, with the lack of damping the rolling axis location is fixed for a given rolling period, and it accounts for the influence of the inertia forces resulting from the water surrounding the ship, here represented by the added masses $m_{yy}$ and $m_{y\Phi}$. If the ship having the same rigidity, rolled in the „free space", i.e. not being immersed in the water, then $m_{yy}$ and $m_{y\Phi}$ would equal zero. In such case the rolling axis would go through the mass centre G, and then it would be :

$$b_w = 0 \text{ and } a_w = z_{Gw} \text{, respectively.}$$

In order to determine rolling axis location for a considered ship it is necessary to calculate the added masses $m_{yy}$ and $m_{y\Phi}$. For the quantities it is hard to find approximate value, which is not the case for e.g. rolling or heaving. The quantities can be calculated analytically by using the „strip theory" and applying the Lewis frame-like forms [3, 5, 7]. However this requires using a special software together with many initial data of a considered ship.

For these reasons a simplified method is proposed below. It can be useful for the approximate determination of rolling axis location on the basis of the ship's data usually available for a designed ship.

## A METHOD OF DETERMINING LOCATION OF ROLLING AXIS

In order to determine the location of rolling axis in which an influence of added water mass is accounted for, a simplified ship representing a given one has been assumed. The simplified ship is a cylinder of rectangular frame-section and its parameters are so selected as to obtain its roll and sway motions as close as possible to those of the basic ship. Hence the ship is of the same mass *m*, B/d ratio and free-rolling period $\tau$. The centre of gravity G of the simplified ship is located, proportionally to its main dimensions, in the same point as that of the real ship. It has also the same transverse moment of inertia of waterplane area, which indirectly gives the same small metacentric radiuses. The last condition makes it possible to determine the main dimensions, $L_s$, $B_s$, $d_s$, of the simplified ship on the basis of the real ship's parameters. It was further assumed that a change of location of rolling axis of the simplified ship, resulting from added mass influence, correctly describes the corresponding change in the basic ship.

Values of the added masses and damping coefficients for the ship of rectangular frame sections (as well as for some other simple forms) are published in [8] and [3] in the form diagrams of the dimensionless values $m'_{yy}$, $m'_{y\Phi}$, of the added masses $m_{yy}$ and $m_{y\Phi}$, in function of the dimensionless ship rolling frequencies $\omega'$.

In order to determine $m'_{yy}$ it is also necessary to know B/d ratio.

Now it is possible to describe successive steps for determination of ship's rolling axis location in compliance with the above made assumption.

Step 1.
**Selection of ship's data**

The demanded ship's data are as follows :

- $L_{bp}$ - length between perpendiculars [m]
- B - breadth [m]
- d - draught [m]
- H - depth to upper deck [m]
- m - ship mass [t]
- d - block coefficient of ship's hull [-]
- a - waterplane coefficient [-]
- $z_G$ - height of the ship mass centre over the ship base plane [m]
- GM - initial transverse metacentric height of the ship, not corrected for free- surface influence [m]

In order to obtain calculation results of a sufficiently general character the data of different ships in various loading conditions were used (Tab.1).

Step 2.
**Determination (calculation) of ship's rolling period $\tau$**

The best solution is to have a value of the rolling period measured on the real ship. However such data are hard available. An approximate value of ship's free-rolling period can be calculated by using the following relationship :

$$\tau = 2\pi\sqrt{\frac{I_x + I_{xx}}{mgGM}} \tag{5}$$

To determine values of $I_x$ and $I_{xx}$ the expressions given in [6] were used :

$$I_x = \frac{m}{12}\left(B^2 + 4z_G^2\right) \qquad I_{xx} = 0.3\, I_x \tag{6}$$

In further considerations the quantity $(I_x + I_{xx})$ is deemed a sufficient approximation of the total moment of inertia of ship mass and added mass of water, respective to ship rolling axis, as it was postulated in its definition.

Step 3.
**Calculation of the main dimensions $L_s$ , $B_s$ , $d_s$ of the simplified ship**

The ship breadth $B_s$ is determined by assuming that values of the moment of inertia, $I_{wx}$, of waterplane areas of the real ship and the simplified one are equal. The $I_{wx}$ of the real ship waterplane is determined by approximating ordinates of its contour, $y_w(x)$, by means of the parabola whose order is expressed by the waterplane coefficient $\alpha$, [4]:

$$y_w(x) = \frac{B}{2}\left[1 - \left(\frac{x}{L/2}\right)^{\frac{\alpha}{1-\alpha}}\right]$$

$$I_{wx} = \frac{LB^3}{2}\cdot\frac{\alpha^3}{(1+\alpha)(1+2\alpha)} \tag{7}$$

Basing on $I_{wx}$ given by (7) and in accordance with the following relationships :

$$\frac{L_s B_s^3}{12} = I_{wx} \quad L_s B_s d_s = L\, B\, d\, \delta \quad \frac{B_s}{d_s} = \frac{B}{d}$$

one obtains the expression for the breadth $B_s$, draught $d_s$ and length $L_s$ of the simplified ship :

$$B_s = \frac{6B}{\delta}\cdot\frac{\alpha^3}{(1+\alpha)(1+2\alpha)}$$
$$d_s = \frac{B_s\, d}{B} \qquad L_s = \left(\frac{B}{B_s}\right)^2 L\,\delta \tag{8}$$

Step 4 and 5.
**Calculation of $\omega'$, read-out of $m'_{yy}$ and $m'_{y\Phi}$ , and calculation of $m_{yy}$ and $m_{y\Phi}$**

The form of the quantities to be calculated, in compliance with the previously applied symbols, is as follows :

$$\omega' = \omega\sqrt{\frac{B_s}{2g}} \qquad m'_{yy} = \frac{m_{yyj}}{\rho A_s}$$
$$m'_{y\Phi} = \frac{m_{y\Phi j}}{\rho A_s B_s} \tag{9}$$
$$m_{yy} = L_s \cdot m_{yyj} \qquad m_{y\Phi} = L_s \cdot m_{y\Phi j}$$

The values $m_{yyj}$ [t/m] and $m_{y\Phi j}$ [tm/m] concern the ship of the one-meter length and of the assumed, simplified frame section.

The remaining symbols stand for :

- $\rho$ – water mass density [t/m$^3$]
- g – acceleration of gravity [m/s$^2$]
- $B_s$ – breadth of the simplified ship [m]
- $A_s$ – ship's frame area [m$^2$] (here : $A_s = B_s \cdot d_s$)

Having read values $m'_{yy}$ and $m'_{yF}$ according to w', one calculates, values $m_{yyj}$ and $m_{yFj}$, and next $m_{yy}$ and $m_{yF}$, by using (9).

Step 6.
**Calculation of rolling axis location**

The calculations were carried out for the simplified ships. First, by using (4) the segments $a_{ws}$ and $b_{ws}$ were calculated and then applied in their relative form by making use of the following relations:

$$a_{ws}/B_s = a_w/B \quad \text{and} \quad b_{ws}/B_s = b_w/B$$

to obtain : $a_w = (a_{ws}/B_s)\cdot B$ and $b_w = (b_{ws}/B_s)\cdot B$

Next, the values of the added masses, $m_{yy}$ and $m_{y\Phi}$, segments $a_{ws}$ and $b_{ws}$, as well as ratios $a_{ws}/B_s$ and $b_{ws}/B_s$ were calculated on the basis of the above presented assumptions and formulas.

The calculations were performed for 9 ships in different loading conditions. Tab.1 contains the input data of the ships, and the calculation results of the relative values of rolling axis location are shown in Tab.2 ( according to the increasing sequence of $z_{Gw}/B$ values).

The segment $a_w = OO_r$ was selected as the quantity determining location of rolling axis. The segment $a_w$ being the distance

*Tab. 1. Input data of 9 selected ships in different loading conditions*

| No. | Ship | Loading cond. | $L_{bp}$ [m] | B [m] | d [m] | m [t] | $\alpha$ [-] | $\delta$ [-] | $z_{GW}$ [m] | $\tau$ [s] |
|---|---|---|---|---|---|---|---|---|---|---|
| 1 | *ferryship* | full | 154.20 | 28.50 | 6.65 | 20660 | 0.915 | 0.672 | - 6.92 | 15.6 |
| 2 | *STENA* | empty | 154.20 | 28.50 | 5.87 | 16160 | 0.815 | 0.672 | - 8.56 | 18.2 |
| 3 | *ferryship* | full | 115.00 | 19.50 | 5.15 | 6680 | 0.755 | 0.546 | - 3.27 | 14.4 |
| 4 | *ROGALIN* | empty | 115.00 | 19.50 | 4.36 | 5532 | 0.725 | 0.529 | - 4.82 | 16.1 |
| 5 | *ferryship* | | 120.46 | 21.70 | 5.64 | 8786 | 0.809 | 0.596 | - 3.10 | 11.5 |
| 6 | *POMERNIA* | empty | 120.46 | 21.70 | 4.68 | 6786 | 0.718 | 0.555 | - 4.75 | 13.6 |
| 7 | *semi-container* | full | 140.00 | 22.00 | 9.14 | 20767 | 0.855 | 0.718 | 0.73 | 19.0 |
| 8 | ship | ballast I | 140.00 | 22.00 | 3.60 | 7235 | 0.712 | 0.635 | - 6.58 | 14.8 |
| 9 | *WARSZAWA* | ballast II | 140.00 | 22.00 | 5.75 | 12231 | 0.745 | 0.669 | - 1.85 | 12.8 |
| 10 | *dry cargo ship MAESTRO* | full | 134.00 | 21.40 | 8.96 | 19971 | 0.800 | 0.754 | 0.17 | 16.7 |
| 11 | *tanker* | full | 272.00 | 43.40 | 15.20 | 152599 | 0.884 | 0.826 | 4.55 | 12.0 |
| 13 | *GIEWONT II* | heavy ballast | 272.00 | 43.40 | 10.45 | 102540 | 0.851 | 0.807 | - 0.28 | 10.9 |
| 14 | | full | 205.00 | 30.50 | 12.09 | 62450 | 0.871 | 0.826 | 2.64 | 13.7 |
| 15 | *bulk carrier UNIWERSYTET* | ore in 5th hold | 205.00 | 30.50 | 7.12 | 35620 | 0.818 | 0.801 | - 1.60 | 10.4 |
| 16 | | heavy ballast | 205.00 | 30.50 | 8.64 | 43616 | 0.834 | 0.807 | - 0.13 | 11.5 |
| 17 | | grain | 205.00 | 30.50 | 11.33 | 58226 | 0.861 | 0.822 | 1.17 | 16.1 |
| 18 | *Ro-Ro ship* | ballast | 182.00 | 31.00 | 9.30 | 34442 | 0.774 | 0.637 | - 3.30 | 28.8 |
| 19 | *POZNAŃ* | ballast; departure | 182.00 | 31.00 | 6.80 | 24012 | 0.696 | 0.607 | - 3.16 | 12.0 |
| 20 | *fishing trawler B- 11* | loaded | 28.47 | 6.70 | 2.76 | 281 | 0.77 | 0.533 | 0.29 | 5.8 |

*Tab. 2. Calculation results of rolling axis location for the selected ships*

| Ship no. | $z_{Gw}/B$ [-] | $a_w/B$ [-] | $b_w/B$ [-] |
|---|---|---|---|
| 2 | - 0.300 | - 0.042 | - 0.241 |
| 8 | - 0.299 | - 0.042 | - 0.258 |
| 4 | - 0.247 | - 0.010 | - 0.208 |
| 1 | - 0.243 | - 0.002 | - 0.237 |
| 6 | - 0.219 | 0.012 | - 0.191 |
| 3 | - 0.168 | 0.040 | - 0.231 |
| 5 | - 0.143 | 0.049 | - 0.072 |
| 18 | - 0.106 | 0.054 | - 0.257 |
| 19 | - 0.102 | 0.067 | - 0.171 |
| 9 | - 0.084 | 0.087 | - 0.087 |
| 15 | - 0.052 | 0.092 | - 0.036 |
| 13 | - 0.007 | 0.091 | - 0.097 |
| 16 | - 0.004 | 0.118 | - 0.042 |
| 10 | 0.008 | 0.095 | - 0.145 |
| 7 | 0.033 | 0.105 | - 0.122 |
| 17 | 0.038 | 0.121 | - 0.083 |
| 20 | 0.043 | 0.105 | - 0.161 |
| 14 | 0.087 | 0.128 | - 0.169 |
| 11 | 0.105 | 0.140 | - 0.062 |

of rolling axis from waterline, correctly describes the influence of the added masses determined relative to the point O, (Fig.1), on location of rolling axis.

It was also assumed that the relative quantity $a_W/B$ was most tightly connected with the relative distance of the ship mass centre from the same waterline, $z_{Gw}/B$.

The relationship $a_W/B = f(z_{Gw}/B)$ is shown in Fig.2. This diagram reveals good correlation between the selected quantities. A significant part of the irregularities observed in the diagram can be deemed as a result of the accepted simplifying assumptions, as well as inaccuracy in reading-out intermediate values. Though the diagram has been obtained for the simplified ships, it can be considered as that correctly describing the selected relationships also for the real ships. The straight line described as :

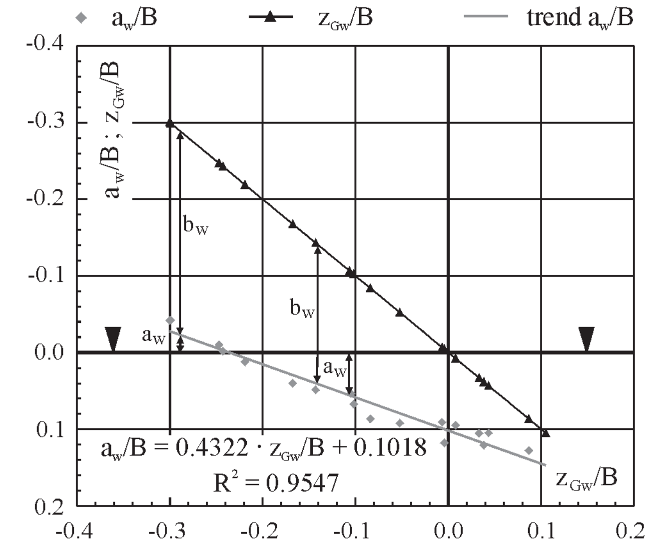

*Fig. 2. Location of ship rolling axis respective to ship waterline*

$$a_W/B = 0.432\,(z_{Gw}/B) + 0.102 \tag{10}$$

fits into the relationship $a_W/B = f(z_{Gw}/B)$ the best.

After a simple analysis of (10) it can be demonstrated that the rolling axis is located approximately 0.1B under the water when the ship mass centre lies on its surface, and that the rolling axis is located on the water surface when the ship mass centre is abt. 0.24B over the water, and finally, that the rolling axis goes through the ship mass centre when it is located abt. 0.18B under the water surface. To make the diagram more clear, also the straight line $z_{Gw}/B = f(z_{Gw}/B)$ was drawn in it.

By taking into account that $z_{Gw} = d - z_G$ the following approximate expressions for the distance of ship rolling axis from ship's waterline and ship mass centre, respectively, can be obtained :

$$a_W \cong 0.43\,(d - z_G) + 0.1B$$

or (11)

$$b_W \cong 0.57\,(d - z_G) - 0.1B$$

This is a simplified form of the expressions (4) in which the hydrodynamic coefficients $m_{yy}$ and $m_{y\Phi}$ functionally depend also on the rolling frequency $\omega$. The, observed in Fig.2, small scatter of the points, i.e.values determined by (4), indicates that rolling axis location weakly depends on other parameters than those represented by the coordinates shown in the diagram in question. In the obtained formulas the dependence on the rolling frequency has place indirectly only because the calculations for all the selected ships have been carried out with taking into account their free-rolling frequencies, hence the diagram of Fig.2 and the formulas (10) and (11) obtained on its basis, concerns just such frequencies.

## FINAL REMARKS AND CONCLUSIONS

- Ship rolling motions are of a great importance in considering the problems of ship stability, strength, cargo lashing, comfort and safety of passengers and crew members; they are also important for correct operation of ship equipment.
- For considering and solving many of the above mentioned problems it is usually sufficient to assume the ship to be an object of one degree of freedom. When such approach is to be used correct determination of location of ship rolling axis is always crucial.
- Due to the water supporting the ship during its motions, which forms constraints of a special type, it is possible to assume the free ship rotation axis to be a straight line passing through – or close to – the ship mass centre. However the total ship mass consists of the mass of the ship itself and the supplementary, „added masses" of water, associated with relevant motions. These conventional masses affect location of ship rolling axis in accordance with places of their occurence.
- If the ship rolling axis is so defined as it has been proposed in this paper the usually assumed location of rolling axis can be improved; the corrections might reach even 0.1÷ 0.2B as it was shown in this paper.
- It is also important that the proposed formulas, (10) and (11), concern resonant rolling, as in real conditions the ship responds to instantaneous excitations with motions of natural frequency, and in irregular waves it „selects" – from an encountered wave spectrum – mainly the frequencies close to natural frequencies of its own motions.
- The proposed simple approximate formulas make it possible to calculate location of ship's rolling axis only on the basis of the typical ship data available at the preliminary design stage.

### NOMENCLATURE

$a_W$ - distance from rolling axis to waterline of a real ship [m]
$a_{WS}$ - distance from rolling axis to waterline of a simplified ship [m]
$b_W$ - distance from rolling axis to mass centre of a real ship [m]
$b_{WS}$ - distance from rolling axis to mass centre of a simplified ship [m]
dof - degress of freedom
D - ship displacement [T or kN]
g - acceleration of gravity [m/s$^2$]
GM - initial metacentric height of a ship, not corrected for free-surface influence [m]
$I_x$ - ship mass moment of inertia relative to X- axis [tm$^2$]
$I_{xx}$ - added mass moment of inertia relative to X- axis [tm$^2$]
L, B, d, H - main dimensions of a ship [m]
m - ship mass [t]
$m'_{yy}$, $m'_{y\Phi}$ - dimensionless coefficients of added mass of water [-]
$m_{yyj}$, $m_{y\Phi j}$ - unitary coefficients of added mass of water [t/m, tm/m]
$m_{yy}$, $m_{y\Phi}$ - coefficients of added mass of water [t, tm]
s - index of a simplified ship
$y_G$, y - swaying translations of ship mass centre G and point 0, respectively [m]
$z_G$ - height of ship mass centre over base plane [m]
$z_{GW}$ - height of ship mass centre over waterplane [m]
$\alpha$ - waterplane coefficient [-]
$\delta$ - hull block coefficient [-]
$\tau$ - period of ship's free rolling [s]
$\Phi$ - heeling angle [rad or deg]
$\omega$ - angular frequency of ship's free rolling [1/s]
$\omega'$ - dimensionless ship rolling frequencies [-]

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**CONTACT WITH THE AUTHOR**
D.Sc. Ludwik Balcer
Faculty of Ocean Engineering
and Ship Technology
Gdańsk University of Technology
80-952 Gdańsk, POLAND
Narutowicza 11/12
e-mail : balcer@pg.gda.pl


## Conference

## E Regional Group E

On 11 December 2003 at Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology, held was the last-in-that-year seminar of the Regional Group of the Section on Exploitation Foundations, Machine Building Committee, Polish Academy of Sciences.

***During the seminar three papers were presented :***

- *A global model of environmental safety of ship power plant* – by R. Liberacki (Gdańsk University of Technology)
- *Research on water-lubricated bearings of ship main shafting* – by W. Litwin (Gdańsk University of Technology)
- *An attempt to diagnosing the main propulsion systems of the ferry ships "Polonia" and "Pomerania" by means of the FAM-C method* – by A. Gębura (Air Force Institute of Technology, Warsaw)

The seminar was a good occasion to offer wishes to Prof. J. Lewitowicz in his 70$^{th}$ birthday and to pass gratulations to him due to his outstanding achievements in the area of exploitation of technical objects, especially in aeronautics.

# On a risk-based method for safety assessment of a ship in critical conditions at the preliminary design stage

## Current report

**Mirosław Gerigk**
Gdańsk University of Technology

ABSTRACT

*The paper presents some current results of the research on development of a method for safety assessment of ships in critical conditions at the preliminary design stage. The method is expected to make it possible to perform the multi-objective and multi-parametric investigations to achieve the required and optimum levels of safety by using the risk-based and formal approaches to ship safety. During the design analysis influence of parameters associated with the hull form, arrangement of internal spaces, loading conditions, position and extent of damage, cargo shift and weather impacts on safety of ships in critical conditions will be taken into account. Current capabilities of the method are illustrated by several calculation examples of influence of different arrangements of internal spaces of a cargo ship on values of the IMO formal subdivision indices such as the "Attained Subdivision Index A" and "Local Subdivision Indices $\Delta A_j$". The method is still under development and its final structure will be presented in the near future.*

**Key words** : ship safety, formal safety assessment, design for safety, ship hydromechanics, survivability

## INTRODUCTION

During the last decade many European institutions conducted the investigations related to the ship safety problems [1, 2, 3, 4, 5]. Recently, the HARDER research programme has been set up to revise the SOLAS Convention, (Chapter II-1 Parts A, B and B-1), it has been aimed at solving the following problems [8] :

- reviewing model tests and methods concerning safety evaluation and assurance including the development of the methods contained in A.265 IMO document [14], and also that applicable to Ro-Ro ships
- survival prediction for various ships in waves including observed mechanisms of capsize
- estimation of probability of sea state occurrence at the time of a casualty
- development of the so called survival factor "s".

Also, at the Chair of Ship Hydromechanics, Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology two projects have been conducted regarding a new method and model for the safety estimation of ship in critical conditions, as well as a model for direct risk assessment of ships in critical conditions [6 , 7]. The major research issues associated with the above mentioned projects were :

- damage stability modelling methods
- large scale flooding
- dynamic effects due to internal (ballast and/or cargo shift) and external (waves, wind) impacts
- development of survival criteria for the ships in damage condition
- investigations on the safety assessment
- and example preliminary designs.

The critical conditions have been defined as the ingress of external water into any group of watertight compartments of a ship, associated with both internal (cargo and/or ballast shift) and external (waves and wind action) impacts.

The knowledge base for the research includes: naval architecture, ship hydromechanics, system approach to safety, safety case/formal safety assessment (FSA) methods, IMO regulations for ship safety.

This paper is devoted to presentation of some current results of the conducted research projects regarding development of a method for safety assessment of ships in critical conditions at the preliminary design stage.

## MAJOR ASSUMPTIONS FOR THE MODERN APPROACH TO SHIP SAFETY IN CRITICAL CONDITIONS

The modern approach to ship safety in critical conditions is based on the general assumption as follows :

**The first assumption**

The factors affecting the ship safety in critical conditions may result from different sources : design, operation and management.

**The second assumption**

The factors affecting the ship safety may come from different groups of factors such as :

- 1st - the external factors not related to the environment :

general human factor, control systems, technical means, legislative actions.
- ♦ 2nd - the environmental factors including the wind, waves and current.
- ♦ 3rd - connected with the ship :
  - ⇨ ship including hull, propeller and rudder
  - ⇨ cargo including arrangement of internal spaces, cargo and ballast distribution and loading condition
  - ⇨ inter-related parameters and characteristics directly connnected with ship safety: intact stability, damage stability, dynamical stability in damage condition, and survivability.
- ♦ 4th - connected with ship operation: the ship management system including the ship safety system.
- ♦ 5th - connected with the shipping company's safety management system and safety culture.
- ♦ 6th - the human factor including both the psychological and physical predispositions, character, morale, integrity, knowledge, experience and training degree.

In the current work the factors of the first four groups have been taken into account in the method for assessment of ship safety in critical conditions.

**The third assumption**

The modern approach to ship safety is connected with combining: the system approach and Formal Safety Assessment (FSA) method which contains : hazard identification, risk assessment and risk reduction.

By taking into account these elements the method in question has gradually been developed to include the following components (shown also in Fig.1) :

- design requirements, criteria and constraints
- risk acceptance criteria
- safety objectives
- ship and environment definition
- design analysis :
  - ⇨ hazard identification
  - ⇨ hazard assessment
  - ⇨ scenario development
  - ⇨ hydromechanical design analysis (intact stability, damage stability, dynamical stability in damaged condition)
- risk assessment
- risk acceptance criteria
- risk reduction (mitigation measures)
- modification of design
- ship safety assessment
- decisions on ships safety.

For estimating the risks the following techniques have been selected to be used within the method :

- ⇨ ALARP (As Low As Reasonable Possible) concept
- ⇨ F-N curve : the frequency of fatalities in function of number of fatalities
- ⇨ Risk acceptance matrix.

For the hazard identification the hazard and operability (HAZOP) studies have been used as the most effective technique. The event tree analysis (ETA) has been used for the scenario identification.

The statistics and investigations into serious casualties, documented in [9], have been assumed the major source of information on hazards and risks involved in shipping. It has been further assumed that the design method should combine both global approach and technical approach [13].

The global approach has concerned the method framework containing the elements just above enumerated. The technical approach has been connected with the developing of :

- ✧ logical structure of design system
- ✧ design requirements, criteria and constraints
- ✧ logical structure of computational model
- ✧ analytical and numerical methods
- ✧ application methods regarding the intact stability, damage stability, dynamical stability in critical and damage condition.

For risk management considerations the top-down risk management method has been applied which is suitable for design for safety at the preliminary design stage.

## METHOD OF SHIP DESIGN FOR SAFETY IN CRITICAL CONDITIONS

A worked-out safety estimation method for ships in critical conditions is associated with solving a few problems regarding the naval architecture, ship hydromechanics and ships safety, and it is novel to some extent.

The logical structure of the method and design system combining the global approach and technical approach is presented in Fig.1.

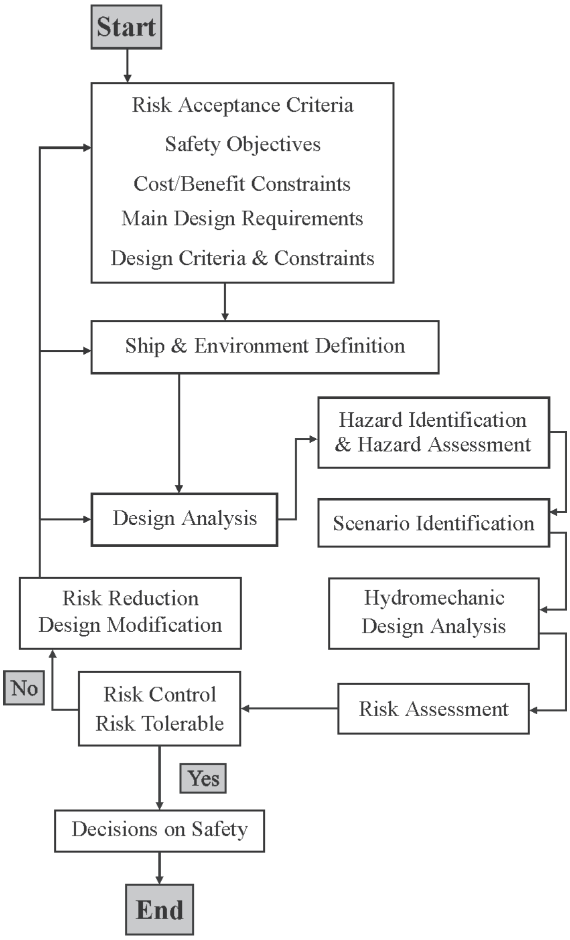



***Fig.1.*** *Logical structure of the design system and computational model combining the global approach and technical approach*

According to Fig.1, the ship and environment are defined as objects of hydromechanics, described by a set of parameters. The "Hydromechanic Design Analysis" module contains the design methods in which both the functions and procedures associated with solving particular ship hydromechanics problems are used. The "Risk Assessment" module includes the methods which combine both the "hydromechanics" and "risk assessment" functions and procedures. The "Main Design Requirements" module is intended to be consisted of :

- ❖ general requirements
- ❖ IMO regulations
- ❖ requirements of classification societies and
- ❖ requirements of conventions (SOLAS`90 and SOLAS`95.

The current set of requirements included into the database has so far concerned the IMO regulations only. The database will eventually include both the risk acceptance criteria, safety objectives, main requirements and design criteria and constraints.

By using the method
two separate design processes can be initiated:

- ♦ iterative approach
- ♦ parametric investigations.

From the design process point
of view the method consists of two sub-methods [10]:

- ★ parametric method - when intact stability and damage stability characteristics are calculated
- ★ semi-probabilistic or probabilistic method - for solving the problems related to the survivability and risk assessment.

The following design options of the computational model, based on the IMO regulations [15], are currently offered :

- ✸ calculation of the "Attained Subdivision Index A"
- ✸ calculation of the "Local Subdivision Indices $\Delta A_j$"
- ✸ calculation of the "Probability of Oil Out-flow"

Moreover it is possible to use the option : calculation of the "Probability of Capsizing in Critical Conditions", not covered by the IMO regulations, i.e. the direct risk assessment method which is still under development.

At the current stage of development of the method
it has been assumed that the method :

- ✦ has scientific and research features rather than purely design one
- ✦ is useful for design of the conventional cargo vessels only.

The method is still under development
and its final structure will be presented in the near future.

## COMPUTATIONAL MODEL OF SHIP DESIGN FOR SAFETY IN CRITICAL CONDITIONS

The most important features
of the computational model in question are as follows :

- ⅄ it is open
- ⅄ it has a hybrid-modular structure
- ⅄ it has a common library of analytical and numerical methods
- ⅄ it has a common library of application methods.

From the practical point of view the computational model is based on a dynamical database concept, and it is original to some extent. The database is modular and related to the logical structure of the computational model. The database enables to provide the safety estimation when ship's hydromechanical characteristics are estimated by using : the numerical calculations (direct methods), model tests results, results from the full scale trials, combined empirical and numerical calculations (semi--direct methods) or empirical calculations (indirect methods).

The computational model has been prepared for the PC Windows–based workstations. The application software is programmed in Fortran, Pascal and C++ languages. The detail structure of the data base including the input data, outputs, graphics and ship design data will be presented in the near future.

At the current stage of work it is possible to use the method and computational model for the research purposes only. A version for implementation for the research and design purposes would be available soon.

## SOME EXAMPLES OF CURRENT CAPABILITIES OF THE METHOD IN QUESTION

Below presented are a few examples of using the method for calculation the probability of flooding any group of compartments, optimisation of the "Attained Subdivision Index A" and calculation of the "Local Subdivision Indices $\Delta A_j$" in compliance with IMO relevant regulations.

Example 1

***Calculation of the probability of flooding any group of compartments***

At the design stage the estimation of the probability of survival of flooding any group of compartments is connected with calculation of the "Attained Subdivision Index A" [15]. The risk assessment is associated with satisfying the criterion :

$$A \geq R$$

where :

$$A = \Sigma\, p_i\, s_i$$

Both the indices, A and R, are calculated according to the formula accepted by IMO. In the presented example the formula given by the Resolution MSC 19/58 – "Subdivision and damage stability of cargo ships over 100 m" has been applied.

A typical set of ship initial data
for the given ship may look as follows :

- main parameters
- stability data, hull form
- arrangement of internal spaces (like the presented in Fig.2)
- loading condition
- set of permeabilities.

**Arrangement of internal spaces :**

- Transverse Subdivision Considered
- Longitudinal Subdivision Considered
- Horizontal Subdivision Considered

**Main Parameters :**

$L_S$ = 174.95
$L_{bp}$ = 163.00
B = 26.50
H = 14.20
d = 9.00

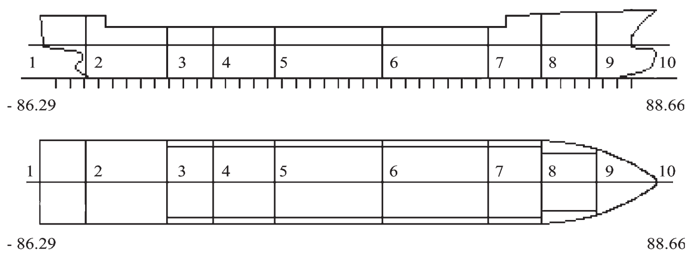



***Fig.2.*** *An example arrangement of internal spaces of a containership*

According to the performed calculations value of the "Attained Subdivision Index A" is as follows [16] :

$$A = \Sigma\Delta A_{pi} = 0.5501$$

The Required Subdivision Index is:

$$R = (0.002 + 0.0009 \cdot L_S)^{1/3} =$$

$$= (0.002 + 0.0009 \cdot 174.98)^{1/3} = 0.54230$$

where :

$$L_S = 174.98 \text{ [m]}$$

The final result satisfies the relation A > R, as :

$$0.5501 > 0.5423.$$

Example 2

***Optimisation of the "Attained Subdivision Index A"***

For more advanced procedures of design for safety of a ship in damage condition the following approach can be applied. In Fig.3 the arrangement of internal spaces of a cargo ship is presented. During the computer simulation three design versions regarding the number and position of transverse bulkheads were taken into account. Tab.1 presents the results of optimisation of the "Attained Subdivision Index A" for all the design versions (Fig.3). During the optimisation process the preliminary positions of some bulkheads were significantly changed.

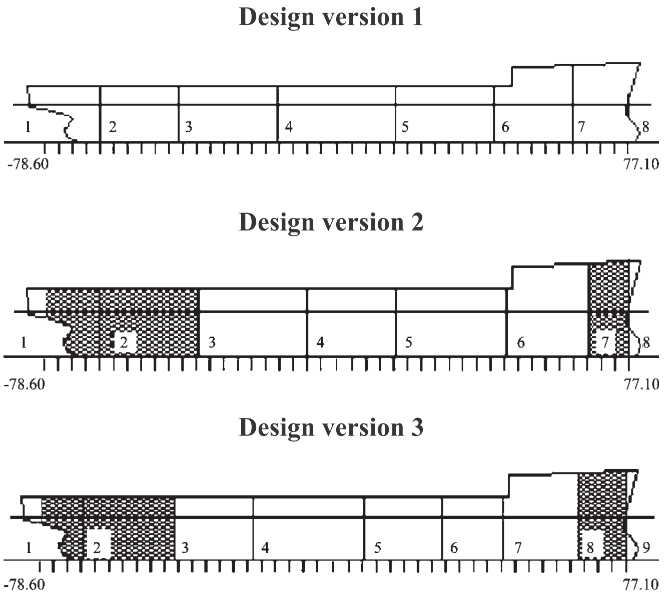



*Fig. 3. Three different preliminary arrangements of internal spaces of the considered cargo ship*

The optimisation method consited in maximization of the objective function represented by the "Attained Subdivision Index A". Its aim concerned reaching possible maximum values of the $p_i \cdot s_i$ factors for each group of the watertight compartments. One of the constraints was that the $s_i$ value should never be equal to null. The major starting design conditions were : number and positions of bulkheads.

The investigations showed that a larger number of bulkheads not necessarily guarantee much higher values of the index "A". Moreover, by applying one bulkhead more the $p_i$ values become greater for the groups including two or more single compartments, but in the same time the conditional probabilities $s_i$ decrease. The values of $p_i$ and $s_i$ factors are interrelated but in an irregular manner. In general, one bulkhead more or less may only slightly affect the "Attained Subdivision Index A" values. The small changes in the positions of bulkheads do not give much different values of the index "A", but the local subdivision indices may be changed significantly.

***Tab. 1.** Results of optimisation of the "Attained Subdivision Index A"*

| Ship type: general cargo<br>Design criterion: optimisation<br>of the "Attained Subdivision Index A" | | |
|---|---|---|
| | **No. of iteration** | **Index A value [-]** |
| **Design version 1** | I | 0.6560 |
| | II | 0.7181 |
| | III | 0.8162 |
| | IV | 0.8171 |
| | V | 0.8174 |
| | VI | 0.8177 |
| | **No. of iteration** | **Index A value [-]** |
| **Design version 2** | I | 0.6050 |
| | II | 0.7261 |
| | III | 0.7951 |
| | IV | 0.7974 |
| | V | 0.7976 |
| | **No. of iteration** | **Index A value [-]** |
| **Design version 3** | I | 0.6690 |
| | II | 0.8210 |
| | III | 0.8225 |
| | IV | 0.8278 |
| | V | 0.8233 |

Example 3

***Calculation of the „Local Subdivision Indices $\Delta A_j$"***

The calculation of the local subdivision indices is a kind of optimisation procedure which may provide the same level of safety for each of the considered watertight compartment. The local subdivision indices were calculated according to the following formula [11]:

$$\Delta A_j = (\Sigma\, p_i\, s_i) / (\Sigma\, p_i)$$

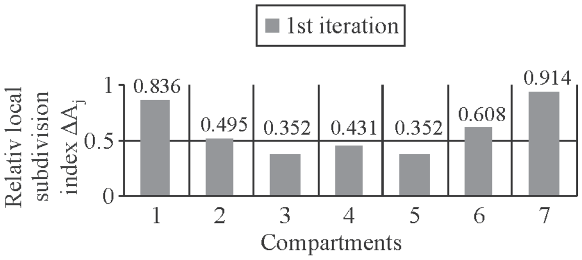
The optimisation process of the indices $\Delta A_j$ is connected with the iterative shifting of the bulkheads against each other until the $\Delta A_j$ values become more or less equal, possibly for all the compartments. In Tab.2 the local subdivision indices $\Delta A_j$ are presented for four performed design iterations. The calculations concerned the design version "1" of the general cargo ship presented in Fig. 3.

| | **Comp. no. 1** | **Comp. no. 2** | **Comp. no. 3** | **Comp. no. 4** | **Comp. no. 5** | **Comp. no. 6** | **Comp. no. 7** |
|---|---|---|---|---|---|---|---|
| $x_1$ [m] | -78.60 | -60.00 | -40.00 | -15.00 | 15.00 | 40.00 | 60.00 |
| $x_2$ [m] | -60.00 | -40.00 | -15.00 | 15.00 | 40.00 | 60.00 | 77.10 |
| $\Delta A_j$ [-] | **0.8363** | **0.4953** | **0.3522** | **0.4311** | **0.3521** | **0.6080** | **0.9140** |
| $x_1$ [m] | -78.60 | -60.00 | -40.00 | -15.00 | 15.00 | 38.40 | 60.00 |
| $x_2$ [m] | -60.00 | -40.00 | -15.00 | 15.00 | 38.40 | 60.00 | 77.10 |
| $\Delta A_j$ [-] | **0.8363** | **0.4953** | **0.3539** | **0.6239** | **0.5478** | **0.6270** | **0.9286** |
| $x_1$ [m] | -78.60 | -60.00 | -40.00 | -16.69 | 15.00 | 38.40 | 60.00 |
| $x_2$ [m] | -60.00 | -40.00 | -16.69 | 15.00 | 38.40 | 60.00 | 77.10 |
| $\Delta A_j$ [-] | **0.8979** | **0.8137** | **0.6104** | **0.7457** | **0.6724** | **0.6453** | **0.9286** |
| $x_1$ [m] | -78.60 | -60.00 | -40.00 | -17.14 | 15.00 | 38.40 | 60.00 |
| $x_2$ [m] | -60.00 | -40.00 | -17.14 | 15.00 | 38.40 | 60.00 | 77.10 |
| $\Delta A_j$ [-] | **0.8979** | **0.9096** | **0.6228** | **0.7415** | **0.6726** | **0.6453** | **0.9286** |

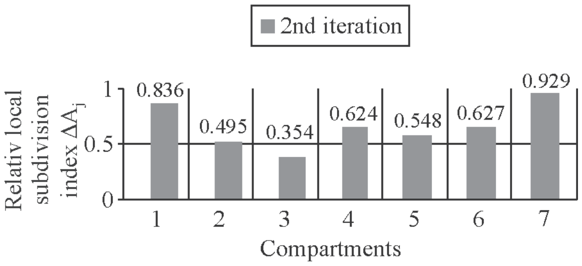
***Tab.2.** Results of the optimisation of the local subdivision indices $\Delta A_j$ for the design version "A" is presented in Fig. 3*

In Fig. 4 the graphical results
of the computer simulation in question are presented.

**Local subdivision indices**




**Local subdivision indices**




**Local subdivision indices**

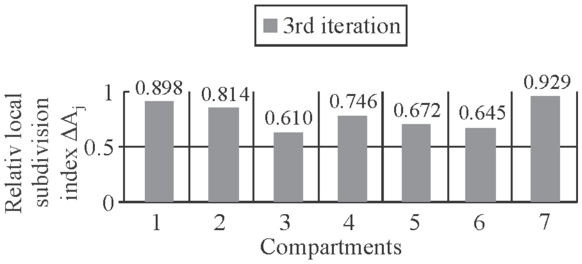



***Fig. 4.*** *Results of the optimisation of the local subdivision indices $\Delta A_j$ for the design version "1" of the general cargo ship, presented in Fig. 3*

## RESEARCH PROJECTS UNDER WAY

One of the projects concerns estimation of the probability of capsizing a ship in critical conditions by using the direct risk assessment method applied to stability problems.

Another deals with consideration of a few novel safety assessment approaches with a view of implementing them into a safety system for ships in critical conditions to support the application of formal safety assessment. The following novel safety assessment and decision-aiding tools applicable to safety management system are considered [14]:

- ✰ expert judgements based on the Taguchi method
- ✰ a safety based decision support system using either knowledge-based expert systems or artificial neural network techniques
- ✰ a fuzzy-logic-based synthesis incorporating the Dempster Shafer approach for making multiple-attribute decision
- ✰ an integration of approximate reasoning approach and evidential reasoning method in design evaluation.

## SUMMARY

Some results of the research projects on an integrated safety estimation method and model for assessing the safety of a ship in critical conditions has been reported. The method is directed towards the ship safety estimation in critical conditions at the preliminary stage of design. By using the method/model the hazard and risk assessment may be performed according to the IMO regulations for cargo ships.

So far, the method and model have been used for investigating the ship safety from the damage stability and survivability point of view. The method/model can be used for guiding ship subdivision for safety. Example results of the investigations of different arrangements of internal spaces of a cargo ship including either transverse or combined subdivision, have been presented. The damage stability, survivability and risk assessment calculations were carried out for each case.

The method is still under development
and its final structure will be published soon.

## Acknowledgement

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**NOMENCLATURE**

| | |
|---|---|
| A | - attained subdivision index |
| $L_S$ | - subdivision length |
| $p_i$ | - probability of flooding the group of compartments under consideration |
| R | - required subdivision index |
| $s_i$ | - probability of survival after flooding the group of compartments under consideration |
| $\Delta A_j$ | - local subdivision index related to each group of compartments under consideration |
| $\Delta A_{pi}$ | - components of the "Attained Subdivision Index A" for each group *i* of compartments |
| $x_1$ , $x_2$ | - positions of bulkheads aft and fore, respectively |

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**CONTACT WITH THE AUTHOR**
D.Sc. Mirosław Gerigk
Faculty of Ocean Engineering
and Ship Technology
Gdańsk University of Technology
Narutowicza 11/12
80-952 Gdańsk, POLAND
e-mail : mger@pg.gda.pl


## Conference

# 24th Symposium of Ship Power Plants

On 27 and 28 November 2003 in Świnoujście, a Polish port town on the Baltic Sea coast , was held 24th international meeting of the scientific workers engaged in the designing, testing and operating of ship power plant devices. The meeting was organized by the Institute of Technical Operation of Ship Power Plants, Maritime University of Szczecin.

**The Symposium's agenda contained presentation of 44 papers split into 5 topical groups :**

- Design and operation of ship power plants (18 papers)
- Renovation of ship power plants (2 papers)
- Failures, tribology and diagnostics (10 papers)
- Control of working processes and computer simulation (9 papers)
- Solid/liquid separation (5 papers).

The authors of the papers represented 7 Polish coastal scientific research centres and 3 Russian ones. Maritime University of Szczecin, Gdańsk University of Technology and Technical University of Szczecin most contributed to preparation of the conference materials. The participants had the opportunity of being acquainted with current production achievements of such companies as : MAN-B&W, Caterpillar, Alfa Laval, Bosch Rexroth, Volvo Penta and Olympus, and of visiting floating objects of Polish Navy Base in Świnoujście.

## Miscellanea

# Agreement on cooperation

In July 2003 the agreement on scientific technical cooperation in the area of ship electric engineering between Polish Register of Shipping (PRS) and the Department of Ship Electric Engineering, Gdynia Maritime University, was signed by Mr. J. Jankowski, D.Sc., General Director of PRS, and Prof. J. Mindykowski, Head of the Department.

Realization of the cooperation is intended on one hand to support developments in the field of ship electric power engineering, and on the other hand to help in developing and modernizing rules for construction and classification of ships and floating objects for sea and inland navigation.

**The common activity to be undertaken will concern** :

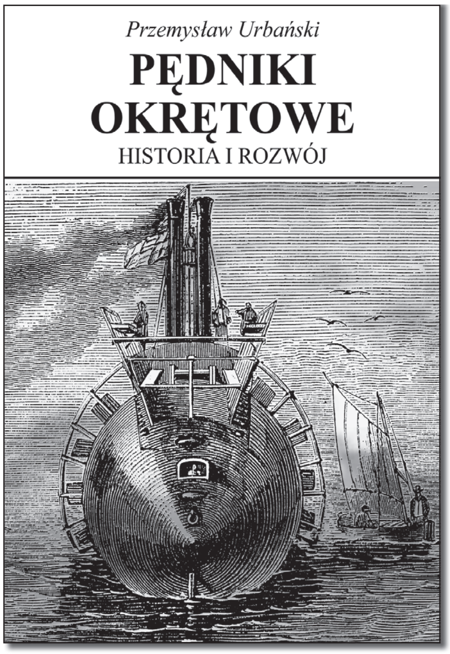
- production and distribution of electric energy
- quality parameters of the energy
- influence of electric energy consumers on its quality
- influence of quality of electric energy on operation of its consumers
- instrumentation for measurement and control of electric energy quality.

**Within the scope the cooperation will be realized by means of** :

- reviewing topics proposed by each of the parties and common choice of priorities
- common carrying out measurements of electric energy parameters on selected objects
- determining research directions on the basis of earlier obtained results
- common analyzing research results from the point of view of their possible application into PRS rules and instructions
- analyzing possible improvements of ship electric plant control & measurement systems, resulting from research work
- elaboration of a portable electric power analyzer applicable to the new and existing ships not equipped with a modified permanent measurement & control system of electric power plant.

Finally common publishing activity in the range of the cooperation topics has been agreed.

# Marine propulsors - history and development




The chronologically presented review of various technical concepts and events enables to properly perceive genesis and ways of development of the contemporary propulsors as well as the role played by the gradually collected experience as well as by the development of engineering sciences.

The entire scope of the book is contained in the eight chapters :

I . Punt-poles, Oars and Paddles
II. Sails and Other Wind Propulsion Devices
III. Paddle Wheels
IV. Screw Propellers :
- The origin, Design Types and Development of Screw Propellers Before 1836
- F.P. Smith's and J. Ericsson's Propellers. The Begining of the Era of the Screw-propulsion
- The Screw-propulsion on the Sailing Ships
- The Evolution of Screw Propellers after 1836

V. Hydraulic Jet Propulsion
VI. Vertical Axis Propellers – Cycloidal Propellers
VII. The Idea of Nature – Inspired Propulsion Devices
VIII. Past and Present Unconventional Propulsion Devices

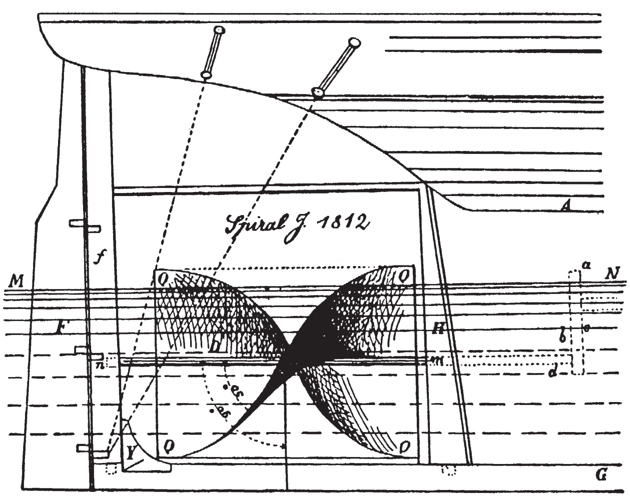
In the year 2001 the Shipbuilding & Shipping publishing house in Gdańsk, edited (in Polish) the book on „*Marine Propulsors - their history and evolution*" written by Przemysław Urbański, retired Assist. Prof., D.Sc., M.E. The book resulted from the research undertaken by the author in co-operation with Hydrodynamics Department, Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology, and Polish Nautological Association in the years 1998 and 1999. In the this way elaborated monography the author has presented the entire image of history and development of marine propulsors, understanding under this notion any device applicable to direct propelling various floating objects beginning from the most primitive ones like rafts and boats to the most specialized e.g. hovercrafts or racing motor boats. Due to its popular-science character the book is dedicated to a wide spectrum of readers professionally or emotionally engaged in shipbuilding, yachting and water sports.

In the largest, 4th Chapter devoted to screw propellers, the geanalogical tree originally elaborated by the author, which illustrates paths of development of propulsors of the kind, is attention deserving. Roots of the tree go back to inventions of Archimedes, Leonardo da Vinci and a Chinese toy of unknown date. The author's concept consisting in presentation in one book all types of propulsors in their historical development, should be deemed equally original.

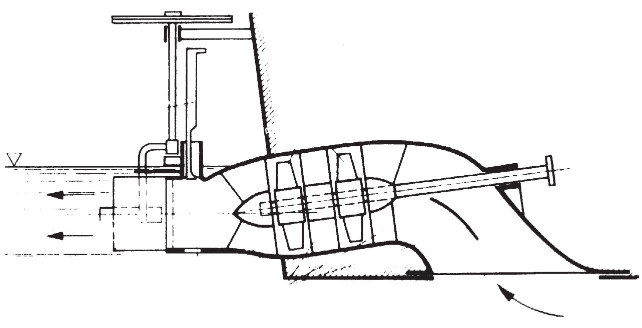
The book is illustrated with a great quantity of figures (303 drawings and fotos) and edited in the form of semi-album.The text of the book is supplemented by :

⇒ Bibliography (78 items)
⇒ List of Illustrations
⇒ Index of Subjects
⇒ Index of Proper Names
⇒ Index of Watercraft's Names.

The presented book was preceded by another interesting book of the same author, titled : „*Two centuries of mechanical propulsion of ships*", edited (in Polish) by Marpress publishing house in 1997, and reviewed in Polish Maritime Research No 2/1997.




*Ressel`s screw propeller (1812)*



*Dowty-Hamilton water-jet propulsive system (1963)*

# Influence of gas turbine controller adjustment on ship propulsion system behavior in rough sea conditions

## Part 2. The simulation investigations

**Marek Dzida**
Gdańsk University of Technology

ABSTRACT

*The paper presents simulation investigations of influence of gas turbine controller adjustment on ship propulsion system operating in heavy sea conditions, based on the model presented in Part I of the paper. The ship propulsion system with two gas turbines driving – through mechanical gear – ship controllable pitch propeller, described in Part I, was used to analysis. The harmonic disturbances due to sea waves with selected frequencies were used as input function in computing the system's amplitude characteristics at different settings of P and PD controllers.*

**Key words** : ship propulsion systems, gas turbines, automatic control, sea waving

## INTRODUCTION

The designing process of the control system of the gas turbine used as ship's prime mover is governed by the general principles of ship power plant designing. However the gas turbine control system has its specific features. These are : transient processes of gas temperature before gas turbine, their influence on gas turbine power and efficiency, and on its mechanical and thermal stresses. In order to limit maximum temperature values it is necessary to use some limitations which influence transient processes in the entire control system. The ship propulsion system with gas turbines driving controllable pitch propeller is affected not only by the excitations due to settings of propeller shaft's angular speed, and of propeller pitch, but also a strong influence on operation of the ship propulsion system is exerted by sea waves (sea state). The description of the problem, the elaborated simulation model together with made assumptions, was presented in Part I of the paper. In this part (Part II) the simulation investigations, carried out by means of the model, and their results are presented and thoroughly discussed.

## STRUCTURE OF THE INVESTIGATED SHIP PROPULSION SYSTEM WITH GAS TURBINES

The ship propulsion system with two gas turbines driving – – through toothed transmission gear – ship controllable pitch propeller (see Fig.1, Part I) was used to analysis.

In the propulsion system the propeller shaft's angular speed controller and the propeller pitch ratio controller was used. For each of the engines the separate control loop was applied. In the speed controller two additional limiters were applied : that of combustion chamber outlet temperature, which is actuated when a given threshold temperature is exceeded, and that of maximum fuel flow to combustion chamber. The controller structure was presented in Part I of the paper.

The gas turbine in the propulsion system in question is a two-shaft turbine with separated power turbine, operating in simple open cycle. The parameters of the analyzed ship propulsion system are given in Tab.1 [1].

***Tab.1.*** *Parameters of the ship propulsion system in question*

| **Engines** | Number 2 | | |
|---|---|---|---|
| Rated power of ship propulsion system | $(\mathbf{N_E})_r$ | kW | 462.4 |
| Rated angular and rotating speed of power turbine (PT) shaft | $(\omega_{PT})_r$<br>$(\mathbf{n_{PT}})_r$ | rad/s<br>rpm | 2513<br>24000 |
| Rated torque of power turbine | $(\mathbf{M_{PT}})_r$ | Nm | 92 |
| Rated power of power turbine | $(\mathbf{N_{PT}})_r$ | kW | 231.2 |
| **Propeller** | Number 1 | | |
| Reduction gear ratio | **g** | $\omega_{PT}/\omega_P$ | 60.00 |
| Wake fraction | **w** | - | 0.218 |
| Propeller diameter | **D** | m | 1.5 |
| Rated propeller pitch ratio | **H/D** | - | 0.7 |
| Rated torque of propeller | $(\mathbf{M_P})_r$ | Nm | 10818 |
| Rated angular and rotating speed of propeller shaft | $(\omega_P)_r$<br>$(\mathbf{n_P})_r$ | rad/s<br>rps | 41.89<br>6.667 |
| Rated power of propeller | $(\mathbf{N_P})_r$ | kW | 453.2 |

For the simulation investigations used was the proportional controller (P) or – alternatively – the proportional-derivative controller (PD), having the following adjustment coefficients:

$k_P$ - gain (amplification) coefficient of controller (for P and PD controllers)
$T_D$ - differential time constant (for PD controller)

The harmonic disturbances due to sea waves were applied to determine – for assumed disturbance frequencies – the amplitude characteristics at different settings of the P and PD controllers.

## HARMONIC DISTURBANCES DUE TO SEA WAVES AFFECTING SHIP PROPULSION SYSTEM

The disturbances from the side of sea waves lead to disturbances of resistance torque of rotary system elements. The loads were determined as the quantities of the determinate amplitudes dependent on the disturbance amplitude [2].

Such disturbance was introduced to the system as a change of the propeller advance speed $V_p$ . The speed $V_p$ is calculated in accordance with the relevant formula given in Part I, in which the speed change due to sea waves has been accounted for.

The mean water speed of advance onto propeller, $c_m$ , with accounted for sea wave action, can be determined from the relevant relationship given in Part I. From the continuous sea wave spectrum only some its definite frequencies $\omega_m$ were selected.

In Tab.2 given are the main wave parameters in different sea states, on the basis of which values of $c_m$ and $\omega_m$ have been determined.

In order to investigate the influence of sea conditions on the operation of the ship propulsion system with gas turbines

***Tab. 2.*** *Main wave parameters in different sea states [2]*

| **Wind velocity** | [°B] | 5 | 6 | 7 | 8-9 | 10 | 11 |
|---|---|---|---|---|---|---|---|
| **Sea state** | [°B] | 3 | 4 | 5 | 6 | 7 | 8 |
| **Mean wave period** | $T_w^m$ [s] | 3.7 | 4.5 | 6.0 | 7.6 | 8.7 | 9.5 |
| **Wave height** | H [m] | 1.25 | 2.0 | 3.5 | 6.0 | 8.5 | 11.0 |
| **Wave length** | L [m] | 21.4 | 31.6 | 56.2 | 90.2 | 118 | 141 |
| **Wave velocity** | c [m/s] | 5.8 | 7.0 | 9.4 | 11.9 | 13.6 | 14.8 |
| **Water speed on wave surface** | $c_o$ [m/s] | 1.06 | 1.40 | 1.83 | 2.48 | 3.06 | 3.64 |

the amplitude characteristics of the system for a harmonic disturbance from the side of the propeller (sea wave influence) were determined. In the calculations the mean ship's speed $V_o$ was assumed constant, and the characteristics were determined for two values of the mean advance speed of water flow onto propeller, affected by sea waves :

- the speed amplitude $c_m$ which corresponds to 3°B sea state ($H = 1.25$ m ; $L = 21.4$ m) and $h_p = 0.9$ m
- the speed amplitude $c_m$ which corresponds to 8°B sea state ($H = 11$ m ; $L = 141$ m) and $h_p = 0.9$ m

In both cases the characteristics were determined in function of the disturbance frequency $\omega_m$ (wave period $T_w$).

## INVESTIGATIONS OF CONTROLLER ADJUSTMENT INFLUENCE ON SHIP PROPULSION SYSTEM OPERATION IN HEAVY SEAS

The simulation model (Fig.7) elaborated by means of the MATLAB (and SIMULINK package), presented in Part I and [1], was used for the investigations in question.

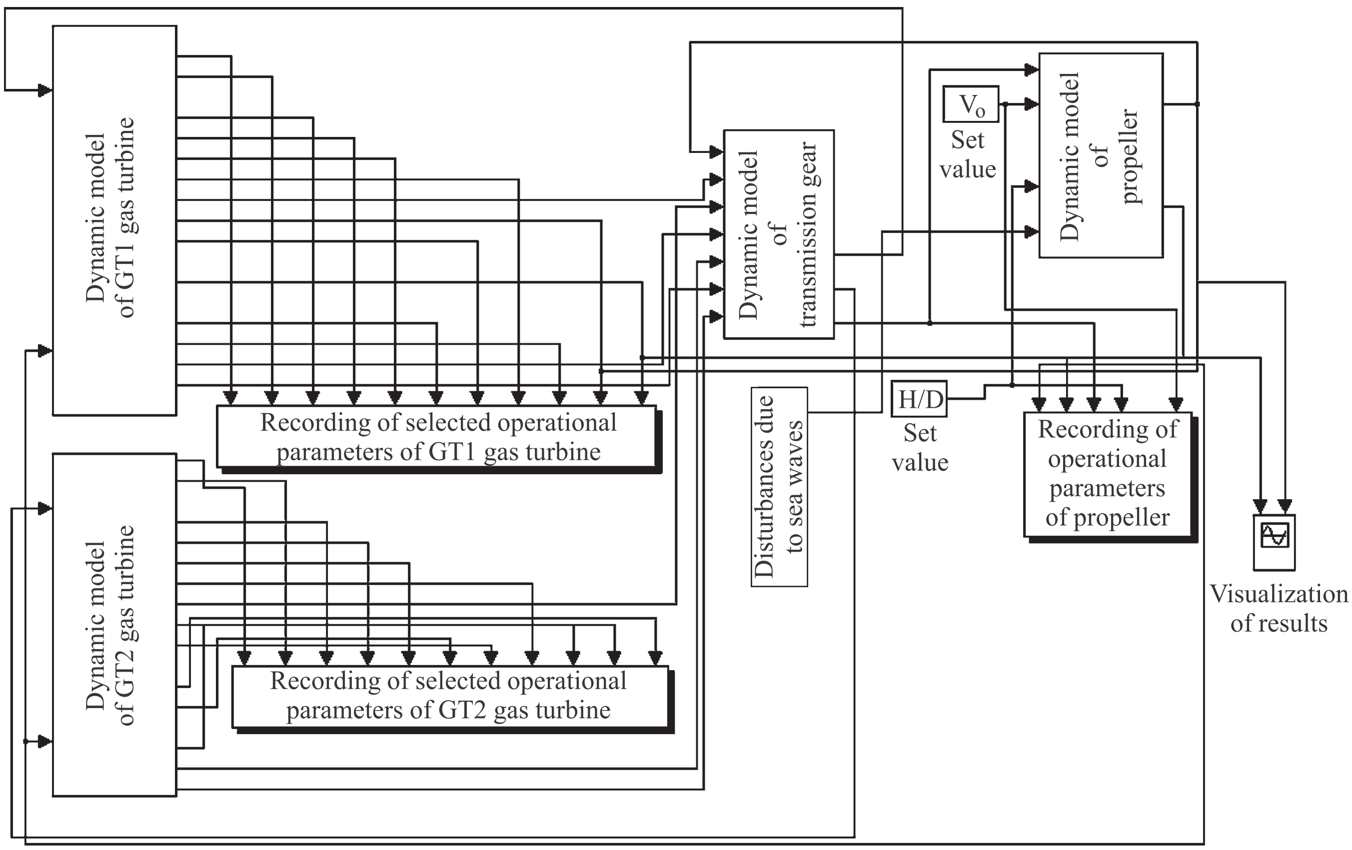



***Fig. 7.*** *Simulation model of ship propulsion system with gas turbines and controllable pitch propeller*

The amplitude characteristics of the propulsion system were determined for two types of gas turbine speed controllers, P and PD, controlling the propeller angular speed. The characteristics were calculated for different controller settings and different disturbance amplitudes corresponding to the sea wave amplitude at 3°B and 8°B sea state. The calculations were performed for the constant value of the propeller pitch ratio :

H/D = 0.7 and the constant linear speed of ship : $V_o$ = idem.

The increments of any variable of the propulsion system, X, were defined as follows :

$$\Delta X = \frac{X - X_o}{X_o}$$

where :

$X_o$ - initial value of a parameter of propulsion system

X - instantaneous value of the parameter

## The ship propulsion system's amplitude characteristics accounting for disturbances due to sea waves

In the calculations of the system's amplitude characteristics in question, the following settings were assumed :

for the proportional controller (P): $k_P$ = 5; 10; 15; (20)

for the proportional-derivative controller (PD) :
$k_P$ = 5 $T_D$ = 0.1 s ; 1 s ; 3 s.

The disturbance period $T_w$ was assumed in the range from 0.1 s to 30 s. The disturbance amplitudes (i.e. the mean propeller speed of advance $c_m$) were chosen as follows :

$c_m$ = 0.745 m/s - corresponding to the wave of

L = 21.4 m, and H = 1.25 m, (3°B sea state)

and :

$c_m$ = 3.450 m/s - corresponding to the wave of

L = 141 m, and H = 11 m, (8°B sea state).

In Fig.8 presented are the gas turbine power amplitude characteristics for the disturbance due to sea waves corresponding to 3°B sea state, at three settings of the P controller of the propulsion system.

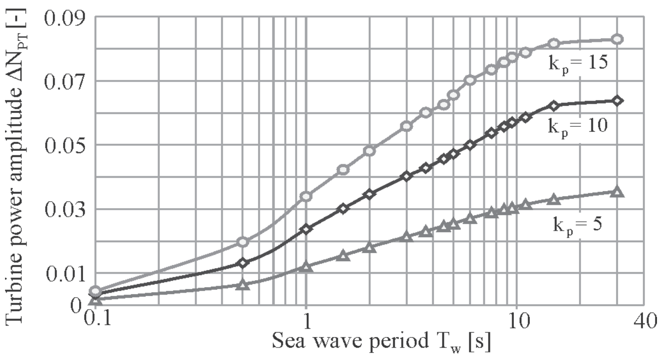



*Fig. 8. The gas turbine power amplitude characteristics for 3°B sea state conditions, at different settings of the P controller with limiters*

With increasing the mean sea wave period $T_w$ and the controller amplification coefficient $k_P$ the gas turbine power amplitudes $\Delta N_{PT}$ also increase. In Fig.9 presented are the relative amplitude characteristics obtained by relating the power amplitude increment $\Delta N_{PT}$ to the disturbance amplitude $c_m$ , for the same settings of the P controller and two disturbance amplitudes due to sea waves. From the presented diagrams it appears that the courses of the relative power pulsations are close to each other, hence in further considerations only the frequency characteristics for the disturbance amplitude due to sea waves corresponding to 8°B sea state ($c_m$ = 3.450 m/s) are taken into account.

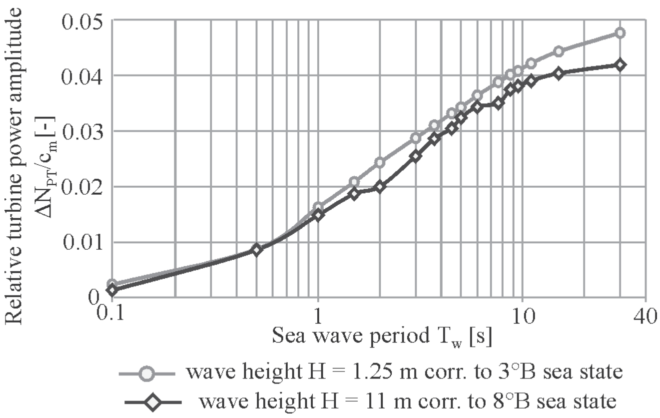



***Fig. 9.** The relative amplitude characteristics of gas turbine power, $\Delta N_{PT}/c_m$ , in function of the sea wave period $T_w$ , at the constant setting $k_P$ = 5 of the P controller with limiters*

In Fig.10 the ship propulsion system amplitude characteristics accounting for wave-induced disturbances, are presented for typical parameters of the propulsion system. Fig.10a shows the amplitude characteristics for the P controller, and Fig.10b – for the PD controller. In both controllers the limiters described in Part I (i.e. the maximum fuel flow limiter and the combustion chamber outlet temperature limiter) were applied.

***Results for the ship propulsion system with P controller (Fig.10a)***

For all quantities characterizing the propulsion system the increase of the controller's amplification coefficient $k_P$ causes the increase of amplitude of the relevant quantity. The amplitudes of the gas turbine power $N_{PT}$ and the fuel flow $m_f$ increase along with increasing the sea wave period $T_w$. The maximum amplitudes of the combustion chamber outlet gas temperature $\theta_3$ move towards higher values of the mean wave periods along with increasing the controller's amplification coefficient $k_P$; e.g. for the amplification coefficient $k_P$ = 5 the maximum amplitude occurs at $T_w$ = 2.5 s, whereas for $k_P$ = 15 it appears at the wave period $T_w$ = 5 s.

***Results for the ship propulsion system with PD controller (Fig.10b)***

At the constant value of the controller amplification coefficient $k_P$ the amplitudes of each of the power propulsion system quantities increase along with increasing the differential time constant $T_D$. For all the analyzed system parameters, except the fuel flow $m_f$ and the propeller speed $n_p$, their maximum amplitudes occur within the interval between the minimum and maximum period of the considered sea wave. At the values of the controller differential time constant $T_D$ equal to 1s and 3 s, the fuel flow characteristics decrease along with increasing the sea wave period $T_w$. At $T_D$ = 0,1s the transient characteristics are almost identical with the transient signals of the P controller having the same value of the amplification coefficient $k_P$. For the propeller speed $n_P$ the amplitude characteristics differ only a little both in function of the sea wave period and of the controller's differential time constant settings.

## The influence of controller limiters on the ship propulsion system's operation in heavy seas

Fig.11 shows the transient signals of the selected quantities of the considered propulsion system in the case of the use of the P and PD controllers both with and without limiters during operation of the system in heavy sea conditions (8°B sea state).

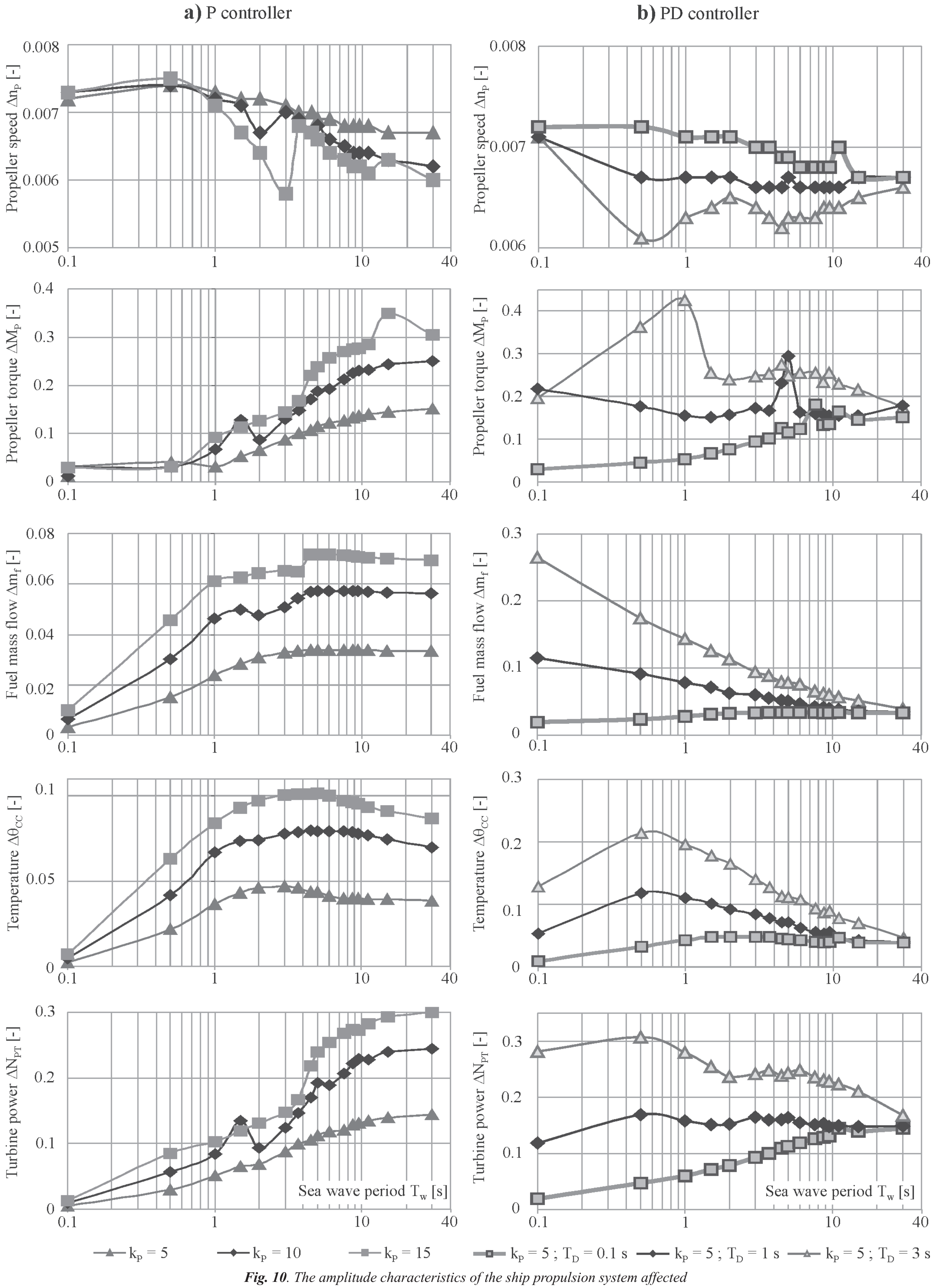



***Fig. 10****. The amplitude characteristics of the ship propulsion system affected by sea wave disturbances of the amplitude corresponding to 3°B sea state;*
***a)*** *for the P controller with limiters* ***b)*** *for the PD controller with limiters*

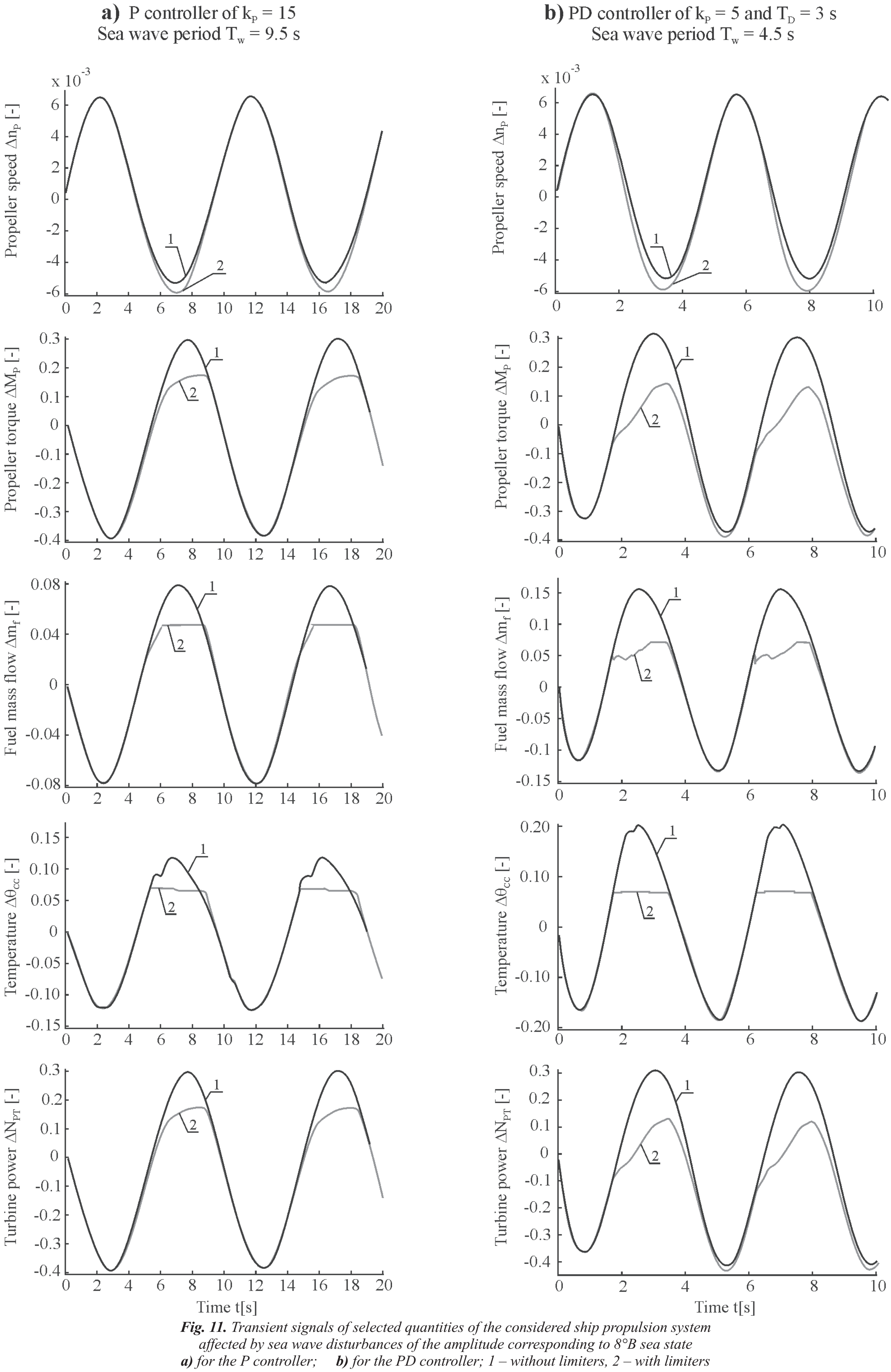



***Fig. 11.** Transient signals of selected quantities of the considered ship propulsion system affected by sea wave disturbances of the amplitude corresponding to 8°B sea state*
***a)** for the P controller;* ***b)** for the PD controller; 1 – without limiters, 2 – with limiters*

The results for the system with the P controller are given in Fig.11a for the sea wave of the mean period $T_W$ = 9.5 s, and in Fig.11b - the results for the system with the PD controller and the sea wave of $T_W$ = 4.5 s. The controllers with limiters make the correct work of the propulsion system possible, not allowing for exceedance of gas temperature and stresses in the propulsion shaft over the permissible limits both in transient and steady states. The work of the system in heavy seas without the limiters in the control system is inadmissible.

## The ship propulsion system's operation in heavy seas at different types and settings of the speed controller

The transient signals of the propulsion system operating in the heavy sea conditions corresponding to 8°B sea state were determined for two mean sea wave periods $T_W$ : 4.5 s and 9.5 s, with taking into account two types of the turbine speed controller: P and PD. The transient signals are presented in Fig.12.

***The transients of the propeller shaft speed $n_P$*** for all investigated controller settings show similar form and they do not much differ to each other. It results from the fact that it is the control signal.

***The transients of the propeller torque $M_P$*** for the P controller of $k_P$ = 5, and the PD controller of $k_P$ = 5 and $T_D$ = 0,1 s do not much differ to each other as well. The amplitudes of changes are the smallest at both sea wave periods $T_W$ : 4.5 s and 9.5 s and all considered controller's settings. The largest amplitudes of changes appear for the P controller of $k_P$ = 15 and the sea wave period $T_W$ = 9.5 s. For the period $T_W$ = 4.5 s the largest amplitudes appear for the PD controller of $k_P$ = 5 and $T_D$ = 3 s.

***The amplitudes of fuel flow to gas turbine combustion chamber, $m_f$,*** for the P controller of $k_P$ = 5 and the PD controller of $k_P$ = 5 and $T_D$ = 0,1 s, associated with the influence of the sea wave of $T_W$ = 4.5 s, are almost identical. For $T_W$ = 9.5 s at the same setting of the P controller they are the smallest. For the PD controller the amplitudes increase insignificantly in comparison with the case of the P controller. In both cases the disturbances due to sea waves do not cause any intervention of the limiters to the propulsion system. For the remaining considered settings of the controllers the amplitudes are much greater, and they are the largest at $T_W$ = 9.5 s for the P controller of $k_P$ = 15, whereas at $T_W$ = 4.5 s - for the PD controller of $k_P$ = 5 and $T_D$ = 3 s.

***The transients of the combustion chamber outlet temperature $\theta_{CC}$*** for the system operating in heavy sea conditions are similar to those for the fuel flow $m_f$.

***The transients of the gas turbine power $N_{PT}$*** at the disturbances due to sea waves are almost the same as the transients of the propeller torque $M_p$. And, for the P controller of $k_P$ = 5 and the PD controller of $k_P$ = 5 and $T_D$ = 0.1s at the wave period $T_W$ = 4.5 s they are almost identical, and the power oscillations due to sea waves amont to $\Delta N_{PT} = (N_{PT} - N_{PTo})/N_{PTo}$ = = 25 %; whereas at $T_W$ = 9.5 s - to $\Delta N_{PT}$ = 32 % for the P controller, and to $\Delta N_{PT}$ = 36 % for the PD controller. For the P controller of $k_P$ = 15 and the sea wave period $T_W$ = 9.5 s, the power oscillations amount to $\Delta N_{PT}$ = 55 % and are the largest out of the results for all considered controller settings. At the wave period $T_W$ = 4.5 s the power changes for the PD controller of $k_P$ = 5 and $T_D$ = 3 s are the largest (ca 60 %).

The performed analysis of the ship gas turbine propulsion system operating in heavy seas shows that in the considered system :

- for the P controller $k_P$ = 5 should be set; and
- for the PD controller $k_P$ = 5 and $T_D$ = 0.1 s
- however, for the PD controller at the constant value of its amplification coefficient and the increasing differential time constant the oscillations of the propulsion system parameters around their steady values set for operation in heavy seas, increase
- for higher sea state the amplitudes of the considered propulsion system parameters increase along with increasing the amplification coefficient $k_P$ of the P controller
- for the PD controller the amplitudes increase with sea state increasing, however at a given sea state the maximum amplitudes do not occur at the maximum sea wave period as it is the case for the P controller.

## INFLUENCE OF CONTROLLER TYPE AND SETTINGS ON OPERATION OF THE SHIP PROPULSION SYSTEM AT STEP CHANGES OF THE PROPELLER PITCH RATIO H/D

The influence of controller's type and settings on operation of the ship propulsion system in question at step changes of the propeller pitch ratio H/D was investigated for the case of calm sea. The same propulsion system simulation model as that presented in Part I, was used for the investigation.

Fig.13 shows the transient responses of the ship propulsion system on step changes of the propeller pitch ratio H/D within the range from 0.7 to 1. The controllers of two types, P and PD with limiters and different settings were applied.

The following control performance criteria were applied for the analysis of the influence of the controller's type and settings :

the minimum control time :

$$t_c = \min$$

and the integral performance criteria :

$$J_1 = \int_0^\infty \left| e(t) - e_{so} \right| dt$$

$$J_2 = \int_0^\infty \left( e(t) - e_{so} \right)^2 dt$$

Fig.14 shows the performance criteria in function of controller settings determined for the selected parameters of the ship propulsion system : $n_p$, $N_{PT}$, $\theta_{CC}$.

From the presented diagrams it results that for the investigated propulsion system at step changes of the ship propeller pitch ratio H/D :

*on the basis of the integral performance criterion $J_1$* :
the P controller of $k_P \approx 11$ should be taken as the optimum one,
*on the basis of the criterion $J_2$* :
also the P controller of the same $k_P$ value;
however no distinct minimum of this integral appears.

From the criterion of the minimum control time $t_c$ of the propeller speed $n_P$ it results that for the P controller a change of the controller amplification coefficient $k_P$ does not much influence the control time. For the PD controller the same criterion reveals that the minimum of the control time occurs within the interval of the differential time constant $T_D$ from 0.1 to 1s.

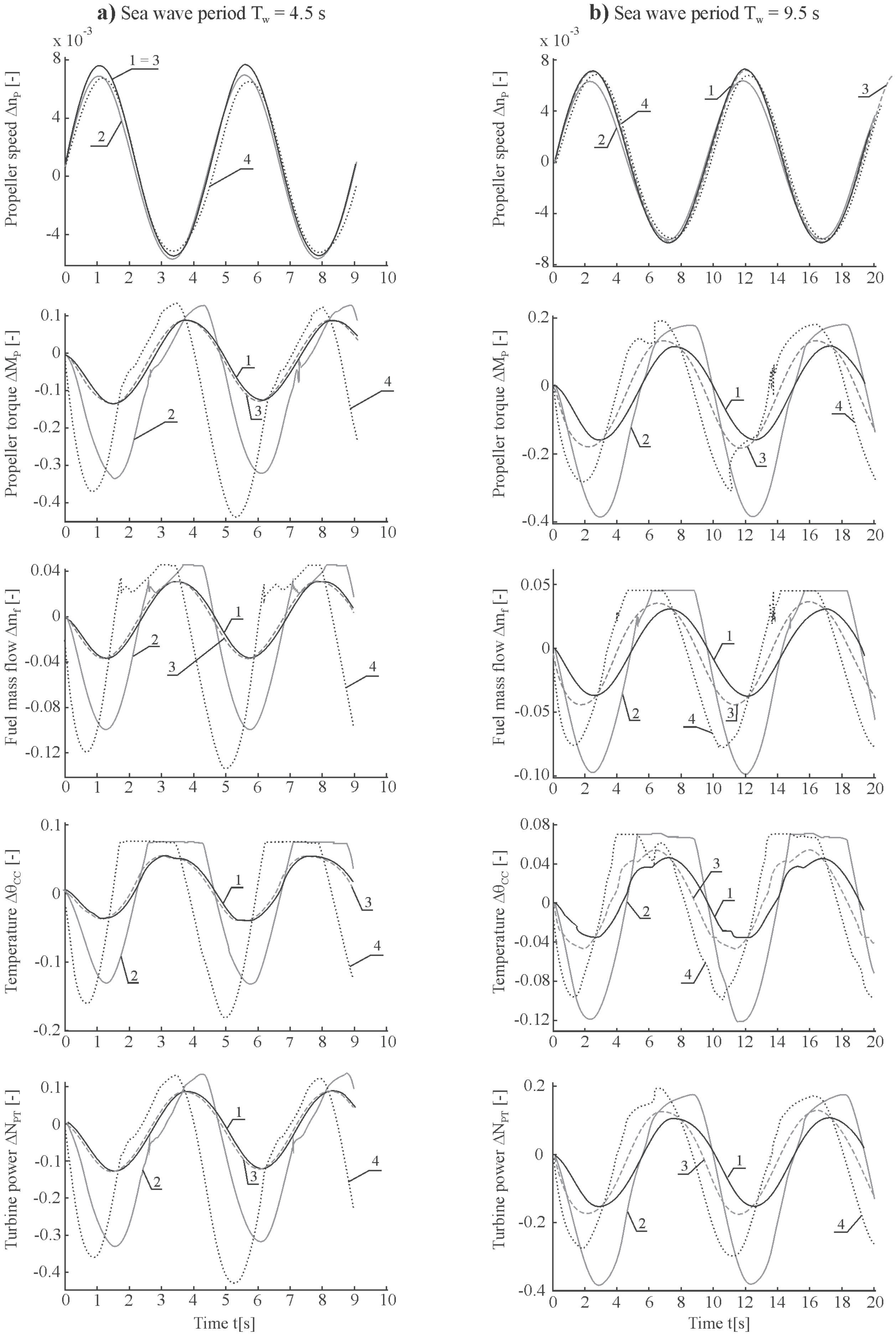



***Fig. 12.*** *Transient signals of the considered ship propulsion system affected by sea wave disturbances of the amplitude corresponding to 8°B sea state:* ***1*** *– P controller of $k_P = 5$;* ***2*** *– P controller of $k_P = 15$;* ***3*** *– PD controller of $k_P = 5$ and $T_D = 0.1$ s;* ***4*** *– PD controller of $k_P = 5$ and $T_D = 3$ s     Both controllers fitted with the limiters*

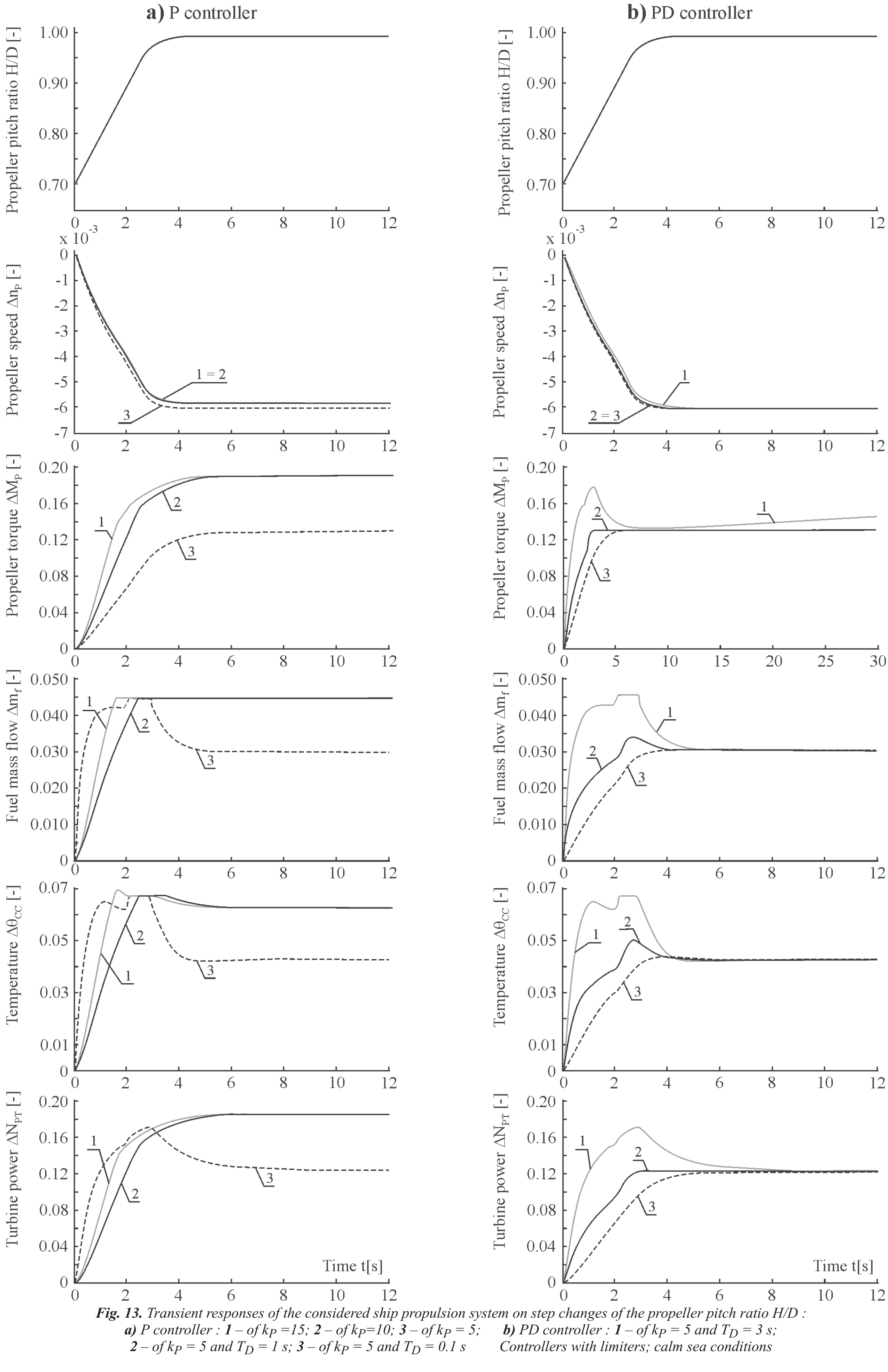
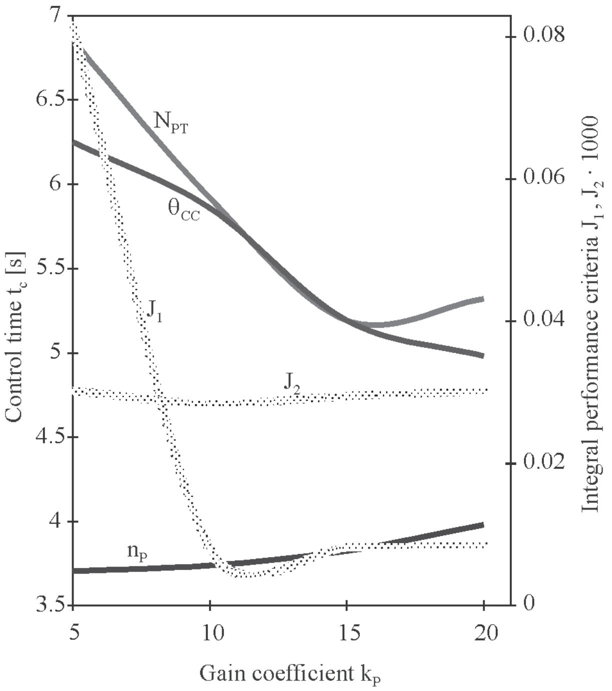



***Fig. 13.*** *Transient responses of the considered ship propulsion system on step changes of the propeller pitch ratio H/D :*
***a)*** *P controller :* ***1*** *– of $k_P$ =15;* ***2*** *– of $k_P$=10;* ***3*** *– of $k_P$ = 5;* ***b)*** *PD controller :* ***1*** *– of $k_P$ = 5 and $T_D$ = 3 s;*
***2*** *– of $k_P$ = 5 and $T_D$ = 1 s;* ***3*** *– of $k_P$ = 5 and $T_D$ = 0.1 s Controllers with limiters; calm sea conditions*

For such quantities as the gas turbine power $N_{PT}$ and the combustion chamber outlet temperature $\theta_{CC}$, the minimum stabilization time in response to step changes of the propeller pitch ratio H/D is achieved:

*in the case of the P controller*
for $k_P$ value within the interval from 15 to 20
*in the case of the PD controller*
for its optimum settings : $k_P$ = 5 and $T_D$ = 1s.

**a)** P controller




**b)** PD controller of $k_P$ = 5

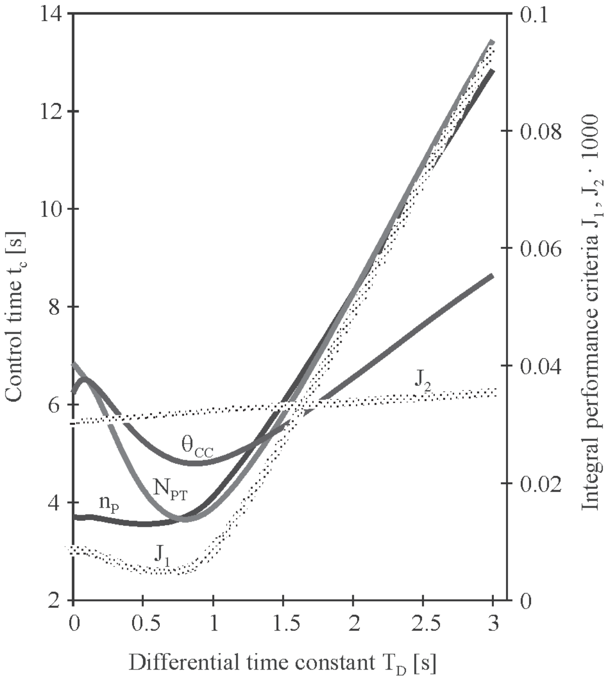



***Fig. 14.*** *The integral control performance criteria $J_1$ and $J_2$ and the ship propulsion system control time $t_c$ in function of the controller settings $k_P$ and $T_D$ at step changes of the propeller pitch ratio H/D, determined for the following parameters of the ship propulsion system: $n_P$ , $N_{PT}$ , $\theta_{cc}$ . Controllers with limiters; calm sea conditions*

## FINAL REMARKS

The presented analysis demonstrated that :

- For the investigated ship propulsion system with gas turbines driving - through mechanical gear – controllable pitch propeller the optimum controller settings for calm sea conditions are not optimum ones for the system working in heavy sea conditions.
- In the control loop of the propulsion system the maximum fuel flow limiter and that saveguarding against increase of combustion chamber outlet temperature should be applied.
- The optimum setting values for the P and PD controllers in regard to sea wave disturbances are lower than the optimum ones regarding to change of ship propeller pitch. It seems reasonable to adjust the controller to the lower setting values, i.e. those obligatory for the propulsion system working in heavy sea conditions. In this case the oscillation amplitudes of its particular quantities are much smaller than those for the controller adjusted to the optimum settings in regard to change of pitch of CP propeller.

**NOMENCLATURE**

$c_m$ - mean speed of water flow onto propeller, caused by sea waves
e - error
$e_{so}$ - error in steady state
$h_P$ - mean depth of immersion of propeller axis
$k_p$ - controller amplification coefficient
$m_f$ - fuel mass flow
t - time
$T_D$ - differential time constant
$T_w$ - sea wave period
$V_o$ - linear speed of ship
$V_P$ - propeller advance speed
$\theta_{CC}$ - combustion chamber outlet gas temperature
$\omega_m$ - frequency of sea - wave - induced disturbance (excitation)

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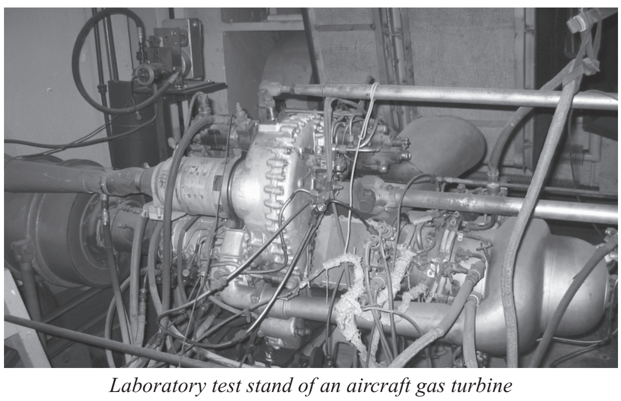
**CONTACT WITH THE AUTHOR**
D.Sc. Marek Dzida
Faculty of Ocean Engineering
and Ship Technology
Gdańsk University of Technology
Narutowicza 11/12
80-952 Gdańsk, POLAND
e-mail : dzida@pg.gda.pl



*Laboratory test stand of an aircraft gas turbine*

# Research on influence of some ship diesel engine malfunctions on its exhaust gas toxicity

**Jacek Krzyżanowski, Kazimierz Witkowski**
Gdynia Maritime University

ABSTRACT

*This paper is a continuation of the previous articles of the authors, published in Polish Maritime Research [1, 2], devoted to pollution of the atmosphere due ship diesel engines in operation. In the paper presented are results of the investigations carried out in the Gdynia Maritime Academy laboratory with the use of a ship one-cylinder diesel engine combusting heavy fuel oil. Two its possible malfunctions : changes of fuel injection pressure and changes of injection advance angle were simulated. Both malfunctions, which might occur as a result of incorrect engine regulation or wearing or contamination of engine elements, were simulated separately. The presented investigations were focused on finding out relationships between the assumed malfunctions and exhaust gas content, especially content of nitric oxides ($NO_X$),very toxic compounds.The paper is ended by several conclusions arising from analysis of the results which have – apart from their cognitive merits – also an utilitarian character as they may be put into practice by ship operators, provided that some limitations resulting from the specific conditions of the reported laboratory investigations are appropriately taken into account.*

**Key words** : atmosphere protection, emission of nitric oxides, laboratory test

## INTRODUCTION

Problems of the sea water protection against pollution from ships have been dealt by ship designers and operators for more than a half of century. In recent years the marine environmental protection was extended over the atmosphere. The revelant legal acts aiming at lowering emission of toxic exhaust gas components, especially nitric oxides ($NO_X$), are in force (App. IV to MARPOL Convention, ISO 8178 standard [3]).

Technical undertakings to cope with the requirements determined in the standards go in two directions. The first one deals with new engines for which the designers and producers search for solutions which would make it possible to obtain $NO_X$ emission low enough to comply with the standards. The other trend is connected with the engines in service. During their designing no sufficient attention was paid to exhaust gas purity. However today they have to be adjusted to comply with the being-in-force requirements for lowering nitric oxides ($NO_X$) emission. As it was proved in the publications [1], [2], [4], [5], [6], [7] a relatively simple way to limit emission of nitric oxides is to apply appropriate adjustment operations.

Despite the adjustment of ship diesel engines is made to comply with the exhaust gas purity standards, the emission may be changed as a result of changing technical state of the engines during service. In this paper presented are results of the research on influence of some injection system's malfunctions of a ship diesel engine on its exhaust gas content. The investigations were performed as an active experiment on the engine charged with a heavy fuel oil and operating in the laboratory conditions.

## LABORATORY TESTS

### *The test object and stand*

The laboratory tests were carried out on the one-cylinder, two-stroke, crosshead engine of longitudinal scavening, which was loaded by means of the water brake. The engine was charged with IF 40 heavy fuel oil. The whole stand together with its measurement instrumentation was described in [1]. In comparison to the earlier presented stand a basic change consisted in adding an Alfa-Laval viscometer which made it possible to continuously measure fuel viscosity and stabilize it at a demanded level.

### *Scope of the tests*

The test program was prepared for performing series of measurements on the engine with two simulated malfunctions of the injection system, as follows :

- changing the fuel injection pressure - to three selected values : 180, 220 and 260 bar
- changing the fuel injection advance angle - to three selected values : -10°, -13° and -16° before the piston's upper dead centre (UDC).

The measurements were performed within the wide range of engine load at the permanent rotational speed of 220 rpm. For each of the above selected values six measurements were performed at the following engine loads : 25%, 40%, 50%, 60%, 70% and 80%. The engine load was defined as the percentage ratio of a given engine torque and that rated.

## *Description of the tests and their results*

The injection system's malfunctions were introduced – during operation of the engine – by means of a supplementary scaled instrumentation of the injector and injection pump. Owing to this it was not necessary to stop the engine before each successive test cycle.

This way of realization of the measurements guaranteed runing them in steady conditions. The elimination of influence of possible disturbances which could arise from multiple starting and stopping the engine improved accuracy and reliability of the obtained results.

The measurement results are presented in Tab.1 and 2 and in Fig.1 and 2.

***Tab.1.*** *Results of analysis of exhaust gas content at three different injector opening pressures*

| Fuel injection pressure | Relative engine load | Exhaust gas content | | | | | |
|---|---|---|---|---|---|---|---|
| p | $M/M_r$ | $O_2$ | CO | $SO_2$ | $NO_x$ | $NO_x$ | $CO_2$ |
| [bar] | [%] | [%] | [ppm] | [ppm] | [ppm] | [mg/m$^3$] | [%] |
| 180 | 25 | 19.0 | 199 | 21 | 136 | 186 | 1.4 |
| | 40 | 18.1 | 238 | 17 | 183 | 251 | 2.1 |
| | 50 | 17.3 | 320 | 17 | 250 | 343 | 2.7 |
| | 60 | 16.7 | 337 | 14 | 316 | 434 | 3.1 |
| | 70 | 14.3 | 1 022 | 21 | 388 | 533 | 4.9 |
| | 80 | 12.2 | 2 849 | 50 | 397 | 545 | 6.4 |
| 220 | 25 | 19.0 | 159 | 27 | 155 | 212 | 1.4 |
| | 40 | 18.2 | 167 | 27 | 211 | 289 | 2.0 |
| | 50 | 17.4 | 243 | 21 | 267 | 366 | 2.6 |
| | 60 | 15.8 | 267 | 19 | 360 | 494 | 3.8 |
| | 70 | 13.7 | 707 | 23 | 436 | 599 | 5.3 |
| | 80 | 9.0 | 6 043 | 30 | 382 | 524 | 8.8 |
| 260 | 25 | 18.8 | 193 | 33 | 186 | 255 | 1.5 |
| | 40 | 18.2 | 209 | 32 | 240 | 329 | 2.0 |
| | 50 | 17.6 | 336 | 32 | 291 | 399 | 2.4 |
| | 60 | 16.0 | 376 | 48 | 369 | 507 | 3.6 |
| | 70 | 13.5 | 1 228 | 76 | 424 | 582 | 5.5 |
| | 80 | 10.8 | 4 050 | 128 | 407 | 559 | 7.4 |

***Tab.2.*** *Results of analysis of exhaust gas content at three different advance angles of fuel injection*

| Fuel injection advance angle | Relative engine load | Exhaust gas content | | | | | |
|---|---|---|---|---|---|---|---|
| $\alpha$ | $M/M_r$ | $O_2$ | CO | $SO_2$ | $NO_x$ | $NO_x$ | $CO_2$ |
| [$^o$] | [%] | [%] | [ppm] | [ppm] | [ppm] | [mg/m$^3$] | [%] |
| -10 | 25 | 18.9 | 210 | 29 | 117 | 160 | 1.5 |
| | 40 | 18.3 | 215 | 24 | 161 | 221 | 1.9 |
| | 50 | 17.4 | 263 | 21 | 215 | 295 | 2.6 |
| | 60 | 16.6 | 345 | 19 | 266 | 365 | 5.2 |
| | 70 | 14.0 | 610 | 24 | 358 | 491 | 5.1 |
| | 80 | 9.9 | 9 882 | 53 | 297 | 408 | 8.1 |
| -13 | 25 | 19.0 | 159 | 27 | 155 | 212 | 1.4 |
| | 40 | 18.2 | 167 | 27 | 211 | 289 | 2.0 |
| | 50 | 17.4 | 243 | 21 | 267 | 366 | 2.6 |
| | 60 | 15.8 | 267 | 19 | 360 | 494 | 3.8 |
| | 70 | 13.7 | 707 | 23 | 436 | 599 | 5.3 |
| | 80 | 9.0 | 6 043 | 30 | 382 | 524 | 8.8 |
| -16 | 25 | 19.1 | 264 | 25 | 215 | 295 | 1.3 |
| | 40 | 18.3 | 231 | 28 | 325 | 446 | 1.9 |
| | 50 | 17.1 | 418 | 30 | 400 | 549 | 2.8 |
| | 60 | 15.6 | 622 | 28 | 499 | 685 | 3.9 |
| | 70 | 13.3 | 1 170 | 34 | 563 | 773 | 5.6 |
| | 80 | 8.8 | 8 061 | 68 | 464 | 637 | 8.9 |

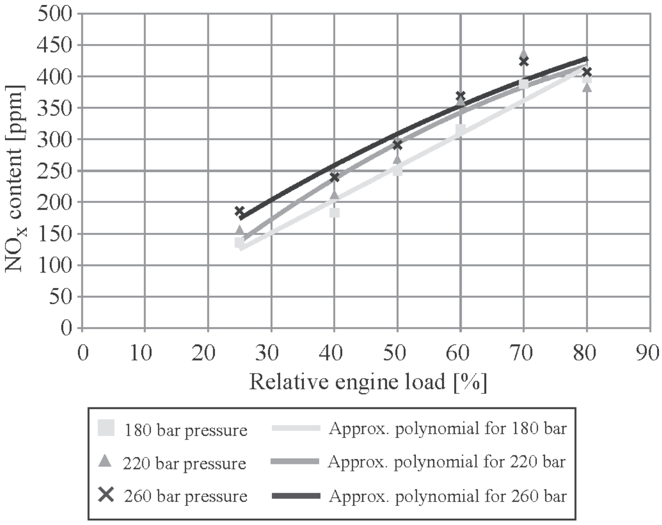



***Fig.1.*** *$NO_x$ content in exhaust gas in function of three different injector opening pressures*

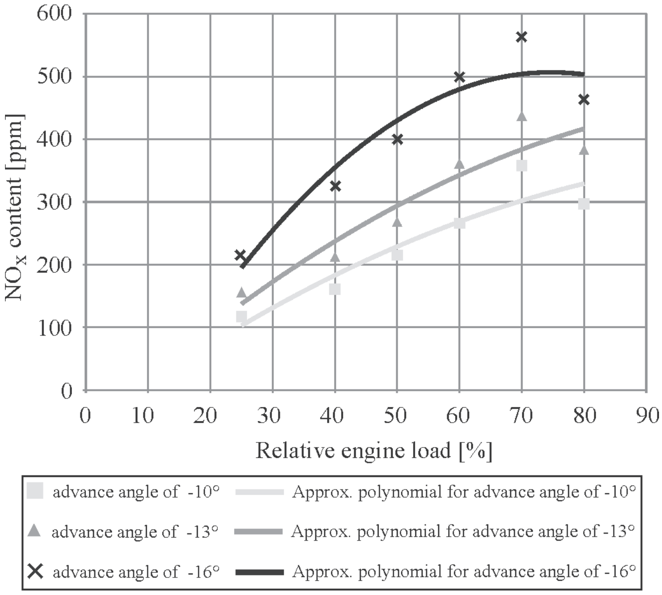



***Fig.2.*** *$NO_x$ content in exhaust gas in function of three different advance angles of fuel injection*

## *Analysis of the results*

The rated value of the injector opening pressure for the tested engine amounts to 220 bar. A change of the pressure may be caused by an incorrect initial setting of the injector (operator's mistake), or its maladjustment during service (a change of technical state of injector's spring). The chosen values differed by 20% of that rated, i.e. they amounted to 180 and 260 bar.

A drop of the injector opening pressure makes that the fuel is injected to the cylinder a little earlier and the injection lasts longer, and spraying the fuel, especially its first portions, is worser. The worsening of the combustion process resulting from that makes $NO_x$ level within the whole range of the applied engine loads, lower.

The rise of the injector opening pressure from 220 bar to 260 bar results in an increase of $NO_x$ emission within the whole engine load range. This can be caused due to improving the quality of fuel spraying and combustion process, which leads to a higher combustion temperature and – in consequence – to a higher $NO_x$ content in exhaust gas.

An incorrect setting of the injector opening pressure, not complying with that recommended by the producer, is always considered as a malfunction which should be removed as soon as possible. However, from the ecological point of view the drop of injector opening pressure makes $NO_x$ content in exhaust gas lower. Hence it may be a simple way which makes it possible to so adjust an existing engine as the $NO_x$ emission standards to be satisfied. However the accompanying consequences should be also remembered, namely : increasing the fuel consumption and possible rising the thermal load on engine combustion chamber elements.

The malfunction consisting in a change of the injection advance angle may result from an incorrect initial adjustment of the engine, or from a random change of that quantity during the engine's operation process. The three following values of the advance angle were selected : -16°, -13° (rated) and -10°.

An increase of the injection advance angle unambiguously leads to important increments of $NO_x$ content in exhaust gas within the entire range of the engine's load, whereas a delay of the injection beginning (a decrease of the injection advance angle) unambiguously makes $NO_x$ content in exhaust gas lower.

However it should be remembered that both an advance and delay of fuel injection starting – in relation to the values recommended by the producer and set during statical regulation of the engine – influences not only the exhaust gas content but also other important operational parameters of the engine by changing combustion process quality. An increase of the advance angle causes an increase of the maximum combustion pressure $p_{max}$ which may lead to mechanical overloading, and its drop – to increasing the fuel oil consumption by dropping $p_{max}$ and increasing exhaust gas temperature.

The investigations also revealed (Tab.1 and 2) that the simulated malfunctions made content of carbon oxide (CO) in exhaust gas greater. Both an increase and decrease of the injector opening pressure and injection advance angle with respect to their rated values led to the above mentioned change.

## CONCLUSIONS

On the basis of the performed tests
the following general conclusions may be offered :

- ➲ The engine's malfunction consisting in an increased injector opening pressure makes $NO_x$ emission to the atmosphere greater.
- ➲ The malfunctions consisting in decreasing the fuel injection pressure and in delaying the fuel injection beginning favourably reduce $NO_x$ emission to the atmosphere. However it should be remembered that excessive deviations from the rated settings recommended by the engine producer may negatively affect its operational cost by increasing the fuel oil consumption and its durability (cost of engine repairs).
- ➲ Occurrence of the considered malfunctions increases the exhaust gas toxicity by increasing content of carbone oxide (CO).
- ➲ In practical applications all the conclusions should be considered with taking into account the limitations associated with the used test object and the realized testing program.

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**CONTACT WITH THE AUTHORS**
D.Sc. Jacek Krzyżanowski
D.Sc. Kazimierz Witkowski
Faculty of Mechanical Engineering
Gdynia Maritime University
81-225 Gdynia, POLAND
Morska 81-87
e-mail : jacekk@am.gdynia.pl


# Miscellanea

## TOP KORAB
### awards in the academic year 2002/2003

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 2002/2003
the awardwinnings were :

- Mr Lech Wykrętowicz, M.Sc. for the project on : *"Preliminary design of diesel power plant of a 150 000 dwt universal bulk carrier of 15 kn speed"*
  The project was elaborated under supervision of Mr. Jerzy Jamroż, D.Sc., the Department of Ship Power Plants.
- Mr Krzysztof Królak, M.Sc. for the project on : *"Preliminary design of a 2000/1500 TEU cellular containership for West-american shipping line"*
  The project was elaborated under supervision of Mr. Bogusław Oleksiewicz, D.Sc., the Department of Designing Ships and Ocean Engineering Objects.

# Submarine atmosphere regeneration

**Ryszard Kłos**
The Naval University of Gdynia

ABSTRACT

*In the paper was presented introduction of Polish soda lime (produced by the Chemical Company Dwory SA) successfully used for filling the carbon dioxide scrubber of a regeneration system built in KOBBEN class submarines for eliminating carbon dioxide from its atmosphere without any additional construction changes. Research on the atmosphere regeneration was carried out on a KOBBEN class submarine. The initial $CO_2$ concentration was achieved by releasing content of two cylinders filled with $CO_2/N_2$ gas mixture. Further $CO_2$ emission adequate to its maximum production by the ship's crew during breathing, was simulated by means of a constant emission simulator.The time of the preventive operation amounted to about 2 hours. The dynamic absorbing capacity of carbon dioxide was validated experimentally.*

**Key words** : submarine, live support systems (LSS), soda lime

## INTRODUCTION

### *Regeneration procedure*

Air regeneration of disabled/distress submarine (DISSUB) is necessary due to tactical or emergency reasons when the vessel is not able to emerge on the surface and to ventilate its atmosphere. Regeneration of DISSUB's atmosphere consists in oxygen enriching and excessive carbon dioxide amount removing. In most situations carbon dioxide elimination is provided by means of soda lime which is a chemical compound used for removing carbon dioxide from gas mixtures. Soda lime consists of calcium hydroxide $Ca(OH)_2$, sodium hydroxide NaOH and/or potassium hydroxide KOH. An approximate content of NaOH and/or KOH is $2 \div 5\%_m$ of the solid phase. On submarines soda lime is placed in a separate absorber connected to an appropriate ventilation system. A suitable reserve of soda lime is also saved on board.

Procedures of using soda lime on the DISSUBs can be different, but it is usually accepted that if carbon dioxide content reaches $1\%_{vol.}$, the absorption of carbon dioxide has to be started. At $2\%_{vol.}CO_2$ content the DISSUB must be ventilated or prepared for abandonment.

Mean oxygen consumption by one human is assumed to be ca 25 $dm^3 \cdot hour^{-1}$, and it is generally accepted that the ratio of $CO_2$ emission and oxygen consumption, per person, (respiratory quotient) amounts to $\xi = 0.8$ [2]. It means that $CO_2$ emission produced by one person during the respiration process, amounting to ca 20 $dm^3 \cdot hour^{-1}$, is assumed. These values are used to calculate the reserve of soda lime for the DISSUB.

### *Soda lime qualities*

Soda lime porosity and bulk density are most important at designing the carbon dioxide scrubber but these qualities are not available from producers. Therefore, it has been necessary to determine soda lime porosity and bulk density.

Bulk density was measured by weighing in air (1000± ±100) $cm^3$ of the bed. The soda lime bed mass was (801±20) g being equivalent to the soda lime bulk density d = (0.80±0.02) $kg \cdot m^{-3}$ as the buoyancy correction could be neglected due to its low value of 1g only, i.e.less than the mass measuring error [1].

***Tab. 1.*** *Basic qualities of granulated soda lime*

| Item | Value |
|---|---|
| Properties covered by the Polish industrial standard : № *ZN-2001/"DWORY"S.A.-70* | |
| Appearance | White or white-grey pellets of the diameter 2.8÷3.5 mm |
| Absorbing capacity with the respect to carbon dioxide | Not less than 26% (by mass) |
| Absorbing capacity with the respect to moisture | Not exceeding 7.5% (by mass) |
| Negative mesh, square mesh dimension 2.0 mm | Not exceeding 5% (by mass) |
| Moisture content | 10÷18% (by mass) |
| Drum strength according to the standard PN-81/c-04533 | Not less than 92% (by mass) |
| Additional properties necessary for fore-designing of the soda lime bed | |
| Bulk density | $(0.80 \pm 0.02)\ kg \cdot m^{-3}$ |
| Porosity | 0.55±0.02 |
| Hydraulic diameter | $(3.470 \pm 0.003) \cdot 10^{-3} m$ |

Soda lime bed porosity, defined as the total free inter-molecular volume to the total volume occupied by a granulated material, was determined by water flooding the bed of the height l = (75±2) mm and diameter D = (90±2) mm, and the volume equal to V = (500±5) $cm^3$, consisting 10236 granules. The volume of the water necessary for flooding the bed was $V_W$ = (276±2) $cm^3$ which gave the porosity $\varepsilon$ = (0.55±0.02), typical for granulates.

Such determination of the mean soda-lime hydraulic diameter was approximate as the soda-lime granules are additionally twisted (thus measurement results may differ even by 10÷20%). According to Sauter [1], the mean hydraulic diameter of soda lime determined from an earlier experiment has been equal to $(3.470\pm0.003)\cdot10^{-3}$m.

### *Scrubber foredesign*

The example calculations of the carbon dioxide scrubber are presented in Tab.2 on the assumption that :

- the air, being in laminar flow, passes through the fully filled scrubber
- the resistance (consisted of that imposed by scrubber construction, the sorbent bed resistance and the scrubber body resistance) is relatively small in comparison with the sorbent bed resistance
- the resistance coefficient for granulates (for one granulate fraction) can be defined according to Blake-Kozena's definition [1]
- the particle diameter of the granular bed is determined according to the Sauter's definition
- air viscosity is not dependent on temperature and pressure [3]
- 1 $dm^3$ of soda lime approximately bonds 90 $dm^3$ of $CO_2$ , [3].

***Tab. 2.*** *Calculation of the sorbent bed dimensions [3]*

| Data : | |
|---|---|
| Flow rate of the breathing gas | $\mathring{V} = 1320\ dm^3\cdot min^{-1}$ |
| Porosity of the bed | $\varepsilon = 0{,}55$ |
| The granule diameter according to Sauter | $d_S = 3.470 \cdot 10^{-3}$ m |
| Shape factor | a = 730 |
| The breathing gas viscosity (air) | $\eta = 1.85 \cdot 10^{-5}$ Pa·s |
| Flow resistance | 100 Pa |
| Maximum volume of the absorbed $CO_2$ | V=2100 $dm^3$ |
| Calculations : | |

$$\begin{cases} l = \dfrac{\varepsilon}{1-\varepsilon}\cdot d_s \cdot \sqrt{\dfrac{\varepsilon}{a\cdot\eta}}\cdot\sqrt{\dfrac{\Delta p}{\mathring{V}}\cdot\dfrac{V(CO_2)}{90}} \cong 0{,}278\ \text{m} \\ d_h = \dfrac{2}{\sqrt{\pi}}\sqrt{\dfrac{V(CO_2)}{90\cdot 1}} \cong 0{,}327\ \text{m} \\ \dfrac{l}{d_h} \cong 0{,}852 \end{cases}$$

l – height of the bed
$d_h$ – hydraulic diameter
$\Delta p$ – bed resistance

## OBJECT

The submarine soda-lime scrubber of the internal dimensions of ca 299 x 278 mm and 276 mm height, was filled with (18.73±0.02)kg of the soda lime (Tab.1.) The calculated bulk density was d = 0.82 kg · $dm^{-3}$ which corresponded to the density earlier determined in the laboratory. An overall view and technical drawing of the canister is given in Fig.1.

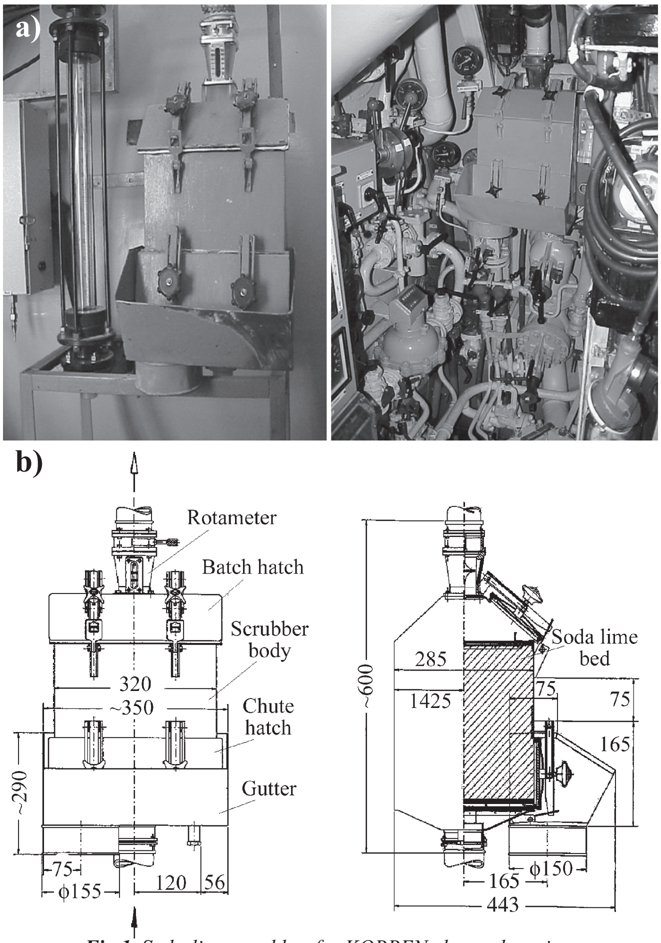


***Fig.1.*** *Soda-lime scrubber for KOBBEN class submarine:*
***a)*** *experimental stand and built in submarine*
***b)*** *engineering drawing*

The total flow resistance (the channel flow resistance and bed flow resistance) in the carbon dioxide scrubber is measured with the aid of a differential manometer. For the air flow rate (1320±10) $dm^3\cdot min^{-1}$ the bed resistance was $\Delta_p$ = (100± ±2) Pa. The shape factor of soda-lime particles, necessary for scrubber foredesign, was determined from the measurement results and soda-lime qualities (Tab.1), as shown in (Tab.2).

## METHOD

### *Carbon dioxide emission*

Research on the atmosphere regeneration was carried out on a submarine of KOBBEN class. Gas cylinders filled with (3.50±0.0) kg of $CO_2$ and (0.25±0.0) kg of $N_2$ were prepared for simulating the initial $CO_2$ concentration in the DISSUB atmosphere.

The initial $CO_2$ concentration was realized by releasing content of the two cylinders into DISSUB's atmosphere (Fig.2). After stabilization of $CO_2$ concentration (ca 10 min - Fig.3), the stable $CO_2$ content amounts to (1.16±0.05)$\%_{vol.}CO_2$ (Fig.3). Further $CO_2$ emission, adequate to its maximum production by the ship's crew during breathing, was simulated by means of a special simulator [1]. Its emission amounted to (10.8±0.05) $dm^3$ of $CO_2$ per min.

### *Monitoring system*

The submarine was equipped with a portable atmosphere monitoring system. The monitoring system consisted of four POLYTRON IR $CO_2$ gas analyzers produced by Drägerwerk

AG Lübeck, placed at bow, midship, stern and a hydro-cabin. The analyzers were connected through a typical personal computer and Advantech data acquisition modules into a RS 485 measuring network (Fig.2). Readouts was recorded every 5 s. The PC unit connected to the RS485 measuring network was capable of monitoring the submarine's atmosphere by means of a special computer program.

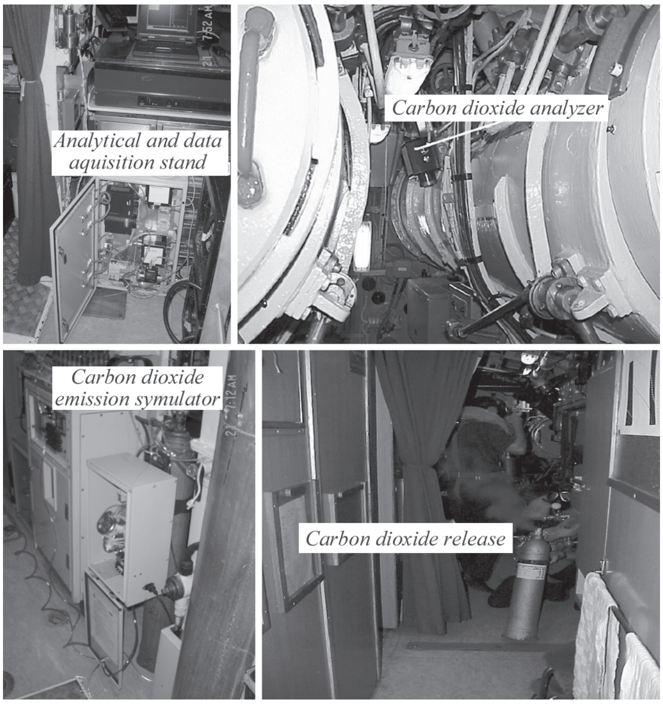


***Fig.2.*** *The system of measuring, $CO_2$ releasing and dosing it into DISSUB atmosphere*

## *Regeneration of the DISSUB's atmosphere*

The scrubber of the regeneration system (Fig.2) was filled with 13,2 kg of soda lime (Tab.1). Regeneration of the DISSUB's atmosphere was initiated with start of carbon dioxide simulator emission. The regeneration was carried out as long as one of the analyzers did not show stable $CO_2$ content of the same value as that before starting a regeneration system. The time of the preventive operation amounted to about 2 hours.

If the dynamic absorbing power of 90 $dm^3$ of $CO_2$ per 1 $dm^3$ of soda lime, is assumed, the soda lime amount placed into the canister (13.2 kg corresponds to 16.1$dm^3$ of soda lime) should absorb about 1450 $dm^3$ of $CO_2$. The $CO_2$ saturation in the soda lime was measured; the measurement results are shown in Tab.3.

***Tab. 3.*** *Carbon dioxide saturation in the soda lime samples located in different layers of canister*

| Sample location | Moisture content [%] | Content of absorbed $CO_2$ [$\%_m$] |
|---|---|---|
| Upper part of canister | 3.2 | 6.8 |
| Middle part of canister | 4.1 | 14.9 |
| Middle part of bottom canister | 3.7 | 10.6 |
| Lower part of canister | 3.6 | 12.0 |
| Average | 3.6 | 11.1 |
| Fresh soda lime | 15.7 | 28.7 |

## CONCLUSIONS

- The simple and effective foredesign method of a $CO_2$ removing scrubber was proved experimentally.
- The soda lime complying with the Polish industrial standard : ZN-2001/"DWORY" S.A.-70 was successfully used for filling the $CO_2$ removing scrubber of the atmosphere regeneration system installed in a KOBBEN class submarine.
- The dynamic absorbing capacity of carbon dioxide was 90 $dm^3$ of $CO_2$ per 1$dm^3$ of soda lime, due to which the effective absorbing capacity of the system in question was guaranteed.

## Acknowledgement

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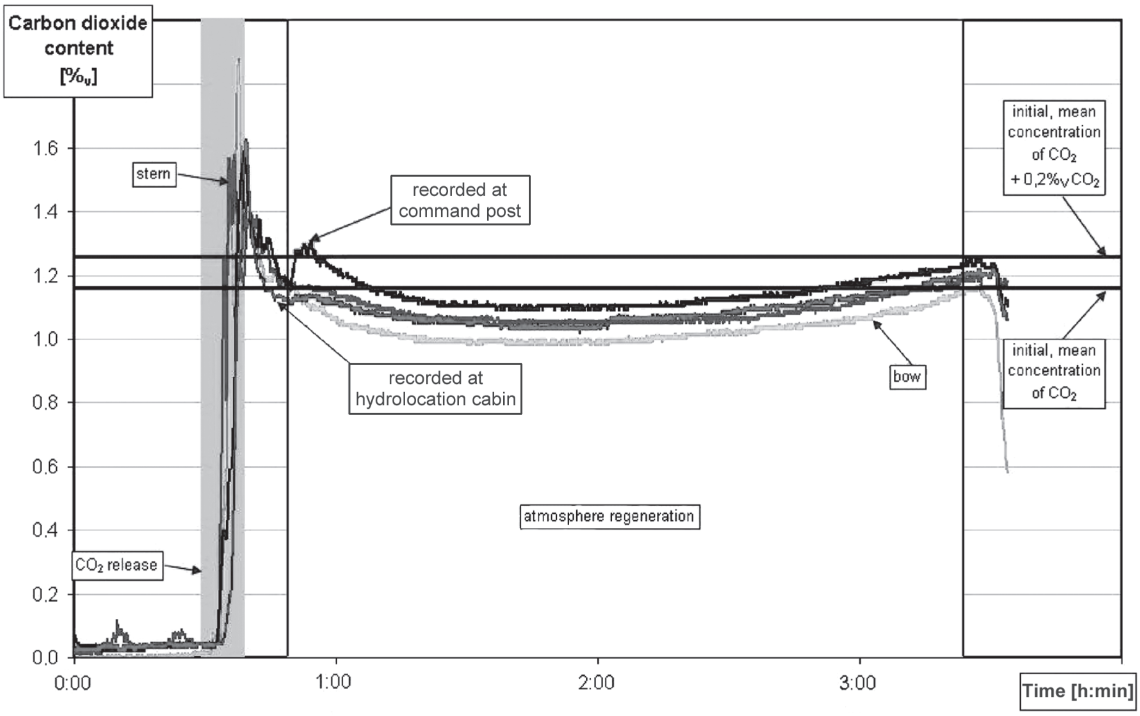


***Fig.3.*** *Rate of carbon dioxide elimination by built in KOBBEN class submarine regeneration system*

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**CONTACT WITH THE AUTHOR**
D.Sc. Ryszard Kłos
Department of Diving Gear
and Underwater Work Technology
Naval University
Śmidowicza 69
81-103 Gdynia, POLAND
e-mail : skrzyn@wp.pl

Conference

# SILWOJ 2003

On 22-24 October 2003
6th Scientific Technical Symposium on :

***Combustion Engines in Military Applications***

was held at Jurata on Hel Peninsula. It was organized by Polish Naval University, Gdynia and Military Engineering Academy, Warsaw.

During the Symposium 57 papers were presented : 14 of which - at 4 plenary sessions and the remaining - during 5 topical sessions on :

- Fuels and oils (6 papers)
- Construction and engineering processes (13 papers)
- Drives and controls (7 papers)
- Operation and Diagnostics (13 papers)
- Technology and construction (4 papers)

The papers dealt with applications of combustion engines in army, airforce and navy, and most of them concerned construction and engineering processes as well as operation and diagnostics of the engines. Scientific workers of Military Engineering Academy, Warsaw, most contributed in preparation of the symposium materials (22 papers), 11 papers were prepared by authors from Polish Naval University, and 5 by those of Rzeszów University of Technology. The remaining papers were elaborated by authors from 12 other universities, institutes and research centres.

Conference

# SemEco 2003

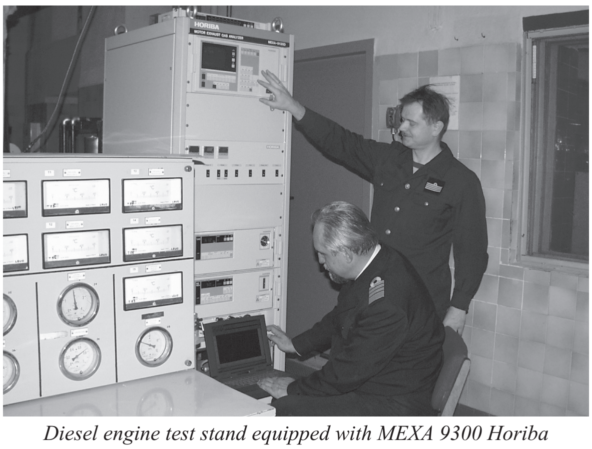
Under this acronym 2nd series of seminars devoted to ecological problems of operation of combustion engines and organized by Naval University, Gdynia, was held in the last year. The seminars have been one of the activities of Maritime Technology Unit, Section of Transport Means, Transport Committee, Polish Academy of Sciences; and their main organizer was Cmdr., Prof. L. Piaseczny.

**The following papers directly concerned ecological problems :**

- *Problems of application of biofuels in the light of the present development state of piston combustion engines* – by J. Merkisz (Technical University of Poznań)
- *Working and ecological features of the engines supplied with rape oil, rape-oil methyl esters and their mixtures with fuel oil* – by Z. Szlachta (Technical University of Kraków)
- *Inventory problems of contaminations emitted by combustion engines* – by Z. Chłopek (Warsaw University of Technology)
- *Influence of partition of fuel dosage injected to diesel engine on its cycle parameters and $NO_x$ emission* by I. Kafar (Polish Naval University)
- *Ecological aspects of application of controlled timing gears* – W. Kozaczewski (Industrial Institute of Motorization, Warsaw).

**The remaining papers dealt with other interesting topics regarding combustion engines :**

- *Steam-gas engines applicable to torpedos – their construction, working parameters and operational features* – by St. Czarnecki (Polish Naval University)
- *Selected problems of modelling supply processes in combustion engines* – by M. Sobieszczański (Academy of Engineering and Humanistics, Bielsko-Biała)
- *Application of connection graphs and state equations to modelling energy systems* – by M. Cichy (Gdańsk University of Technology)
- *Modelling the process of charge inlet into engine's cylinder* – by M. Łutowicz (Polish Naval University)
- *Identification of the compression process in ship engine cylinder on the basis of indication diagram data* by S. Polanowski (Polish Naval University).



*Diesel engine test stand equipped with MEXA 9300 Horiba exhaust gas analyzer, at Polish Naval University, Gdynia*

# Since the beginning of the year 2004
# POLISH MARITIME RESEARCH
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## Faculty of Ocean Engineering and Ship Technology

## Gdańsk University of Technology

Among 10 faculties of the Gdańsk University of Technology (GUT) which its academic community currently consisted of 17000 students and 2500 staff, one of the oldest is the Faculty of Ocean Engineering and Ship Technology (FOEST), being the only in Poland which has continuously been educating engineers, doctors and professors in the field of marine technology since 1945.

During the 55 years of post-war activities, the FOEST educated about 5000 B.Sc. and M.Sc. graduates, as well as about 260 persons of Ph.D. and D.Sc. degrees. Eleven of the Faculty's professors were granted the dignity of Doctor Honoris Causa, and 3 of them – chairs of Polish Academy of Sciences.

### ORGANIZATION

**The Faculty is organised of the following departments of** :

- ★ Mechanics and Strength of Ship Structures
- ★ Underwater Technology, Hydromechanics and Design of Ships
- ★ Ship Manufacturing, Materials and Quality Systems
- ★ Ship Power Plants
- ★ Ship and Ocean Structures Deck Equipment and Systems
- ★ Ship Automation and Turbine Propulsion.

**The main facilities of the Faculty are** :

- ❖ The main building of the Faculty including the administration offices, two large lecture theatres and a number of other lecture rooms, research rooms including the computer facilities, computer laboratories, technical laboratories
- ❖ Ship Hydromechanics Laboratories
- ❖ Ship Automation and Turbines Laboratory
- ❖ Ship Deck Equipment and Ship Machinery Laboratory
- ❖ Ship Research and Training Centre in Iława
- ❖ Laboratory of Ship Hull Structures
- ❖ Open Air Laboratory for Ship Hydrodynamics in Iława
- ❖ Towing Tank and Cavitation Tunnel
- ❖ Ship Machinery Laboratory
- ❖ Laboratory for Underwater Vehicles and Composite Structures
- ❖ Laboratory of Material Science.

### EDUCATIONAL ACTIVITIES

The below given tables present the educational specialities and lines available in the Faculty. They are divided into two groups : 10-semester studies for M.Sc. degree and 7-semester studies for B.Sc. degree, both finalized by a diploma thesis.

*Lines of M.Sc. studies*

| Speciality | Line of studies |
|---|---|
| Shipbuilding and Ocean Engineering | • Ship hydromechanics and design<br>• Technology and material science<br>• Structural mechanics and construction |
| Machinery, Power Plants and Deck Equipment | • Ship Power Plants<br>• Automation of Ship Power Plants<br>• Gas and Steam Turbines<br>• Ship Deck Equipment |

*Lines of B.Sc. studies*

| Speciality | Line of studies |
|---|---|
| Ship and Offshore Objects Technology | • Steel structures<br>• Composite structures<br>• Underwater Technology |
| Power and Propulsion Systems | • Design, Construction and Operation of Power Plants<br>• Gas and Steam Turbines |
| Management and Marketing in Maritime Economy | |

Training for industrial personnel is one of the main area of educational activities of the Faculty. For example the Faculty owns the technological licence on "Non-hammering straightening of welded steel & aluminium thin-walled structures". Special application courses for engineers and workers engaged in this area are organised each year.

### OTHER ACTIVITIES

Besides educational and training activities, the Faculty's scientific personnel carries out multilateral cooperation with Polish and foreign universities (more than 70 academic centres), takes part in activities of different international organizations and associations as their members, participates in scientific conferences and symposia.

Moreover, during the last 5 years the Faculty organised or co-organised about 20 international and domestic scientific conferences and symposia.

A usual practice of the FOEST is also hosting the young scientists and students from different countries.

## RESEARCH ACTIVITIES

The Faculty conducts a wide range of research in the area of naval architecture, marine engineering and ship technology.

**The main research areas are outlined below** :

- Computer aided ship design methods, non-linear optimisation methods in waterborne transport and ship design, computational geometry and geometric modelling of ship hull forms, wave resistance theory, ship hull form analysis and optimisation methods, CAD models in design of sailing vessels and yachts, stability safety analysis of sailing vessels.
- Stochastic analysis of ship safety in sea environment for standardisation of ship safety, criteria for standardisation of freeboard on the basis of prescribed ship safety, prediction and analysis of manoeuvring characteristics of ships, numerical simulation of ship manoeuvres, analysis of ports and waterways from the point of view of ship manoeuvring safety, methods for initial ship design from the point of view of survivability, analysis of the risk of ship capsizing due to parametric resonance, experimental analysis of stability of fast planning mono-hulls, design and analysis of ship propulsors.
- Ship mechanics related to a damaged ship, numerical simulation of a damaged ship behaviour in natural conditions, safety of ships in the damage conditions, subdivision and damage stability of Ro-Ro vessels, subdivision of tankers from the point of view of environment protection, stabilisation of ship motion in waves.
- Analysis of ultimate and fatigue strength of ship structural elements under multi-axial deterministic and probabilistic loading, analysis of ship structure vibrations, application of numerical methods including the FEM to ship structures, determination of sea loads on ships and offshore structures, theory and modelling of floating objects as complex dynamic systems, determination of hydrodynamic loading of liquid-filled tanks, analysis and modelling of segmented structures with multi-float support.
- The mechanisms of initiation and propagation of fatigue and brittle failures in ship materials and structures, development and conditions for complex destruction mechanisms of ship structures, methods of analysis of durability of ship structures, optimisation of ship manufacturing processes, methods of automated dynamic quality control of shipyard manufacturing processes, ecological aspects of manufacturing processes in the shipbuilding industry, research on ship repair and conversion methods with accounting for ecological aspects.
- Optimisation of reliability of ship propulsion and other systems, analysis of damages of ship power plant elements, requirements for the power, durability, reliability and safety indices of ship power plants and their models, comparative analysis of classical and novel methods of ship power plant design with the use of durability, reliability and safety probabilistic models, forming of durability, reliability and safety of ship power plants at various design stages, rational procedures of designing the ship power plants for high level of durability, reliability and safety.
- Ship automation, ship power plant control, dynamic diesel turbocharger's systems, steam and gas turbine ship propulsion, dynamics of rotors and gas turbine engines, self–excited vibrations of turbomachine rotors.

**The Faculty is a unique centre in Poland that is suitably equipped for carrying out** :

- full-scale tests of hull structure parts under ultimate load
- research and training in the field of ship manoeuvrability by using various large manned models of ships (in cooperation with the Foundation for Safety of Navigation and Environment Protection in Iława, co-founded by the Faculty)
- educational projects offerred to the European community of shipbuilding students, to be performed in the own research facility located at the Jeziorak Lake in Iława; the scope of investigations covers: manoeuvrability, resistance, ship stability (conducted for the last 4 years with constantly improving results)
- applications of underwater techniques (remotely operated vehicles, techniques for naval operations)
- experiments with the use of an air model turbine rig, and of gas turbine test stand.

## INTERNATIONAL COOPERATION

**The Faculty is a member of the following organizations** :

- European Association of Universities in Marine Technology
- The Association of Polish Maritime Industries
- Association of Shipbuilding Faculties of Nordic Countries.

**Actually, the Faculty participates in the three EU-funded projects, and the Leonardo da Vinci Programme** :

- SANDWICH – Advanced Composite Steel Structures
- HARDER – Harmonisation of Rules and Design Rationale
- EURO-MTEC – European Marine Technology Education Consortium.

Tempus, Copernicus, Phare ACE, Tessa and SOCRATES ERASMUS are other international or EU programmes that the Faculty has already participated in.

**The members of the Faculty staff participate in activities of** :

- ISSC – International Ship and Offshore Structures Congress
- ITTC – International Towing Tank Conference
- PRADS – Practical Design of Ship Structures
- STG – Schiffbautechnische Gesellschaft
- GL Poland – Polish Committee of the Germanischer Lloyd
- DNV – International Committee of Det Norske Veritas
- IMO – Committees of the International Maritime Organization
- PAN – Polish Academy of Sciences (7 Committees and Sections)

**Representatives of the Faculty also act as the members of 7 Scientific Societies and Technical Councils of different Polish organizations, as well as of** :

- European Association of Reliability
- ASME ( USA)
- Association Internationale Cybernetique (Belgium)
- UNESCO - UNISPAR – (University-Industry-Science-Partnership United Nations Educational, Scientific and Cultural Organization).