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# POLISH MARITIME RESEARCH

**Special issue**, 2004


## CONTENTS




**The papers published in this issue have been reviewed by** :
*Prof. A. Brandowski ; Prof. J. W. Doerffer*
*Assoc. Prof. M. Dzida ; Prof. J. Girtler*
*Assoc. Prof. K. Kosowski ; Prof. M. Krężelewski*
*Prof. K. Rosochowicz ; Prof. J. Szantyr*

# Editorial

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

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

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

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

# Editorial Board

# Editor's message

*This is a special issue of Polish Maritime Research quarterly, devoted to results of the research on design, manufacture and operation of a new generation of ships for Baltic Sea shipping. The research was realized in the frame of the EU-supported project aimed at creating multi-mode shipping routes in that region, and satisfying the conceptual assumptions of Baltic Sea status as a Sensitive Sea Area. For that reason great importance has been attached to ecological problems in this project. We hope that this initiative of the Editors and the Prinicpal Coordinator of the project will be met with kind acceptance.*

**Editor-in-Chief**

# NEW GENERATION OF THE BALTIC ECOLOGICAL SHIPS IN THE EUROPEAN EUREKA RESEARCH PROJECTS

## Project E!2772
## *Baltecologicalship*

### Chief executor and coordinator of the whole project :

***Gdańsk University of Technology***
***Faculty of Ocean Engineering and Ship Technology***

**Gdańsk 2004**


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WARSZAWA

# Environment friendly ships for Baltic area

**Krzysztof Rosochowicz**, Prof., D.Sc.
(Chief coordinator of the E!2772 project)

**Tomasz Łącki**, M.Sc., Eng.
(Leader of the design office)
SINUS Ltd. Co.

ABSTRACT

*The paper outlines the origin of the E.2772! project from the EUREKA group of European projects. Justification of undertaking the subject is given as well as the structure of institutional performers constituting the project executive consortium. Characterised is the final result in the form of preliminary designs of four modern ecological Baltic short shipping vessels: container carrier, product tanker, ro-ro ship and river-sea type ship. Comprehensive bibliography contains a list of all the project reports, which may be made available to the interested parties after a contact with the project coordinator. The paper opens a series of topical papers selected from the different problems dealt with in the project.*

**Keywords**: EUREKA project Baltecologicalship, project execution, preliminary designs of four ships

## INTRODUCTION

The structural changes and enormous acceleration of the world economy cause a necessity of developing new generations of surface transport, including sea and inland waterway shipping. This applies to transportation of the investment and consumer goods as well as to passenger traffic. In the highly developed areas, strongly urbanized and inhabited by wealthier societies, one can hardly imagine further development of the traditional transport of goods through the road or even railway network, to meet the growing demand. With cautious forecasts of the increase of freight transport in the EU territory by approx. 38% by 2010, concentrated in the main land transport corridors and around large urban areas, activation of the sea and inland water transport incorporated into multimodal transport networks becomes a problem of vital importance.

The emphasis put on the development of sea transport has been for the last several years a clear policy of the European Union, where surface transport is to be divided into long distance transport between established and intensively developed central reloading nodes and short distance feeder transport. In the case of sea transport, the latter is defined as short shipping. This problem is also characteristic of the Baltic, which has become practically an EU *Mare Nostrum.*

The amount of cargoes transported annually in the Baltic area reaches 450 million tons. Approximately 5500 thousand trips are performed served by the 500 Baltic ports. In recent years, the annual increase of the volume of transported goods has been as big as 30%. It is estimated that some 5000 ships are operated in the Baltic, 40% of them are older than 20 years and will have to be replaced by new generation ships in the nearest future.

The Baltic short shipping development trends include mainly container carriers (300% increase of the volume of transport in 10 years), passenger ferries and product tankers. The actual structure and size of that new generation fleet will depend mainly on the long term development forecasts of the volume and type of cargo and the passenger tourist traffic.

Regardless of location of the Baltic multimodal transport corridors, ecological requirements will have to be met connected with new conception of the Baltic Sea status as PARTICULARLY SENSITIVE SEA AREA, to be implemented by 2005. That involves a necessity of creating a fleet of specialised new generation Baltic feeder service ships fulfilling strict ecological criteria. Designs of such ships, based not only on the verified design practice but also on the state-of-the-art knowledge, should be on offer in the shipyards, and shipping companies aspiring to participation in the European transport system should provide for operating such ships.

## PRO-ECOLOGICAL REQUIREMENTS

Regardless of the type of ship, the following general pro-ecological requirements have to be met :

➲ Significant reduction of:
 ⇨ all kinds of emissions and pollution connected with ship operation
 ⇨ effects of possible sea averages and disasters
 ⇨ negative effects of the solid waste utilization processes
 ⇨ effects of vibration and noise
 ⇨ negative effects of ship construction and repair processes.

➲ Improved safety and reliability of ship operation (structure and systems).

➲ Application of additional technologies, equipment and systems for proper utilization of the operational waste products.

Sources of hazards to the sea environment from an operating ship are schematically presented in Fig.1.

Meeting all the requirements of the new generation Baltic ships will certainly cause higher construction costs but it is estimated that they will be compensated for by e.g. lower insurance premiums, lower port charges and smaller hazards to the marine environment.

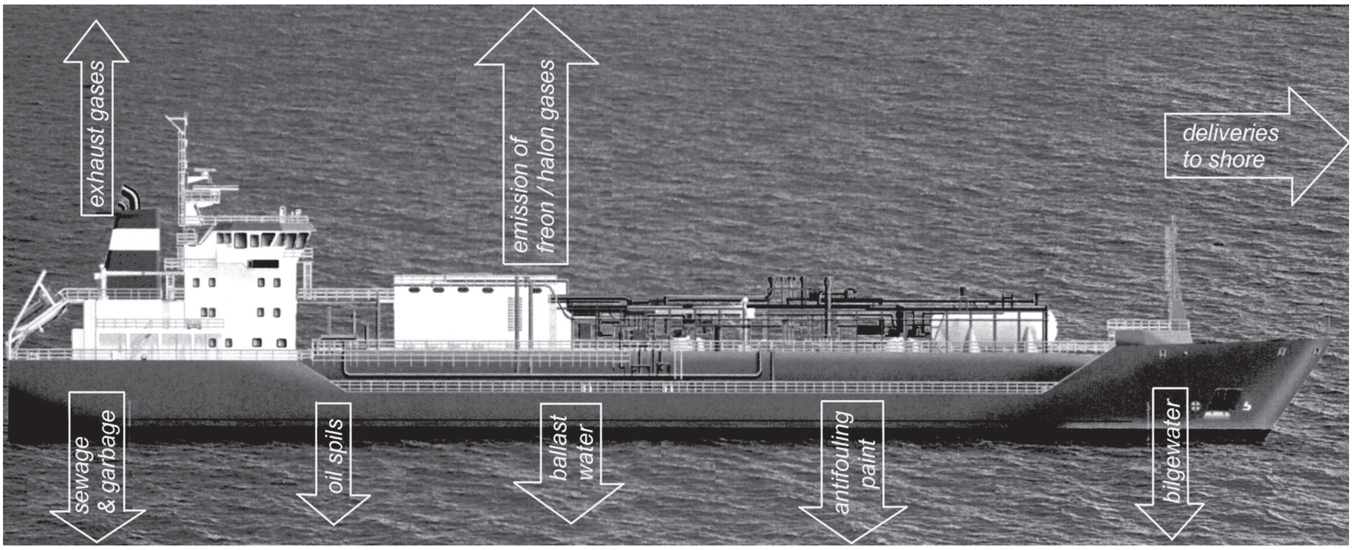



*Fig. 1. Ecological hazards created by ship (acc. to DNV – Maritime Solutions – Clean Performance advertising materials)*

## PROJECT EXECUTION

In order to perform the task of establishing the Baltic ecological multimodal transport corridors with new generation ships, a European project E!2772 Baltecologicalship was set up in the EUREKA group and a consortium was formed of the following participants:

### *Technical universities*

- Gdańsk University of Technology, Faculty of Ocean Engineering and Ship Technology – chief executor and coordinator of the whole project; Faculty of Chemistry as responsible for preparations of selected shipbuilding sector companies for the ISO 9000 and 14000 certification
- Szczecin University of Technology, Faculty of Maritime Technology.

### *Companies from the small and medium enterprise (SME) sector*

- SINUS Limited Company, performing design work on the conceptions of 4 types of Baltic ships
- Toka Consulting Company, cooperating in the ISO reviews
- Ship repair yard, which unfortunately withdrew from the project at an early stage.

### *Foreign partners*

- Boy&Partners-Loma Company, Sweden, co-performing the ISO reviews
- Björn Carlsson Ecoship Engineering AB, Malmö, Sweden, performing own design work on the product tanker conception.

The consortium performed the work in two stages.

At the definition stage, studies were undertaken on the needed structure of the Baltic region sea transport as well as design conceptions and requirements of the ships.

At the design and testing stage, designs of the ships were carried out and, as they were being completed, laboratory tests and analyses of hull shapes were performed and also verifying calculations and analyses of machinery, equipment and system solutions were carried out. In effect, the designs were corrected as appropriate and submitted to final evaluation. Parallel work was carried out to recognize the production capabilities of selected Polish shipyards. An advanced modernization conception of a selected shipyard was developed to ensure effective realisation of the contract.

The project participants, both the university teams and the small and medium enterprise (SME) sector companies, created an efficient and well functioning organism which has generated up-to-date designs forming a prospective development offer of the shipbuilding sector in its aspirations for the European operating structural funds.

At the same time, as a consequence of activities in the Swedish branch of the project, in cooperation with Polish specialist groups, 40 shipbuilding sector enterprises, including two medium size shipyards, have been prepared for beginning the ISO 9000 and 14000 standard certification process. The Swedish conception of a product tanker has obtained approval of a Swedish owner and after many rounds of negotiations in different shipyards, final contract talks are in progress in two selected shipyards.

## THE SINE SERIES SHIPS

The project comprised the following ship types :

- SINE 202 universal container carrier
- SINE 203 oil product tanker
- SINE 204 ro-ro ship
- SINE 205 river-sea ship.

The designed ships had to fulfil the Det Norske Veritas (DNV) CLEAN class conditions, the first in the world complex set of classification requirements to guarantee ecological cleanness of a ship. A range of different solutions have been applied in the designed ships in order to meet the requirements.

In general, the solutions comprise :

- In connection with the increased safety requirements of ship in operation and in case of disastrous collision, the following solutions have been applied :
  - diesel-electric propulsion system (better use of fuel, minimum engine starting and stoppage time, significant limitation of water pollution by oil from the propulsion system)
  - pod propulsion and steering system (effectiveness increased by at least 10%, better manoeuvrability, less vibration and noise, elimination of rudder, minimum ship backing time, elimination of the cooling system)

- ✰ two separate power plants (breakdown safety of propulsion and hull)
- ✰ double shell in the fuel tank and hold area (elimination of spills).

❖ In connection with restrictions on the emissions of gases to atmosphere and of the operational and technological media to water, the following solutions have been applied :

- ✰ use of low sulphur content fuels
- ✰ use of exhaust-gas catalyzers (limitation of $NO_x$ and $SO_x$)
- ✰ design constraints on oil spills and cooling medium spills, elimination of freon
- ✰ minimalization of ballast water exchanges in order to avoid the hazards from propagation of microflora and microfauna
- ✰ use of exclusively natural media in the fire fighting systems
- ✰ limitation of the oil content in bilge waters to a 15 ppm level with an automatic STOP device switching the system off when the value is exceeded
- ✰ use of ecologically safe internal and external ship paints
- ✰ use of special ship sewage treatment procedures.

❖ In connection with solid waste treatment aboard ship, to be transferred to shore for utilisation, waste sorting machines have been applied.

Main parameters of the designed ships are listed in the Table.

Fig. 2 presents general arrangement plan and 3D visualisation of the ro-ro ship stern showing elements of the systems necessary to meet the CLEAN class requirements. Figs. 3, 4 and 5 present the product tanker, container carrier and river-sea ship, respectively. The ship documentation has been prepared to the level of preliminary design of the feeder service means of transport.

## SUMMARY

- ➢ The Eureka group European projects are an element of system support to the development of complex innovative products of small and medium enterprises obtaining their knowledge from the resources of scientific and research institutions. They create one of the mechanisms of organizing a European research and development network.
- ➢ The project execution consortium, sponsored by the Committee for Scientific Research on the Polish side and the VIANOVA and NUTEK government agencies on the Swedish side, brought together a research potential of 2 technical universities (3 faculties), many small and medium enterprise sector companies from the Polish side and 2 companies of that sector from the Swedish side and effectively combined science, knowledge, theory and design practice in realisation of a complex innovative task.
- ➢ The cooperation proved effective. Four ship designs have been created on the Polish side and three designs (product tanker) on the Swedish side. Forty Polish shipbuilding sector enterprises have been prepared for the ISO 9000 and 14000 standard series certification and also talks are in progress about an order of 4 product tankers for a Swedish owner. These are immediate effects of the cooperation.
- ➢ The obtained and processed knowledge – a basis of the design work – has been gathered together in 159 reports and other source materials, which will also be used for preparing a monograph publication (see bibliography).
- ➢ The product tanker design, selected with the use of synthesis of knowledge from earlier European projects, has been introduced as a subject of continued studies and analyses to the next Eureka group projects (ASPIS).
- ➢ Selected interesting topics of the analyses (in hydromechanics, design, structural design, ship systems, economy, production engineering) performed in the project will be published later on in this journal.

***Table***

| **Parameters** | **Type of ship** | | | |
|---|---|---|---|---|
| | **SINE 204 Ro-ro ship** | **SINE 203 Product tanker** | **SINE 202 Container carrier** | **SINE 205 River-sea ship** |
| $L_{oa}$ [m] | 150.76 | 138.10 | 138.1 | 89.45 |
| $L_{pp}$ [m] | 147.75 | 132.0 | 132.0 | 87.47 |
| B [m] | 24.80 | 22.50 | 22.50 | 11.40 |
| H to main deck [m] | to deck 4 19.60 | 12.80 | to main deck 11.20 | 5.45 |
| $T_{des.}$[m] | 6.00 | 8.00 | 7.60 | 4.40/2.80 |
| T [m] | 6.50 | 8.70 | 8.55 | |
| DWT | 7400 | 12800 | 8550 | 2900 |
| Speed [kn] | 20.0 | 14.0 | 18.05 | 12.0/14 |
| Range [n. miles] | 8000 | 6000 | 10000 | 2500 |
| Power [kW] | 19500 | 5980 | 11200 | 2550 |
| Generator [kW/V] | 690 | 1120/660 | 1120/660 | 2x1120/690 |
| Propulsion [kW] | pod 2x6500 | pod 2x2200 | pod 2x4350 | pod 2x700 |
| Prop. Ø [m] | 3.8 | 3.2 | 3.2 | 1.7 |
| Pod weight [t] | 42 | | 47 | 25 |
| Load | 7200 $m^2$ – trailer deck | 16900 $m^3$ | 950 TEU or 475 FE4 | TEU 20 Holds – 54 Deck – 36 385 $m^3$ |
| DNV Class | +1A1, Ro-Ro, Cargo Vessel, E1 | +1A1 Tanker for Oil Products, ESP, EO ETC, ICE-!C, RPS, ICS, W1, CCO, LCS (SID) | +1A1 Container Carrier, EO, W1, S.C., RP | +1A1 General Cargo Carrier HC, EO, ICE-1C, RP |

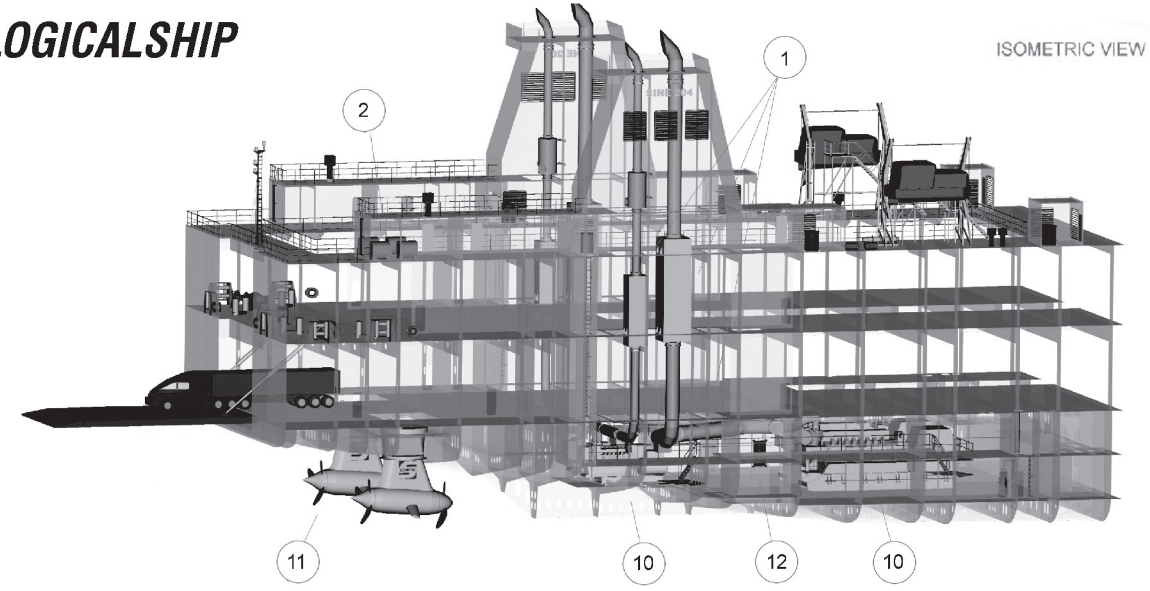
# Project E!2772 BALTECOLOGICALSHIP

EUREKA

## ENVIRONMENT FRIENDLY SHIP

FOR EXAMPLE RO-RO CARGO VESSEL "SINE 204"

1. Reduction of the NOx & SOx emission to the atmospherefrom combustion machinery exhaust gases.
2. Garbage sorting system.
3. Not any ozone depleting substances are used as refrigerants.
4. Only natural substances are used as a medium in fixed fire fighting systems.
5. The active ingredients in the antifouling are of a non-TBT tipe.
6. Discharge of ballast water are restricted to minimum in order to prevent transport of harmful aquatic organisms and pathogens.
7. Bilge water system is fitted out in oil filtering equipment and 15 ppm alarm combinet with an automatic stopping device.
8. Sewage treatment and discharge are carried out acc. to approved sewage treatment.
9. Fuel tanks with double skin construction.
10. Diesel-electric propulsion system.
11. Podded drive system.
12. Propulsion redundancy system achieved by two separate engine rooms.



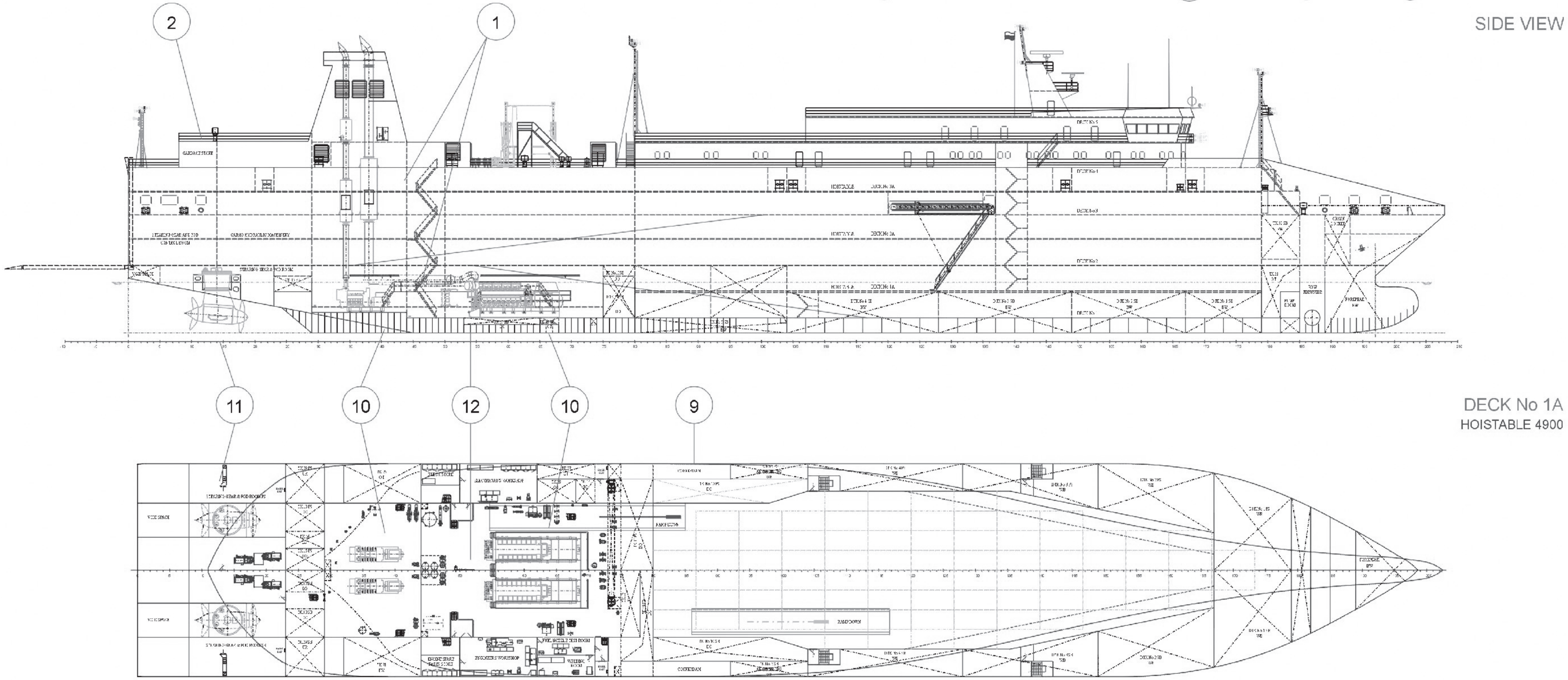




**Fig. 2.** *Ro-Ro ship (SINE 204)*

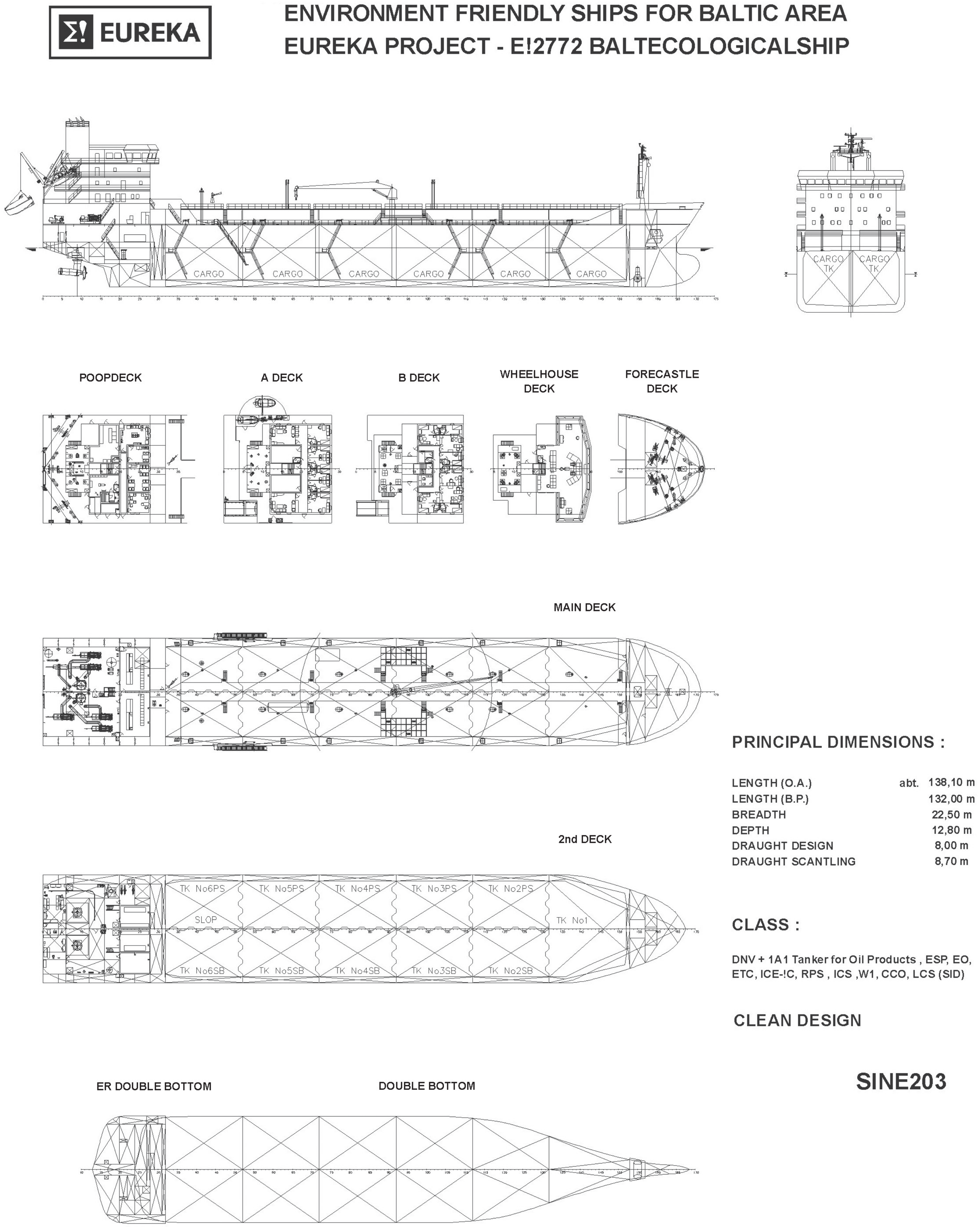
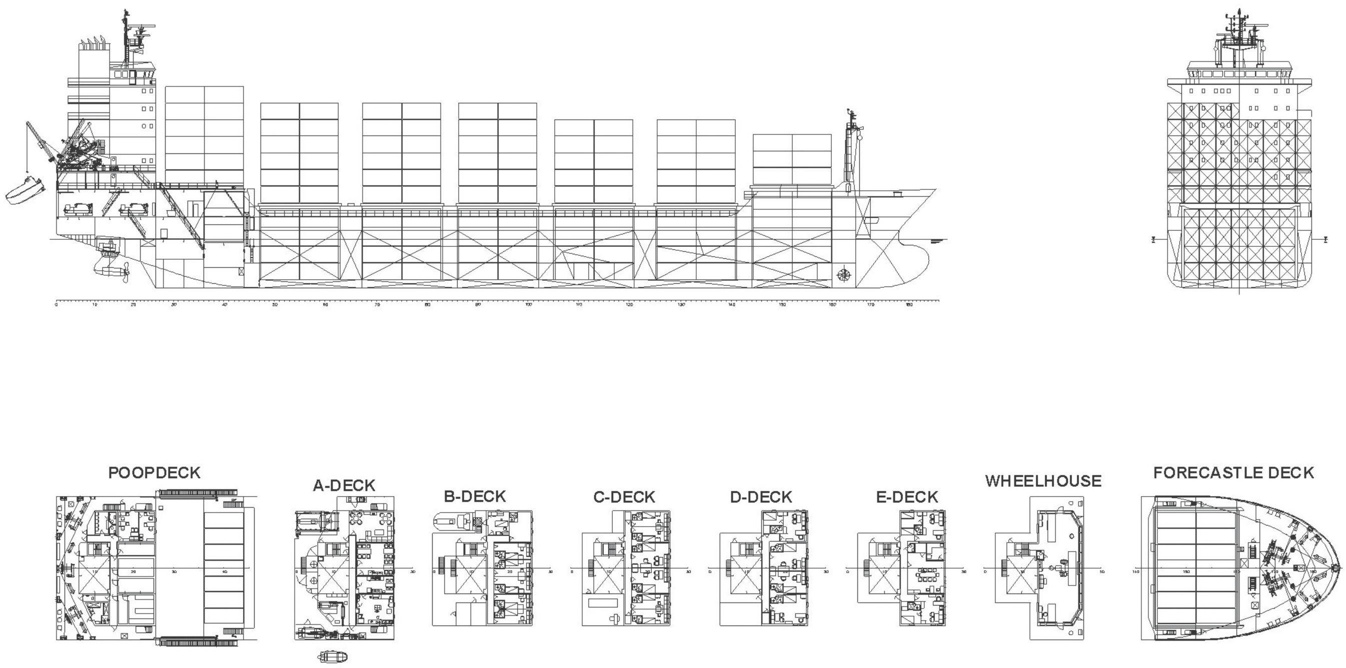
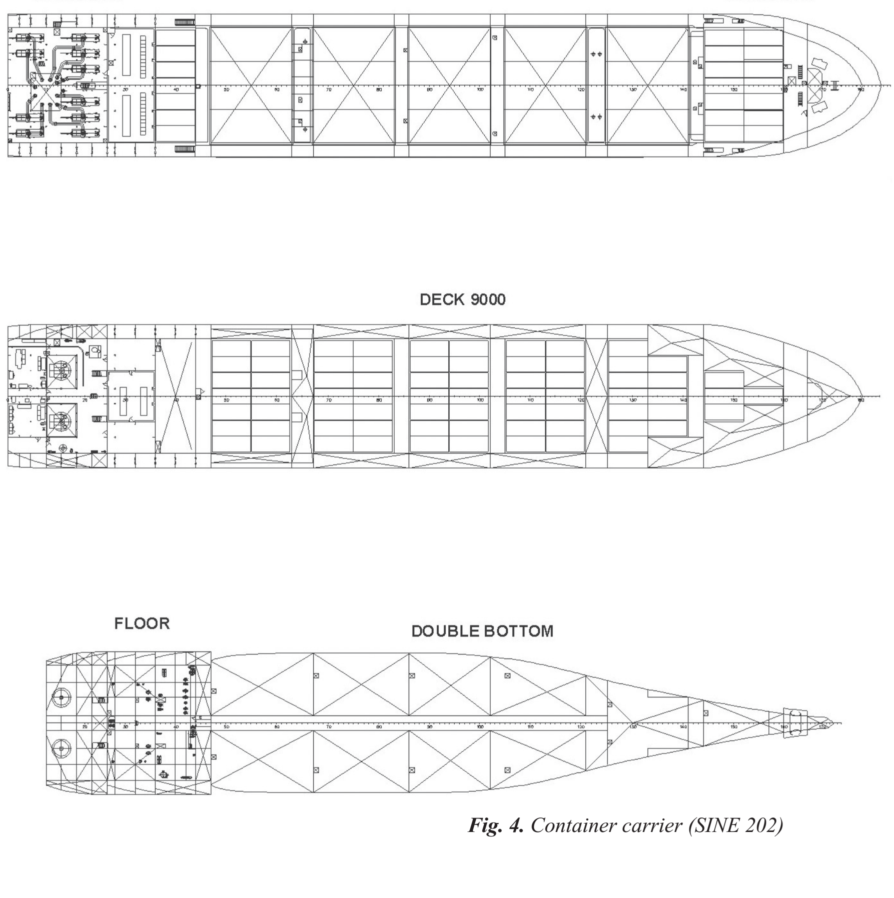



***Fig. 3.*** *Product tanker (SINE 203)*

## ENVIRONMENT FRIENDLY SHIPS FOR BALTIC AREA
## EUREKA PROJECT - E!2772 BALTECOLOGICALSHIP




### PRINCIPAL DIMENSIONS :

| | | |
|---|---|---|
| Length overall | Loa = | ~ 138,10 m |
| Length between perpend. | Lbp = | 132,00 m |
| Breadth moulded | B = | 22,50 m |
| Depth | H = | 11,20 m |
| Scantling draught | Ts = | 8,55 m |
| Design draught | Td = | 7,60 m |
| Service speed | V = | 18,5 kn |
| Deadweight at 7,60 m draught | | ab.8550 DWT |
| CREW | | 14 + 1 |

### CLASS :

DNV + 1A1 Container Carrier, EO, W1, SC, RP

### CLEAN DESIGN

**SINE202**

*Fig. 4. Container carrier (SINE 202)*

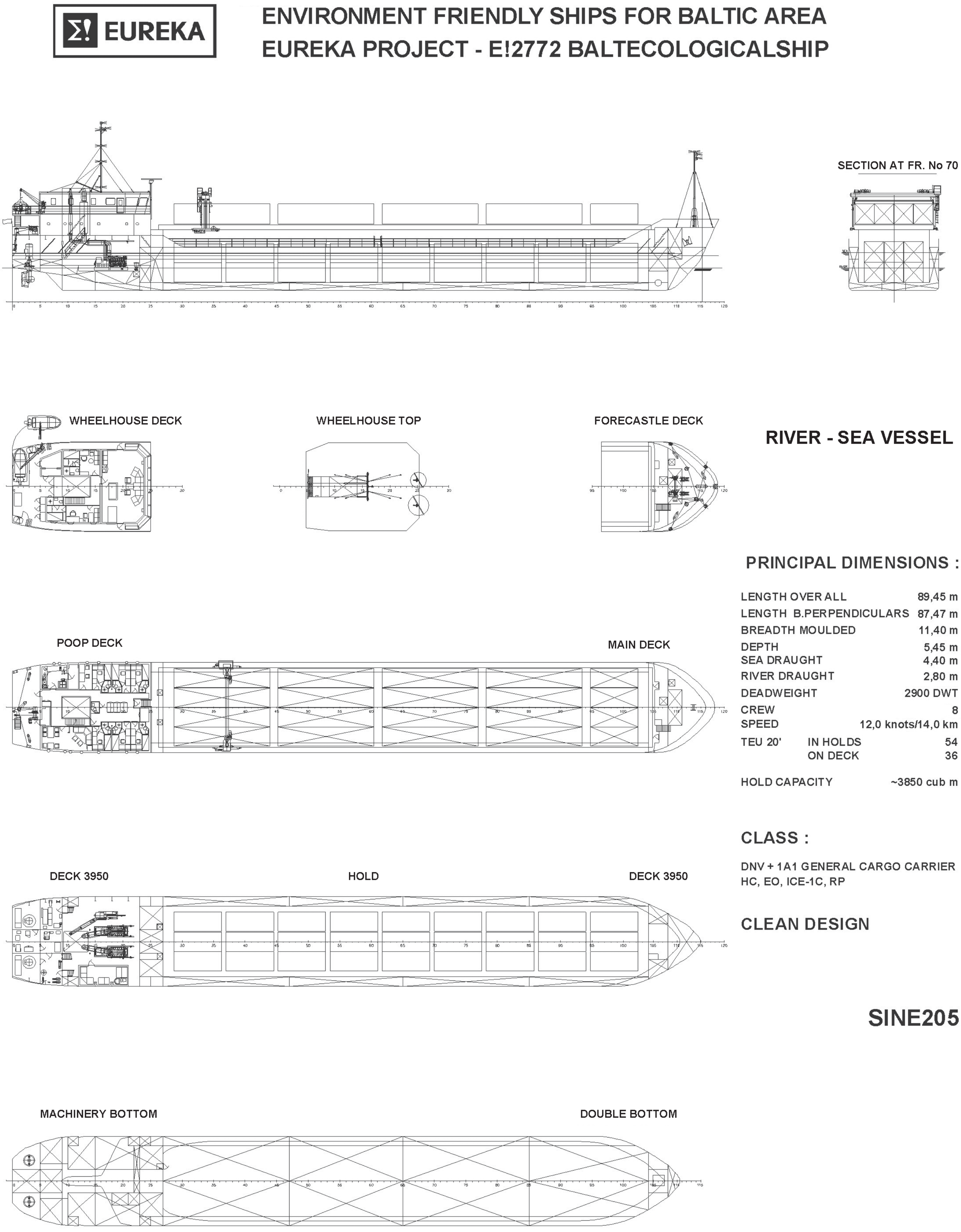



*Fig. 5.* *River-sea ship (SINE 205)*

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# Parametric method of preliminary prediction of the ship building costs

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ABSTRACT

*Paper presents results of studies on a parametric method of predicting ship building costs – – useful in the preliminary design. Conception and theoretical basis of the method are presented, devised are also approximation formulae for estimating the building costs of the ship hull, ship equipment and the power plant with propulsion system. Factors of importance for the ship building costs are identified and a computational algorithm formulated. The useful character of the method is illustrated by examples of building cost predictions for four different ship types designed in the Eureka project E!2772, i.e.: SINE 202 universal container carrier, SINE 203 oil product tanker, SINE 204 ro-ro ship and SINE 205 river-sea ship.*

**Keywords**: ships design theory, ship building costs

## INTRODUCTION

At the initial stages of designing a ship, its parameters are determined by the ship design theory methods allowing to find design solutions fulfilling the owner design requirements as well as the valid international conventions, e.g. freeboard rules or the ship damage stability requirements. In the optimum design methodology it is also necessary to formulate the designed ship evaluation criterion.

The criterion measure should be a ship property – of a measurable economic effectiveness character – important for the owner's operations on the market and containing, in a direct or indirect form, ship investment costs, predicted operating costs, evaluation of the ship transporting and money earning capability as well as the investment and acting capital costs. The ship evaluation criterion measure must be expressible by a set of ship technical parameters $\bar{\mathbf{x}}$ determined in the ship design process - decision variables of the design optimization mathematical model. Optimization models in the methods of ship design theory may be particularly useful in the work of ship design offices at the time of stronger competition on the world shipbuilding and shipping markets.

This paper contains results of research studies on a parametric method of preliminary prediction of the ship building costs, based on a small set of its main design parameters. The developed method of ship building cost prediction (a top-down method) may be used as an evaluation criterion in the preliminary ship design optimization models. The presented method is a contribution to the development of ship design theory as well as a tool to be applied in practical ship design.

The problem of methodology of ship building cost prediction at the preliminary design stage has been a subject of many publications, both those already classic, like the works of Benford [1, 2, 3], Sójka [4], Buxton [5], Fetchko [6], Dart [7] and Fisher [8] and the more recent items, e.g. [9, 10, 11, 12, 13].

## AIM AND SCOPE OF THE WORK

Collecting data on **real costs** of building ships is a difficult task – in particular in relation to the ships currently under construction – as the data are treated as confidential business information of shipyards and owners. For evaluation of the economic value of a designed ship important are, apart from the production costs, also the predicted future operating costs, availability of cargoes, freight rates etc. in an appropriate time period, e.g. 15 to 30 years of the expected ship operation. Acquiring a reliable prediction tool is a significant difficulty in the research work on the development of preliminary ship design optimization methods.

The ship building and operating cost data are scarce in the literature, usually they are related to different time periods and are presented in a form of diagrams or tables of little use to the optimization method computational algorithms.

The presented research was initially aimed at results consisting in the development of ship design theory through devising an original parametric method of ship building cost evaluation at the preliminary design stage, a method of a mathematical structure appropriate for the computer optimization algorithms.

A useful effect of the research was demonstrated by using the method to perform the building and operating cost predictions for ships designed in the EUREKA project E!2772.

## REQUIREMENTS OF THE METHOD

In the early phases of preliminary ship design the building cost may be predicted by means of parametric relations corresponding with the level of the identified technical parameters. At that stage no data are available to predict the cost from the equipment and material pricelists or from the maker's proposals. The parameters in question are main dimensions, component weight estimates, speed, propulsion power etc.. The method is based on the available data related to ship parameters and building costs as well as information on the cost structure.

In view of the non-homogeneity of data, as they come from different sources and are differentiated as regards :

- territory – they pertain to shipyards in different countries, operating in different economic systems
- monetary system – they are expressed in different currencies
- time – they pertain to different periods in the past.

The costs have been reduced to comparable values – different currencies were converted to US dollars at the rates valid in the respective periods and then updated to the current value using an average US inflation rate (in the years 1990÷2003) of 3% (according to the Bank of America indices).

It has been assumed that the total ship building cost including :

- material cost
- labour cost
- other shipyard costs

consists of :

- hull construction
- ship equipment
- power plant and propulsion system.

It has been also assumed that costs of each of these groups are related to the weight of a respective group. Approximating functions were determined from the collected data to express the unit cost in US dollars per ton of a group weight. The power plant horse-power was converted to an equivalent power plant weight. An advantage of such approach is balancing the designed ship displacement with the sum of component weights, e.g. by the Normand method, already in the preliminary stages of the design process.

Depending on a functional type of the ship, the hull cost is considered for two cases, as in [11] :

- single-deck hulls, e.g. tankers or bulk carriers
- multi-deck hulls, with well developed internal volume subdivision, e.g. passenger ships, car carriers or ro-ro ships.

The ship equipment cost is considered in three classes, differing in respect of the equipment quality and of saturation of the ship with equipment. The equipment unit prices and the assembly labour cost may differ considerably. The equipment class variant depends on the owner requirements as well as on :

- functional type of the ship (e.g. classic general cargo ships usually have low equipment class whereas passenger ships have high equipment class, etc.)
- necessity of installing special equipment, e.g. refrigerating or air conditioning plant, according to owner's requirements.

The power plant cost predicting formulae are related to diesel engine plants, where :

- lower range of the main engine weight (power) corresponds to medium-speed engine power plants
- upper range of the main engine weight (power) corresponds to low-speed engine power plants.

Cost shares of the individual ship building stages in the total cost, according to an empirical estimation quoted in [14], are the following:

| Building stage | Cost of the stage | Impact on total building costs |
|---|---|---|
| Preliminary design | 3% | 60% |
| Other design stages | 7% | 25% |
| Ship production | 90% | 15% |

The estimation shows that the design stage, having itself approx. 10% share in the total ship building costs, determines 85% of those costs. Expenses on the design quality – proper choice of the ship main parameters, production technology, structural materials, equipment types etc. – have a significant impact both on the shipyard's and owner's economic effects.

## UNIT COSTS

Preliminary studies have shown that analytical relations between a unit cost $\mathbf{q_j}$ of a ship technological group and its weight $\mathbf{m_j}$ may be approximated with sufficient accuracy by means of power functions containing four constant structural coefficients $\mathbf{c_{i,j}}$ determined by the least squares method. An identical structure of the unit cost and weight binding formula has been assumed for all the technological groups :

$$q_j = c_{0,j} + c_{1,j} \cdot m_j + c_{2,j} \cdot m_j^{c_{3,j}} \tag{1}$$

The unit cost determined by this formula comprises:

- material costs
- labour costs
- other shipyard costs.

Structure of the formula was determined from tests performed with different test functions, using the following selection criteria:

- good approximation quality in the considered range of design parameters
- easy determination of numerical values of the formula as well as derivative relations, which is an important advantage when manual calculations have to be performed, e.g. with a calculator, and also when an optimum design mathematical model is formulated (a problem of accurate determination of the Jacobian or Hessian determinant).

The structural coefficients of approximation formulae, given in Table 1, have been determined by means of the regression analysis.

***Table 1.*** *Values of structural coefficients of the ship building cost predicting formulae*

| j | Weight group | Symbol | Unit cost | $c_{0,j}$ | $c_{1,j}$ | $c_{2,j}$ | $c_{3,j}$ |
|---|---|---|---|---|---|---|---|
| | | [t] | [$/t] | [-] | [-] | [-] | [-] |
| 1 | Single deck hull | $m_h$ | $q_s$ | 8056.4 | 0.031 | -936.42 | 0.1949 |
| 2 | Multi deck hull | $m_h$ | $q_m$ | 16023.8 | 0.053 | -4927.0 | 0.1084 |
| 3 | Low standard equipment | $m_e$ | $q_l$ | 35512.9 | -1.175 | -12957.0 | 0.0518 |
| 4 | Average standard equipment | $m_e$ | $q_a$ | 23196.6 | 61.590 | -69.62 | 0.9903 |
| 5 | High standard equipment | $m_e$ | $q_h$ | 29751.0 | 33.006 | -63.37 | 0.9344 |
| 6 | Power plant | $m_p$ | $q_p$ | 25230.2 | 2.928 | -161.66 | 0.63435 |

The preliminary ship building cost predictions, confronted with actual market prices (as published by the Centromor foreign trade agency in the „Budownictwo Okrętowe" monthly), were evidently on the high side. That may be attributed to the fact that prices for some types of ships have decreased during the last decade not only in a relative sense (allowing for inflation) but also in an absolute sense, as an effect of sharp competition on the shipbuilding market and dumping prices quoted by the Far East shipyards.

Correction of the predictions was performed by calibration of the formulae with the use of the mentioned published actual prices. New corrected structural constants of the approximation formulae are given in Table 2.

***Table 2.*** *Corrected structural coefficients of the ship building cost predicting formulae*

| j | Weight group | Symbol | Unit cost | $c_{0,j}$ | $c_{1,j}$ | $c_{2,j}$ | $c_{3,j}$ |
|---|---|---|---|---|---|---|---|
| | | [t] | [$/t] | [-] | [-] | [-] | [-] |
| 1 | Single deck hull | $m_h$ | $q_s$ | 1994.663 | 0.015549 | -154.0222 | 0.2471932 |
| 2 | Multi deck hull | $m_h$ | $q_m$ | 3241.7105 | 0.0194715 | -592.6707 | 0.1637139 |
| 3 | Low standard equipment | $m_e$ | $q_l$ | 6528.5325 | 9.6026706 | -12.56288 | 0.979517 |
| 4 | Average standard equipment | $m_e$ | $q_a$ | -2599.825 | -1.395667 | 9146.288 | 0.0213938 |
| 5 | High standard eqipment | $m_e$ | $q_h$ | 9749.0427 | 14.66748 | -16.71265 | 0.9963722 |
| 6 | Power plant | $m_p$ | $q_p$ | 16720.374 | 0.7839685 | -221.3641 | 0.510682 |

The unit construction cost $q_s$ [$/t] of a single-deck hull of weight $m_h \in (1000 \div 30000)$ [t] is the following :

$$q_s = c_{0,1} + c_{1,1} \cdot m_h + c_{2,1} \cdot m_h^{c_{3,1}} \quad (2)$$

and that of a multi-deck hull of weight $m_h \in (1000 \div 30000)$ [t] :

$$q_m = c_{0,2} + c_{1,2} \cdot m_h + c_{2,2} \cdot m_h^{c_{3,2}} \quad (3)$$

The unit cost $q_e$ [$/t] of ship equipment of weight $m_e \in (100 \div 3000)$ [t], depending on the equipment standard :

*low standard equipment* :

$$q_l = c_{0,3} + c_{1,3} \cdot m_e + c_{2,3} \cdot m_e^{c_{3,3}} \quad (4)$$

*average standard equipment* :

$$q_a = c_{0,4} + c_{1,4} \cdot m_e + c_{2,4} \cdot m_e^{c_{3,4}} \quad (5)$$

*high standard equipment* :

$$q_h = c_{0,5} + c_{1,5} \cdot m_e + c_{2,5} \cdot m_e^{c_{3,5}} \quad (6)$$

The unit cost $q_p$ [$/t] of the power plant of weight $m_p \in (100 \div 2500)$ [t] :

$$q_p = c_{0,6} + c_{1,6} \cdot m_p + c_{2,6} \cdot m_p^{c_{3,6}} \quad (7)$$

The formula validity ranges correspond to the statistical samples used for their approximation. As the unit cost functions are smooth, a moderate extrapolation is admissible.

## THE SHIP BUILDING COST PREDICTION ALGORITHM

In the preliminary ship design stages, the component weight groups are usually determined by methods based on the parent ship data, e.g. by the Normand or Bubnov method or from the empirical and statistical relations, e.g. those given by Watson [11]. In both cases the component weights are expressed by main parameters $\overline{x} = (x_1, x_2, ..., x_n)$ of the designed ship in functional analytical (modular) formulae :

$$m_h = m_h(\overline{x}) \; ; \; m_e = m_e(\overline{x}) \; ; \; m_p = m_p(\overline{x}) \quad (8)$$

so selected that they approximate the actual relations in the best possible way. The modular formulae determine the weight increases due to the changes of main design parameters of the ship, described by the $\overline{x} = (x_1, x_2, ..., x_n)$ vector, in relation to the parent ship parameters $\overline{x}_0 = (x_{01}, x_{02}, ..., x_{0n})$, which in the domain of linear relations, for the $\mathbf{m_j}$ weight group, is described by the expression :

$$m_j(\overline{x}) = m_j(\overline{x}_0) + \Delta m_j(\overline{x}) = m_j(\overline{x}_0) + \\ + \sum_{i=1}^{n} \frac{\partial m_j(\overline{x})}{\partial x_i} \Bigg|_{\overline{x}=\overline{x}_0} (x_i - x_{0i}) \quad (9)$$

The individual group costs are described by an expression of the following structure :

$$Q_j(\overline{x}) = q_j(\overline{x}) \cdot m_j(\overline{x}) \quad (10)$$

The total ship building cost $\mathbf{Q_t}$, depending on the functional type, saturation with equipment and equipment quality, may be expressed as a sum of respective components :

$$Q_t(\overline{x}) = Q_{s,m}(\overline{x}) + Q_{l,a,h}(\overline{x}) + Q_p(\overline{x}) \quad (11)$$

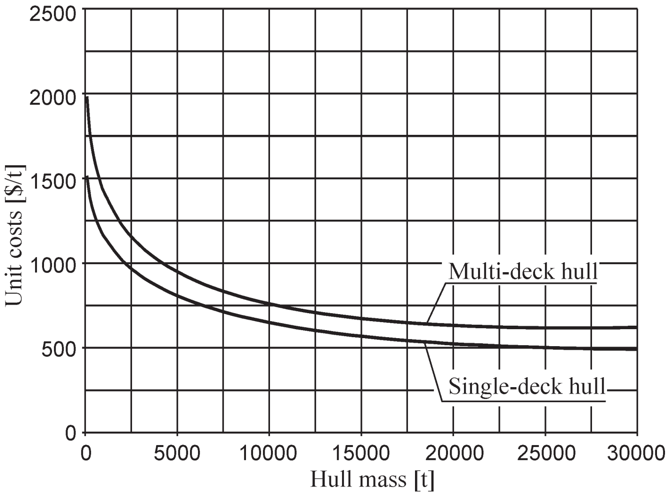


***Fig. 1.*** *Ship hull construction costs*

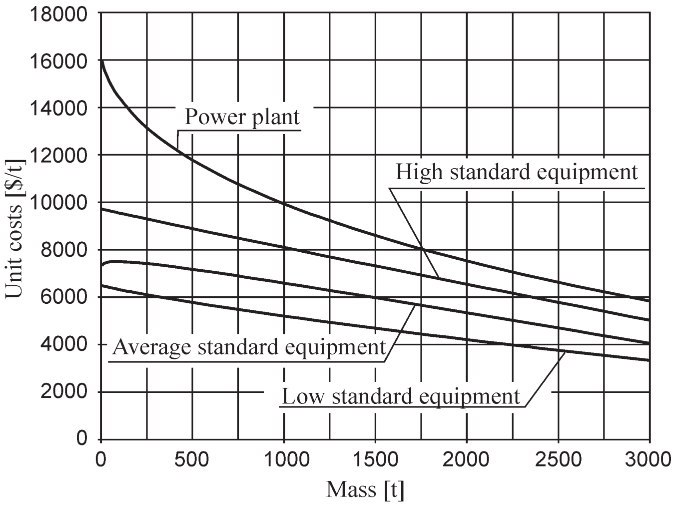


***Fig. 2.*** *Ship equipment and power plant costs*

***Table 3.*** *Main design parameters of ships and predicted building costs in US dollars*

| **Building cost prediction simulation parameters for the baltecologicalships project ships** | **Unit** | **SINE 202 Container carrier** | **SINE 203 Product tanker** | **SINE 204 Ro-ro ship** | **SINE 205 River-sea ship** |
|---|---|---|---|---|---|
| Deadweight | [t] | 8 550 | 14 300 | 7 400 | 2 950 |
| Displacement | [t] | 15 600 | 19 922 | 15 850 | 3 938 |
| Length over all | [m] | 138.70 | 138.10 | 156.70 | 89.45 |
| Length between pp | [m] | 132.00 | 132.00 | 147.75 | 87.47 |
| Breadth moulded | [m] | 22.50 | 22.50 | 24.80 | 11.40 |
| Depth to maindeck | [m] | 11.20 | 12.80 | 19.60 | 5.45 |
| Design draught | [m] | 7.60 | 8.00 | 6.00 | 4.40 |
| Scantling draught | [m] | 8.55 | 8.70 | 6.50 | 2.80 |
| Speed at design draught | [kn] | 18.5 | 14.0 | 20.00 | 12.00 |
| Power at 1800 rpm | [kW] | 11 200 | 10 000 | 19 520 | 3 640 |
| Range of operation | [nm] | 10 000 | 6 000 | 8 000 | 4 000 |
| Number of decks : 1 − single-deck ; 2 − multi-deck | [-] | 1 | 1 | 2 | 1 |
| Equipment standard : (1 − low, 2 − average, 3 − high) | [-] | 1 | 1 | 2 | 1 |
| Hull weight | [t] | 2 855 | 3 530 | 6 050 | 710 |
| Equipment weight | [t] | 900 | 877 | 1 760 | 198 |
| Power plant weight | [t] | 430 | 343 | 697 | 100 |
| Predicted ship building cost | [$] | **~12 700 000** | **~12 200 000** | **~23 100 000** | **~3 544 373** |
| Hull cost | [$] | **~2 678 160** | **~3 139 070** | **~5 407 590** | **~869 866** |
| Equipment cost | [$] | **~4 801 380** | **~4 700 970** | **~9 989 330** | **~1 227 157** |
| Power plant cost | [$] | **~5 228 790** | **~4 330 630** | **~7 666 480** | **~1 447 350** |

## BUILDING COST PREDICTIONS FOR SHIPS DESIGNED IN THE EUREKA PROJECT E!2772

The table above contains design parameters of the analysed ships, used for predictions of building costs. The total building cost of a ship consists of the hull construction cost, equipment cost and the cost of power plant with propulsion system.

## SUMMARY AND CONCLUSIONS

- A parametric (top-down) method has been presented of predicting the building costs of ships, to be used at the preliminary design stages. The method may be used to determine a ship evaluation criterion measure in the design methodology based on formal mathematical optimization models. The optimum design models may prove useful in the work of ship design offices, particularly at the time of strong competition on the world shipbuilding and shipping markets.
- The presented original parametric method of the estimation of ship building costs, with a mathematical structure applicable to computer optimization algorithms, is a contribution to the development of ship design theory.
- It is also a tool for practical ship design and was used for performing predictions of the building and operating costs of ships designed in the EUREKA project E!2772.
- It is expected that empirical verification of the prediction results will be the cost estimations prepared in the respective shipyards constructing the analysed ships.

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# Parametric method of prediction of the ship operating costs

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ABSTRACT

*Paper presents results of studies on a parametric method of predicting ship operating costs - useful in the preliminary ship design. Conception and theoretical basis of the method are presented, identified are also factors of significant importance for the ship operating costs, taking into account changes in the value of money. Approximation formulae for estimating the operating cost components have been developed as well as a computational algorithm based on a minimum Required Freight Rate (RFR). The useful character of the method is illustrated by examples of operating cost predictions for four different ship types designed in the Eureka project E!2772, i.e.: SINE 202 universal container carrier, SINE 203 oil product tanker, SINE 204 ro-ro ship and SINE 205 river-sea ship.*

**Keywords**: ship design theory, ship operating costs

## INTRODUCTION

In the preliminary ship design methodology formulated as an optimization problem, the aim is to determine such design solution which extremises a selected objective function (choice criterion) dependent on the sought ship parameters. The criterion measure may be a measurable ship property important for the economic effects of owner's business operations. An important element of a preliminary ship design methodology are the ship operating cost prediction methods with costs expressed as relations dependent on the design decision variables and the design mathematical model parameters.

At the time of formulation of ship design requirements, information on the future ship operation market conditions is very limited, uncertain and difficult to evaluate and any attempt to use it to perform expensive and time consuming studies aimed at optimizing the design requirements appears inefficient. Therefore, it may be justified to use simplified methods based on a small set of easy-to-get data of an index character in order to obtain approximate estimations with simple algorithms. That type of approach inspired the work on the presented method of predicting the ship operating costs by balancing the discounted costs and incomes.

The main design requirements of cargo ships, apart from the ship function, are usually: deadweight capacity **Pn** or cargo capacity **Pc**, service speed **v**, holds volume **Vc** and autonomous steaming range **R**. Results of studies on the methods of predicting ship operating costs in early stages of design, performed on different levels of accuracy of the analysed models, were frequently published, e.g. [1÷11]. Taking into account a greater number of model parameters leads to problem formulations which may be solved only with complex computational systems with the use of the optimization algorithms, e.g. those described in [12÷14].

## PROBLEM FORMULATION AND ASSUMPTIONS

A problem is under consideration where a ship of given deadweight capacity **Pn**, to be operating on a route of given length **R**, calls during trip at **s** ports for loading or unloading and it should achieve an assumed rate of return **e** during **m** years of ship operation. The required freight rate **RFR** is sought that would be a minimum rate in given technical and economic conditions of ship operation but would cover all the ship operating costs taken into account in the model.

Cargo supply in ports is generally of a stochastic character, particularly in the tramp shipping. A simplifying assumption is adopted in the method, allowing to use a deterministic model in the considered line shipping case. The stochastic character of cargo supply is provided for in an indirect way by introducing a coefficient $\boldsymbol{\varepsilon}$, expressing an average use of ship cargo capacity, which with sufficiently long ship operation periods allows to approximate an actual mean cargo supply. An advantage of this approach is easy way of determining the coefficient, e.g. by analysis of the log-book records of ships operating on a given shipping line.

The Required Freight Rate (**RFR**) is a rate that the owner should obtain in order to ensure the assumed rate of return **e** (with the borne ship investment and operating costs) in the ship operating period of **m** years.

The choice of **RFR** as a measure of covering the ship operating costs is based on reasoning that with an established level of actual freight rates on a given line, the best profitability will achieve a ship with the lowest required freight rate. If the future actual freight rates will be higher than the determined minimum freight rate **RFR** then the real rate of return **e*** will be higher than the assumed rate **e**, and if the opposite occurs then investment will not bring the assumed rate of return **e**. According to Schneekluth [9], in real operating conditions a minimum freight rate ship offers the greatest probability of achieving the required rate of return on shipping investment.

## INVESTMENT COSTS

We assume, as in [11], that the total ship building cost **J** consists of the propulsion system cost **Js**, dependent only on the ship speed, and of other investment costs **Jp**, dependent on the ship size expressed by the deadweight capacity:

$$J(Ne, Pn) = Js(Ne) + Jp(Pn) \tag{1}$$

According to [3], the **Js** and **Jp** costs are functions dependent mainly on the installed horse power **Ne** and deadweight capacity **Pn** and the functions are increasing slower than linear functions :

$$\begin{aligned} Js(Ne) &= Cs \cdot Ne^{2/3} \\ Jp(Pn) &= Cp \cdot Pn^{2/3} \end{aligned} \tag{2}$$

In this study, the **Cs** and **Cp** constants in expressions (2) were determined by the method described in [15].

## SHIP ANNUAL CARGO CAPACITY (ACC)

The Annual Cargo Capacity (**ACC**) of a ship operating, on average, **Z** days in a year, is determined as proportional to the ship cargo capacity and the number of trips in a year, with averaging capacity usage coefficient $\varepsilon < 1$:

$$ACC = n \cdot \varepsilon \cdot \lambda \cdot Pn \tag{3}$$

where $\lambda$ is a deadweight usage coefficient :

$$\lambda = \frac{Pn - Pz}{Pn} \tag{4}$$

where **Pz** means the weight of stores.

The number of trips in a year **n**, with calling at **s** ports and the route length **R**, is expressed by :

$$n = \frac{Z}{T} = \frac{Z}{\beta \cdot Tm} = \frac{Z \cdot v}{\beta \cdot R} \tag{5}$$

where **T** means duration of one trip and **Tm** is the actual sailing time in one trip.

The $\beta > 1$ coefficient is a correction taking into account the $T_j$ times of reloading operations and roadstead waiting time in one trip :

$$\beta = \frac{T}{Tm} = \frac{Tm + \sum_{j=1}^{s} T_j}{Tm} \tag{6}$$

The value of $\beta$ coefficient depends on the conditions on a given shipping line, number of ports, cargo handling efficiency etc.

## SHIP ANNUAL OPERATING COSTS (AOC)

Taking the ship operating cost analysis [1] and [9] as a basis, it has been assumed that the ship Annual Operating Cost (**AOC**) depends mainly on the fuel cost. In order to simplify the model, it was assumed that the average annual cost of lubricating oil and power plant repairs (dependent on the engine power) would be taken into account as a correction coefficient $\mu > 1$; then the annual operating costs are expressed as :

$$AOC = \mu \cdot n \cdot Tm \cdot Cj \cdot Gj \cdot Ne \tag{7}$$

**Cj** means unit fuel price [\$/t] and **Gj** means unit fuel consumption [t/(kW·h)].

Therefore, the annual ship operation cost is :

$$AOC = \mu \cdot n \cdot Tm \cdot Cj \cdot Gj \cdot \left( \frac{\varepsilon \cdot \lambda \cdot Pn}{\eta} \right)^{2/3} \cdot \frac{v^3}{Ca} = Kc \cdot n \cdot R \cdot Pn^{2/3} \cdot v^2 \tag{8}$$

where **Kc** means a respective product of parameters.

## DISCOUNTED AVERAGE ANNUAL COST (AAC)

With the required annual investment rate of return **e** and assumed **m** years of ship operation, the discounted Average Annual Cost (**AAC**) – operating cost and investment cost – is expressed as follows :

$$\frac{AAC}{CRF(e,m)} = J + \frac{AOC}{CRF(e,m)} \tag{9}$$

where **CRF** means Capital Recovery Factor. With the investment rate of return **e** and **m** years of ship operation, the **CRF** is determined from the formula :

$$CRF(e,m) = \frac{e(1+e)^m}{(1+e)^m - 1} \tag{10}$$

If a predicted rate of inflation **i** and the owner's income tax (on the profit calculated as a difference between the freight incomes and operating costs) rate **t** are to be taken into account in the model, then the Capital Recovery Factor formula will be :

$$CRF^*(e,m,i,t) = \frac{(e+i)(1+e+i)^m}{\left[(1+e+i)^m - 1\right](1-t)} \tag{11}$$

In such case, **CRF*** should be inserted in all the formulae containing the **CRF** expression.

The discounted average annual cost is :

$$AAC = J \cdot CRF(e,m) + AOC \tag{12}$$

## MINIMUM REQUIRED FREIGHT RATE (RFR)

If the **AAC** is covered by continuous incomes from the cargo transport freight rate during the year then the rate of return **e**, as assumed in the **CRF**, is achieved with that freight rate.

The required freight rate may be calculated from the required freight income to cover the **AAC** and the Annual Cargo Capacity (**ACC**) :

$$RFR = \frac{AAC}{ACC} \tag{13}$$

After respective substitutions and transformations, an explicit relation is given between the required freight rate and the constants and variable parameters of the model :

$$RFR = \frac{R}{Pn^{1/3}} \cdot \left[ \frac{\beta \cdot CRF(e,m)}{\varepsilon \cdot \lambda \cdot Z} \left( Cp \cdot v^{-1} + \frac{Cs}{\eta^{4/9} \cdot Ca^{2/3} \cdot Pn^{2/9}} \cdot v \right) + \frac{\mu \cdot Cj \cdot Gj}{\eta^{2/3} \cdot \varepsilon^{1/3} \cdot \lambda^{1/3} \cdot Ca} \cdot v^2 \right] \tag{14}$$

As all the factors and components of this equation have positive values, their impact on the minimum required freight rate may be evaluated in a simple way.

## EXAMPLES OF THE RESULTS OF SHIP OPERATING COST PREDICTIONS

The following table contains technical and economic parameters of the ships designed in the Eureka project E!2772, which were used to perform the ship operating cost predictions with the method described here.

## EVALUATION OF RESULTS AND CONCLUSIONS

An impact of some model parameters on the predicted operating cost is intuitively obvious. However, the obtained analytical relations allow to analyse the impact determined by the index exponents of parameters in the model. The following observations and conclusions may be drawn from the analysis :

- increased **RFR** means higher ship operating costs
- increased required rate of return **e** – expressed by increased **CRF** – implies higher **RFR**

***Table.*** *Design parameters and operating cost predictions of the project ships, in US dollars*

| RESULTS OF SHIP OPERATING COST PREDICTION SIMULATIONS | Symbol | Unit | SINE 202 950 TEU 8 550 DWT Container carrier | SINE 203 14 300 DWT Product tanker | SINE 204 7 400 DWT Ro-ro ship | SINE 205 2 950 DWT River-sea ship |
|---|---|---|---|---|---|---|
| Assumed ship deadweight capacity | Pn' | [ t ] | 8 550 | 14 300 | 7 400 | 2 950 |
| Price of a reference ship | J | [ $ ] | 12 708 330 | 12 170 670 | 23 063 402 | 3 544 370 |
| Price of a reference power plant | Js | [$] | 5 228 790 | 4 330 630 | 7 666 479 | 1 447 350 |
| Power of a reference ship engine | Ne | [kW] | 11 200 | 10 000 | 19 520 | 3 640 |
| Displacement of a reference ship | D | [t] | 15 600 | 19 922 | 15 850 | 3 938 |
| Speed of a reference ship | v | [kn] | 18.5 | 14 | 20 | 12 |
| Deadweight capacity of a reference ship | Pn | [t] | 8 550 | 14 300 | 7 400 | 2 950 |
| Cargo capacity efficiency coefficient | ε | [-] | 0.85 | 0.85 | 0.85 | 0.85 |
| Deadweight efficiency coefficient | λ | [-] | 0.9 | 0.9 | 0.9 | 0.9 |
| Number of operating days in a year | Z | [days] | 340 | 340 | 340 | 340 |
| Trip time to sailing time ratio | b | [-] | 1.1 | 1.1 | 1.1 | 1.1 |
| $\frac{\text{Cost (fuel+repairs+lub. oil)}}{\text{Cost (fuel)}}$ | g | [-] | 1.4 | 1.4 | 1.4 | 1.4 |
| Fuel price | Cj | [$/t] | 200 | 200 | 200 | 200 |
| Unit fuel consumption | gj | [g/kWh] | 170 | 170 | 170 | 170 |
| Rate of return | e | [-] | 0.06 | 0.06 | 0.06 | 0.06 |
| Number of years of ship operation | m | [-] | 20 | 20 | 20 | 20 |
| Tax rate | t | [-] | 0.18 | 0.18 | 0.18 | 0.18 |
| Average annual inflation rate | i | [-] | 0.03 | 0.03 | 0.01 | 0.03 |
| Steaming range | R | [nm] | 10 000 | 6 000 | 8 000 | 4 000 |
| Assumed ship speed | v | [kn] | 18.5 | 14 | 20 | 12 |
| Required Freight Rate | RFR | [$/t] | 75.8 | 24.2 | 80.2 | 50.8 |
| Number of trips in a year | n | [-] | 14 | 17 | 18 | 22 |
| Investment cost | IC | [$] | 12 708 329 | 12 170 666 | 23 063 402 | 3 544 373 |
| Discounted annual fuel cost | FC | [$] | 3 307 984 | 2 953 557 | 5 765 344 | 1 075 095 |
| Discounted annual cargo handling cost (10 $/t) | HC | [$] | 1 795 257 | 3 787 056 | 2 099 716 | 1 004 459 |
| Discounted Average Annual Cost | AAC | [$] | 6 800 989 | 8 366 532 | 10 519 965 | 2 553 058 |
| Annual Cargo Capacity | ACC | [t] | 89 763 | 189 353 | 104 986 | 50 223 |

- technical progress, characterised by better ship resistance--propulsion efficiency and expressed by greater **Ca** coefficient, causes decrease of **RFR**
- impact of the ship deadweight capacity **Pn** on **RFR** is ambiguous. For specific conditions of a task, the final effect depends on relations of the remaining model parameters and constants
- the increase of efficiency factors in the model causes reduction of the required minimum freight rate, i.e. reduces the operating costs
- increased ship hull and equipment prices cause increase of **RFR**
- increased fuel price **Cj** implies increased **RFR**
- technical progress in the engine design, leading to reduction of the unit fuel consumption **Gj,** decreases **RFR**.

The presented ship operating cost prediction method analyses the cost as dependent on a number of factors. The studies presented in this paper were performed with average values of the factors, as given in the Table summarizing the operating cost simulation calculations. The obtained results should therefore be read together with the respective parameter values, which reflect the current and future technical and economic relations.

**NOMENCLATURE**

**Ca** - Admiralty coefficient
**Cp** - other cost coefficient
**Cs** - installed power plant cost
**e*** - real rate of return
**Gj** - unit fuel consumption
**Jp** - other investment cost
**Pc** - cargo capacity
**Pz** - weight of stores
**s** - number of ports for reloading in one trip
**T** - duration time of one trip
$\mathbf{T_j}$ - time of reloading operation in one trip
**Tm** - duration of one way voyage
**Vc** - volume of holds
$\beta$ - waiting time correction coefficient
$\eta$ - deadweight/displacement ratio
$\lambda$ - deadweight usage coefficient
$\mu$ - correcting factor for operating costs

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# Methodology of assessing the production capacity of selected shipyards

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ABSTRACT

*The paper deals with selected subjects in the European EUREKA „Baltecologicalship” E!2772 project. Within the project, preliminary designs of four feeder (short shipping) ships have been developed based on the market trend research, design knowledge and experimental investigations. The following tasks were to be performed for one of the ships: defining optimum building conditions (timescale, production technologies, ecological aspect), selecting shipyards meeting the minimum production capacity requirement. For that purpose an index method was developed (based on the design and production indices). Then an index analysis was performed of the production capacities of selected shipyards (and also different configurations of cooperation between them) and results were presented in a form of the Gantt diagrams. Final conclusions were drawn.*

**Keywords** : structural and production process coefficients, index method, labour demand, worstand - hours (Stg)

## PROBLEM DEFINITION

An increasing competition for contracts to build new cargo ships requires continuous watching the shipbuilding market as well as improving production methods and capabilities. At present, Polish shipyards have an order book large enough to ensure survival of the industry but they are not free of the shortage of capital [2]. This is a temporary situation and it makes shipyards seek solutions necessary to win the competition. An example of such action is implementing the conception of the so called fractal factories [1].

In view of a complex and multistage process of building a ship from conception to delivery, certain stages may be distinguished where solutions should be sought to improve economic competitiveness and to meet the ecological production requirements.

In the current economic conditions in the country and capital weakness of the large production enterprises, one can hardly expect any significant investments in the shipyard technical infrastructure in the nearest future. Therefore, solutions should be looked for in the production engineering and organization. With so defined strategy of competitiveness, the following aims may be formulated :

**In the technological conditions of Polish shipyards, a broad collaboration between shipyards and the cooperating enterprises, particularly from the small and medium enterprise (SME) group, should be developed in order to use the production and innovative potential in an optimum way. Such cooperation should be based on mutual economic benefits and long-term planning as well as on the shipbuilding market development trends.**

Such collaboration depends on fulfilling the following conditions :

1. The manufacturers have a well prepared (small number of modifications) and easily accessible (e.g. in an electronic form) documentation for customers. The present level of computer software and hardware allows to prepare complete ship construction documentation. [3, 4, 5].
2. Ship structure production quality standards for the project in question are implemented at the leading shipyard and at subcontractor plants, together with agreements between the quality control units.
3. Specialist teams are shifted between the cooperating enterprises to perform specific tasks, or production of some structural sections is outsourced and then transported to the leading shipyard for assembly in dock.
4. An efficient and effective transport operates between the enterprises (between the Gdańsk-Gdynia area and the Szczecin area shipyards the waterway transport will be most appropriate).
5. Technologies not meeting the ecological production requirements are eliminated.
6. A clear and unified ship construction process is developed (together with all the quality control instruments) and implemented in the project executing enterprises.
7. A unified algorithm of assessment of the subcontractor production capacity is used to subdivide the tasks in a rational way.

The above listed conditions comprise a broad range of different problems. The problems of optimum production process planning and subdivision of technical tasks among the subcontractors were chosen for further analysis.

With those conditions, a process algorithm was proposed in order to perform the main task. In view of the broad range of conditions, items 1, 2, 3, 4, 5 were assumed fulfilled and the emphasis was placed on the production process conditions 6,7.

The analysis was performed on the SINE-203 product tanker hull [6], Fig.1. A series of simplifying assumptions related to the structure and production process were adopted. The more significant ones include :

- the subject of analysis is **steel hull** of the ship, without outfitting
- the hull structure was simplified, with panel and stiffener elements only (the plate thicknesses and diversity of stiffeners were maintained) and with subdivision into flat and bent elements
- real arrangement of external shell seams was replaced with a virtual arrangement of the same total length (without locating seams on the shell).

With the above listed assumptions, a material balance and weld quantification were prepared.

***Main dimensions :***

| | | | |
|---|---|---|---|
| Length overall : | $L_{oa}$ = 138.100 m | Speed : | $V_s$ = 14.0 knots |
| Length b. perpendiculars : | $L_{pp}$ = 132.000 m | Hull weight : | $M_k$ = 4 250 t |
| Width : | B = 22.50 m | Draught : | T = 8.70 m |
| Depth : | H = 12.80 m | Underdeck volume (at T) : | V = 19 375 $m^3$ |

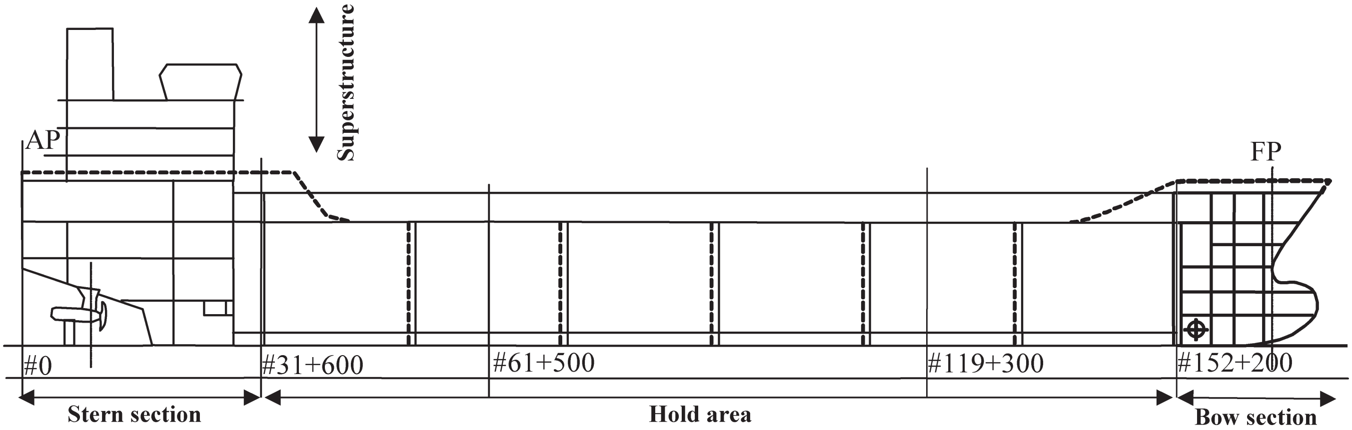


***Fig. 1.*** *Subdivision of the SINE 203 ship into structural areas*

## UNIFICATION OF THE TECHNOLOGICAL PROCESS OF SHIP HULL CONSTRUCTION

Unification (schematisation) of the technological process was prepared to make comparisons possible between technological capabilities of different shipyards. The process schematic diagram is presented in Fig.2. The most important stages of a technological process are included.

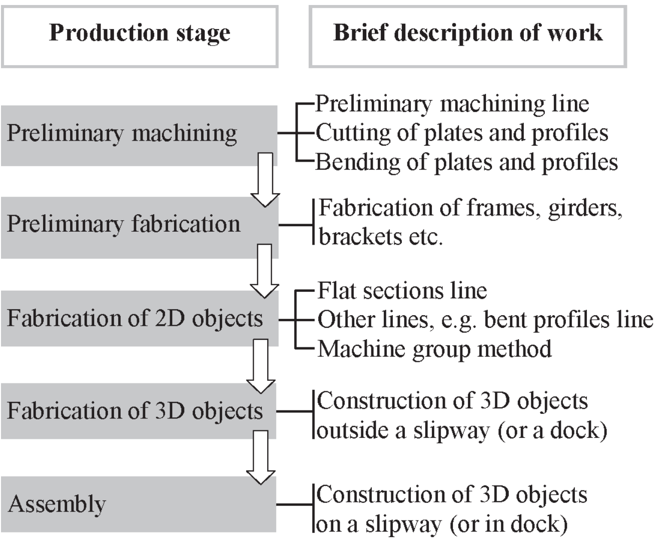


***Fig. 2.*** *Unified production process of the ship hull construction*

## SELECTION OF SHIPYARDS FOR THE COOPERATION ANALYSIS

Analysis was performed with reference shipyards A, B and C of different production capacities. A hull production cycle of six months was assumed.

**Shipyard A** has much greater production capacity than that required by the SINE 203 oil product tanker.

**Shipyard B** covers a variant of cooperation between **Shipyard A** and a small production capacity shipyard. Cooperation includes the preliminary machining, preliminary fabrication and 2D object fabrication stages.

**Shipyard C** covers a variant of cooperation between shipyards of similar production capacities, where the preliminary machining, preliminary fabrication, 2D object fabrication and 3D object fabrication is performed by one shipyard and assembly in dock is performed by the other shipyard. This is a production cycle often practiced by the SME sector shipyards.

## INDEX METHOD OF ASSESSING THE SHIPYARD PRODUCTION CAPACITY

### *Traditional method*

Assessing the capability of constructing a hull in the existing production conditions requires an individual approach, i.e. an optimum production process should be found for a given hull geometry in a given shipyard with its existing infrastructure.

### *Index method*

The indices determine the structure and production process (coefficients) assigned to the characteristic operations of individual construction stages, which allows to express objectively (in workstand-hours - Stg) the labour demand of respective stages. By determining the indices of individual shipyards, we obtain a picture of production capacities, which helps to take cooperation decisions (Fig.3).

The **production process coefficients $T_i$** are determined in relation to the shipyard infrastructure and they describe :

- productivity, expressed by [t/Stg], [$m^2$/Stg], [m/Stg]
- size and weight, expressed by dimensions - a, b, c [m] and structure weight [t].

The $T_i$ values may be determined in several ways from :

- counting the basic production operation times - classic approach [7]
- an adopted system of integrated calculation coefficients [8]
- experience, i.e. an average value of annual production balance.

The **structural coefficients $W_i$** are determined in relation to the hull structure (quantity of material to be processed in the production stages in order to obtain a hull) and they are :

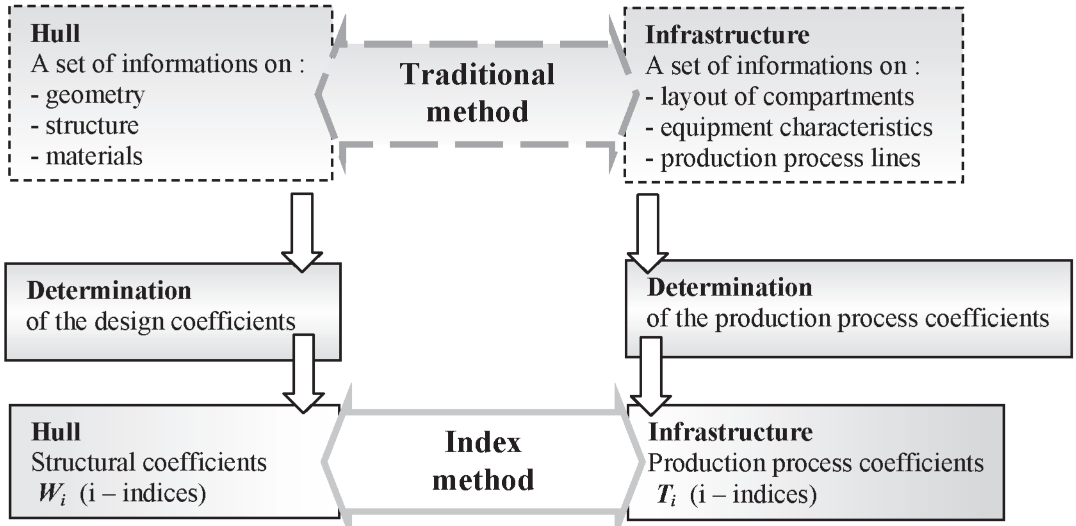


***Fig. 3.*** *Flow diagram of the methods of comparing known hull structural and material conditions with production capabilities of a given shipyard*

- productivity related, expressed by [t], [$m^2$], [m], [pcs]
- size and weight related, expressed by dimensions - a, b, c [m] and structure weight [t].

**Production process coefficients.** Using the adopted unified production process divided into five stages (Fig. 2), coefficients were assigned to the respective work scopes. Their values were determined by evaluation of the production capacities (from the integrated calculation indices [7]) for specific work scopes in each of the analysed shipyards. The coefficients are presented in Table 1.

**Structural coefficients**. From the documentation and the prepared material and weld length balance, coefficients were assigned to the respective work scopes. The coefficients correspond with the quantities of material to be processed in the production stages. The coefficients are presented in Table 2.

A total of 15 production process coefficients and corresponding structural coefficients were adopted. In assigning values to the structural coefficients, a different hull structure subdivision was assumed for each shipyard.

## RESULT OF ANALYSIS

By linking the production process coefficients and structural coefficients, the operation times $\mathbf{R_i}$, expressed in [Stg], were obtained for the respective workstands, in accordance with the work scope assumed (Table 3).

Gantt diagrams were prepared from the tables for different variants. Examples of hull construction schedules are presented here below for the proposed **A, B, C** shipyards.

The schedules presented here for the selected shipyards are one of the feasible variants with the assumed coefficients.

***Table 1.*** *Identification of production process coefficients*

| Item | Coefficients | Description / Unit / Productivity | Scope of operations |
|---|---|---|---|
| ***Preliminary machining line*** | | | |
| 1. | $T_1$ | Productivity in [$m^2$ / Stg] | Plate straightening/cleaning/painting |
| 2. | $T_2$ | Productivity in [m / Stg] | Profile straightening/cleaning/painting |
| ***Fabrication of 3D objects*** | | | |
| 12. | $T_{12}$ | Fillet weld length [m/Stg] related to the 2D stage | Construction of 3D objects outside the slipway (or dock) |
| 13. | $T_{13}$ | Field dimensions (a/b/c) and load lifting capacity ($t_{min}$) depending on the structure subdivision | Demand for the operating area and crane lifting capacity needed to fabricate the object |
| The 1, 2, 12, 14 coefficients are of the productivity evaluation character<br>The 13, 15 coefficients are of the size and weight character<br>[Stg] - workstand-hours. Time needed to perform a given scope of work on the workstand. Number of workers is not taken into account | | | |

***Table 2.*** *Identification of structural coefficients*

| Item | Coefficients | Description / Unit / Productivity | Quantity of material to be processed |
|---|---|---|---|
| ***Preliminary machining line*** | | | |
| 1. | $W_1$ | Determination of the plate sheet area in the structure [$m^2$] (3x12) - assumed sheet size | Plate sheet area in the structure |
| 2. | $W_2$ | Determination of the length of stiffeners in the structure [m] | Total length of stiffeners in the structure |
| ***Fabrication of 3D objects*** | | | |
| 12. | $W_{12}$ | Fillet weld length [m/man-hour] related to the 3D stage | List of welds differentiated according to type |
| 13. | $W_{13}$ | Field dimensions (a/b/c) and load carrying capacity ($t_{min}$) depending on the structure subdivision | For a given hull structure subdivision, demand for the operating area and carrying capacity of the transport means |
| The 1, 2, 12, 14 coefficients are of the productivity evaluation character<br>The 13, 15 coefficients are of the size and weight character<br>[Stg] - workstand-hours. Time needed to perform a given scope of work on the workstand. Number of workers is not taken into account | | | |

***Table 3.*** *List of reference coefficients for the SINE 203 ship*

| | | | | |
|---|---|---|---|---|
| | | ***Preliminary machining line*** | Type of result | $R_i$ [man-hours] |
| **1.** | $\mathbf{R_1}$ | $\mathbf{W_1}$ [m$^2$] / $\mathbf{T_1}$ [m$^2$/Stg] | Number | $R_1$ |
| **2.** | $\mathbf{R_2}$ | $\mathbf{W_2}$ [m] / $\mathbf{T_2}$ [m/Stg] | Number | $R_2$ |
| | | ***Fabrication of 3D objects*** | | |
| **12.** | $\mathbf{R_{12}}$ | $\mathbf{W_{12}}$ [m] / $\mathbf{T_{12}}$ [m/Stg] | Number | $R_{12}$ |
| **13.** | $\mathbf{R_{13}}$ | $\mathbf{T_{13}}$ - Constructed object (a/b/c), object weight [t]<br>$\mathbf{W_{13}}$ - Workstand dimensions (a/b/c), load carying capacity [t] | Yes / No | $R_{13}$ |
| One working month has 178 hours, with 3-shift work it is 534 [h] | | | | |

$\mathbf{R_i}$ is a coefficient with index **i**. If it is a productivity-type coefficient then it assumes an assigned value [Stg]. The size and weight coefficients are assigned logical values: 1 when the operation is possible or 0 when the operation is not possible.

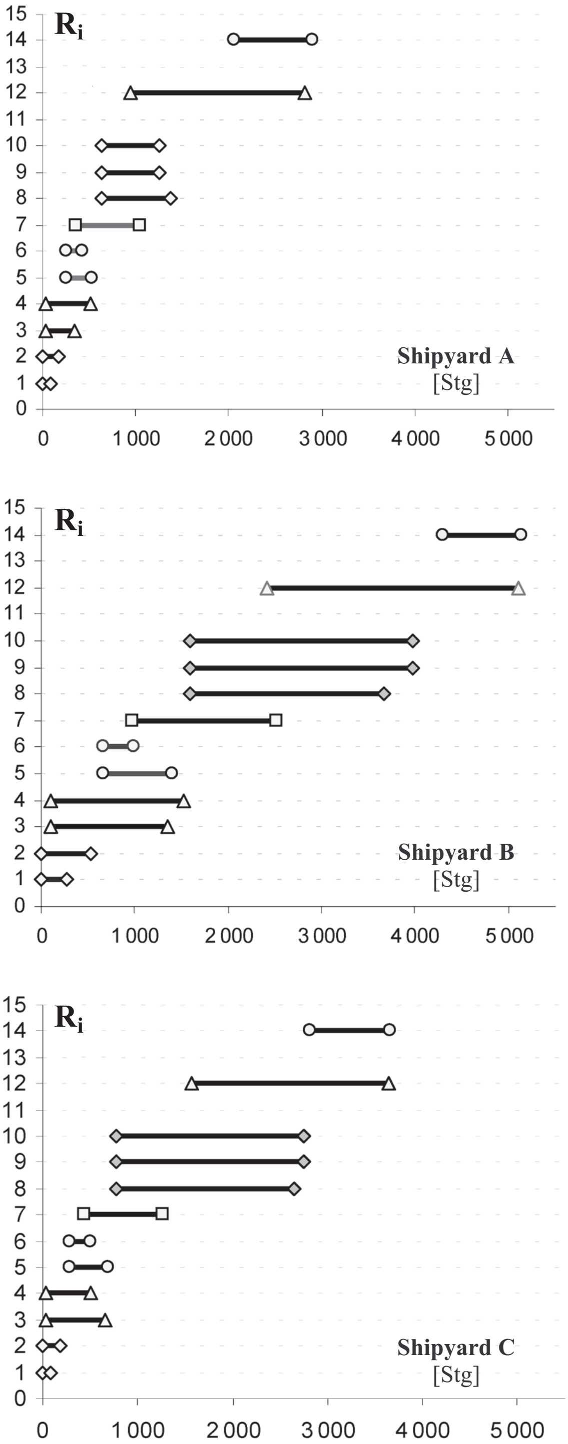


***Fig. 4.*** *Three hull construction schedules*

In the analysed cases, the size and weight coefficients were assigned the 1 values.

The **Shipyard A** schedule shows a correct hull construction process with the assumed coefficients.

The **Shipyard B** schedule presents an incorrect process. Too large percentage (scope) of the tasks were entrusted to a weaker partner, which caused lengthening of the process.

The **Shipyard C** schedule presents a correct process of the cooperation of shipyards, none of which would be individually able to build the hull in the planned time of 6 months. With such way of determining the labour demand, the question of calculating the values of [Stg] and man-hours in different shipyards remains open. Negotiations between the parties may be expected.

## ECONOMIC ASPECT OF THE CONSTRUCTION

With so determined construction schedules (Fig. 4), the hull construction cost analysis may be carried out. The simplifying assumptions are the following :

- the [Stg] cost is the same for each construction stage
- the [Stg] cost for different subcontractors is the same.

The schedules may be presented in a form of the so called logistic curve (S curve) - Fig. 5.

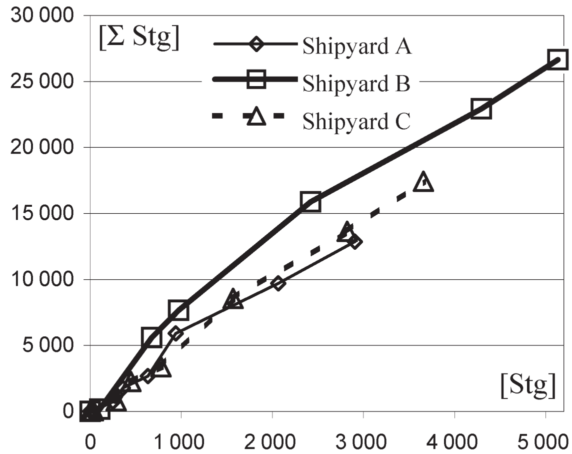


***Fig. 5.*** *The S curves for shipyard A, B, C*

Three S Curves, reflecting the work intensiveness, were obtained for three shipyards (the assumed construction conditions).

**Shipyard A** - carries out the hull construction in a most productive way, i.e. very intensively and in a short time.

**Shipyard B** - carries out the hull construction in a least productive way and in a long time.

**Shipyard C** - carries out the hull construction in a slightly worse way than Shipyard **A**, but still within acceptable limits.

If we assume that the whole sum needed for the construction, e.g. 100 units, is borrowed with an e.g. 20% per annum interest, then the total cost will be as shown in Fig.6.

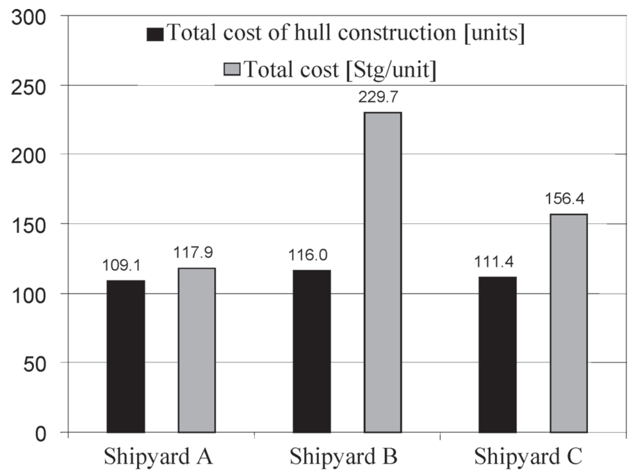


***Fig. 6.*** *Construction costs and [Stg] costs for the analysed shipyards*

If a [Σ Stg/total hull construction cost (units)] relation is formed then the Stg cost will be obtained (Fig.6). The following results were obtained for the analysed cases: **Shipyard A** had the best and **Shipyard B** the worst result. Although **Shipyard C** had only a little longer construction time than **Shipyard A**, the increase of [Stg/unit] is evident. The reason of distinct differences in the analysis (Total cost [Stg/unit]) is simultaneous cumulation of the longer construction time (**Shipyard B** and **C**) and longer credit interest payments. The hull outfitting was not taken into account in the analysis but it may be expected with certainty that cost differences ([Σ Stg / Σ units], Fig.6) between shipyards will be significantly increased.

With the use of parametric description of a greater number of ships (structural coefficients) and parametric description of shipyard (and subcontractor) infrastructure, the following studies may be performed :

- ❖ production program for the leading shipyard, which may be optimized in respect of the distribution of tasks between the shipyard itself and the subcontractors
- ❖ methods of financing the individual tasks
- ❖ determination of a minimum profitable production treshold and maximum production capacity of the leading shipyard
- ❖ from the above studies, determination of a long-term production schedule for the whole industry branch; such preparatory work will allow to survive the low demand period and to start production of new ships as soon as the demand for them returns.

These studies may be performed by building a mathematical model of the shipbuilding industry branch. This is an interdisciplinary task for the shipbuilding, banking, environmental engineering etc. experts.

## SUMMARY

- ➲ The **index method** gives an answer whether in the technical conditions characteristic of a shipyard the construction is possible, how long time it will take (with the assumed parameters), what will be the construction process „efficiency”.
- ➲ The index-based analysis may be applied to any type of ship. The differences between ship hull types will consist in the values of the structural coefficients, particularly those connected with the structural subdivision of hull, i.e. weights and sizes of sections and blocks.
- ➲ Comparison of the indices characteristic of a hull structure with indices characteristic of a shipyard infrastructure allows to assess the construction capability in a rational (objective) way.
- ➲ The accuracy (reliability) of the proposed approach depends on correct indication of important points in the production process and proper correlation with the indices (characteristics) of the constructed hull.
- ➲ The method allows to compare in a simplified way the production capabilities of several shipyards in relation to the construction of the same hull. That gives a technical basis for the choice of contractor(s).

**Advantages of the index method**.
with precise identification of the production capabilities of Shipyard A, B, C :

- ❈ the proposed index method allows to perform a variant preliminary assessment of the hull construction effectiveness
- ❈ an exact information is obtained, in the form of a number or a logical value, related to the index character
- ❈ the index structure (method of calculation) reflects its scope and role from the point of view of efficient execution of the process
- ❈ the whole construction process may be covered with an appropriate number of indices
- ❈ by increasing the number of indices, the construction process may be more precisely modelled together with cost estimations of individual operations (if needed)
- ❈ by carrying out a similar index analysis, one can easily learn of the strong and weak points of another shipyard.

**Disadvantages of the index method** are the following :

- ★ some operations of the construction process are left out
- ★ too big scope of work represented by one index may lower the priority of an important operation, which in turn may lead to a too low (or too high) index value
- ★ partial overlapping of the index ranges, which leads to too high end values
- ★ the indices must be calculated by the same method for all the compared shipyards, regardless of their production profiles.

Change of the method of calculation of one index makes it necessary to recalculate the indices for other shipyards.

**Directions of development**: some actions may be indicated to improve the proposed method :

- ⅄ appoint a group of potential partners in the region
- ⅄ determine the indices, by the proposed method, for the appointed group of partners
- ⅄ analyse the so far achieved results in hull construction :
  option 1 - individual work of a shipyard,
  option 2 - in cooperation with other partners
- ⅄ compare and summarize the results.

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# Service margin - solution of the problem or a problem waiting for solution ?

**Janusz Stasiak**, D.Sc., Eng.

ABSTRACT

*Subject of the paper is the problem of predicting sea-going ship resistance and/or horsepower in real operating conditions. Attention is drawn first of all to the need and possibility of revising the very much outdated but still used "service margin" method, which consists in adding an arbitrary percentage margin to the value of resistance or horsepower, relatively precisely determined for the calm water conditions. A negative impact in this respect is shown of the generally used delivery-acceptance procedures, where particular importance is attached to the ship propulsion tests on the "measured mile". The need of revising the "service margins" is a consequence of an obvious need for most efficient ships from the technical as well as economic point of view. Secondly, it is a "must" of permanent improving the ship design quality - the adequacy and accuracy of design methods. The work presents the "wave service margin" coefficient models. It is assumed that they may contribute to the necessary rationalization of the procedure of real ship resistance and/or horsepower determination. The work is based on the results of resistance tests of a series of ships designed within the Baltecologicalship project. The tests were carried out in the Chair of Ship Theory and Design of the Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology.*

**Keywords** : sea-going ship, design, seagoing qualities, resistance, main propulsion horsepower, service margins

## INTRODUCTION

### *The role of ship hydromechanics in ship design*

There is no doubt that the hydromechanical properties of a sea-going ship are a fundamental quality (a set of qualities) of the craft. They determine its physical substance and, to a large extent, its functional effectiveness. It seems therefore fully justified to think that ship design, aimed at creating the best possible ship, will be particularly receptive to all the existing test models and will stimulate development of more and more adequate and precise models.

However, in the confrontation with reality that may appear to be wishful thinking. In fact, the ship design methods have been based for decades on the same hydromechanical solutions developed at the end of the 19th and beginning of the 20th century. The later developed solutions, quite useful and definitely of better quality, have not been appropriately used in design. As that pertains to the basic and vital ship characteristics, such as transverse stability, load-lines, unsinkability and the resistance-propulsion properties, it is difficult to qualify the ship design as a modern and professional practical speciality. One can hardly expect its products to be, as a rule and not only by accident, as good as they can objectively be.

Therefore one has to agree, though regretfully, that :

„*The naval architecture is a science of vague assumptions, based on debatable figures taken from inconclusive instruments, performed with equipment of problematical accuracy by persons of doubtful reliability and questionable mentality*” [a)].

That opinion has to be accepted because in the more and more institutionalized and bureaucratic shipbuilding and shipping system there are no signs of will and action aimed at changing the situation. On the contrary, quite clear are mechanisms and tendencies of maintaining the status quo in this respect.

The problem discussed here is also important as it is related to the theory of ships - a research and practical science, by definition aimed at creating new more accurate and practically useful models of hydromechanical phenomena. Lack of demand from the practical design causes decreasing interest in creating such models and concentration on the in fact ineffective „polishing” of the hydromechanical solutions used in ship design procedures for years. That particular pragmatism[b)] of the theory of ships closes the „positive feedback” cycle, which effectively petrifies and even deepens the hydromechanical backwardness of shipbuilding.

### *Service margin*

The above characterised situation is reflected in predicting the ship service resistance and/or horsepower. Now, like in the past decades, the main design and research effort is concentrated on the idealised part of the problem, related only to the calm water conditions. Ship resistance and/or horsepower in real sea conditions is treated „marginally”: they are not used as the hull size and shape selection criteria and their values are roughly predicted by means of a „service margin”. The „service margin” method is the simplest of all the possible ways of predicting the real values of resistance $\mathbf{R_{TS}}$ $(\mathbf{V_E})$ and/or the ship main propulsion horsepower $\mathbf{P_{TS}}$ $(\mathbf{V_E})$ typical of the normal sea conditions and the required service speed $\mathbf{V_E}$. It consists in determining those values as a product :

$$R_{TS}(V_E) = (1+k_w) \cdot R_T(V_E)$$

or : (1)

$$P_{TS}(V_E) = (1+k_w) \cdot P_T(V_E)$$

where :

- $\mathbf{R_T(V_E)}$ or $\mathbf{P_T(V_E)}$ are the ship calm water resistance or horsepower values, respectively, for the speed $\mathbf{V_E}$
- $\mathbf{k_w}$ is a dimensionless resistance and/or horsepower reserve (margin) coefficient, whose values are determined by a rather subjective selection from the $\mathbf{k_w} \in (0.15 ; 0.35)$ range.

In the past, up to mid 1960s, the so defined „service margin” method was a justified practical design approach. At that time it was the only practical solution of the complex and important problem, not yet investigated and, needless to say, modelled. Sticking to that method later when much more accurate and useful real sea resistance (horsepower) prediction design tools became available[c)] has been a mere ship design

„shop floor neglect". More importantly, that has been a neglect with an evident effect - the resistance and horsepower „oversize" ships.

As it was demonstrated (e.g. in [1, 2, 3]), ships designed in the mid-twentieth century did not use in service ca. 20% of their installed horsepower, whereas their average (taken from several dozen sea trips) service speed was in the range of 60% to 103% of the design speed. In the course of years, when the cargo ship design speed and horsepower increased quickly, the oversizing problem was becoming (it had to become) more and more clear. It gained importance also because of the significant improvement of ship hull construction technology.[d)] Therefore, practical economic indications could be seen to rationalize the design methods - to include the seagoing qualities into the ship hull design criteria or at least to reduce the service margins. It could have been expected that such revision would soon become a fact. Nothing of the sort has ever happened.

Still a lot is being done in the ship design process to find a hull shape with a still water resistance lower by so much as a fraction of percentage point; the seagoing qualities of such ship are generally a second-rate matter, most often totally neglected. And still the so scrupulously minimized resistance is then roughly increased by a **15** to **35 percent** service margin. Therefore, it may and should be assumed that even modern sea-going ships „carry" at least a dozen or so percent reserve of never used main propulsion system horsepower.

### *The ship delivery-acceptance procedure*

It seems that the reasons of such a deformed situation in the ship design resistance and/or horsepower prediction question should be sought first of all in the firmly established and institutionalized ship delivery-acceptance procedures, where a particular role is assigned to the ship speed (propulsion capability) test on the measured mile.

That test, intended to be an objective verification of real propulsive and speed qualities of the ship, in fact does not provide such a verification. It does stimulate, however, working out such design resistance-propulsion characteristics which meet the respective contract requirements in the test (still water) conditions. For the shipyard, negative result of that test means a sure financial loss determined in the contract. On the other hand, designing a ship having the resistance-propulsion characteristics well suited to the typical operating conditions, regardless of the measured mile test results, does not bring the shipyard (designer) any additional financial profits.

As long as such a ship acceptance procedure is in use, one should not expect any significant changes (rationalizing) in designing the ship resistance-propulsion properties. Taking into account :

- strong adherence of the shipbuilding and shipping industry to the generally accepted solutions, particularly those of the administrative and legal character
- objectively illusory but individually irrefutable conviction on the part of the owner of the necessity of checking the ship performance characteristics at the moment of delivery
- an evident interest of the shipyard and designer in applying a sanctioned service margin with its broad range of values, in order to „hide" all the inaccuracies of the resistance and horsepower predictions,

it will take long for any changes in this respect to occur. Therefore, it may be assumed that the service margin will still be used in the ship design procedure for a long time.

In this situation, what can and should be reasonably done is **constant updating of the service margin values** – improving the accuracy of their $\mathbf{k_w}$ coefficients. This task belongs to the domain of the theory of ships. It is an immanent duty of that speciality to generate more and more adequate models of all those hydromechanical phenomena which determine the functioning of ships. That duty is absolutely indispensable in relation to the phenomena and problems connected with the ship safety and operational efficiency.

### *The purpose and subject of investigation*

Apart from the above presented remarks, the purpose of this paper is also to present some real possibilities of determining more adequate values of the service margin coefficient $\mathbf{k_w}$. Those possibilities in general consist in making an effective use of the **seakeeping ability methods**. In particular, they consist in **generalization of the long-term prognosis of additional ship resistance** from sea waves $\mathbf{R_{AV}}$.

It seems that the so obtained expressions of the type :

$$\frac{R_{AV}}{R_T} = f\left(Fn_E, K\right) \tag{2}$$

where :

$\mathbf{R_T}$ - is the ship still water resistance, $Fn_E = \dfrac{V_E}{\sqrt{g \cdot L}}$ is a dimensionless design ship speed and **K** represents the ship hull governing geometric characteristics, may be correct measures, if not the total $\mathbf{k_w}$ coefficient then at least its very important component **k – the service margin wave part coefficient**.

One way or another, such expressions may be an important contribution to the necessary rationalization of the service margin values.

The paper is based on the results of ship resistance tests carried out in the Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology, in the „Balteco-logicalship" project and is an additional summary of those tests. The test materials are presented and discussed in detail in [5, 6, 7, 8].

## LONG-TERM PROGNOSIS OF THE INCREASED RESISTANCE OF A SHIP MOVING IN HEAD WAVES

### *Definitions*

- The ship resistance increase (additional resistance) was predicted (determined) only for the **head sea conditions** i.e. for wave incidence angle **β** equal to **180°**.
- **The long-term prognosis of resistance increase** (additional resistance from sea waves) $\mathbf{R_{AV}}$ of a ship moving with constant speed **V** were determined as probabilistic estimation of a mean value of a set of random short-term additional resistance $\mathbf{R_{AW}(x)}$ prognosis for that ship :

$$R_{AV} = \sum R_{AW}(x) \cdot p(x) \tag{3}$$

where :

- **x = {a, b, c}** is a set of discrete random **sailing conditions**, which have assigned to them the resistance increase values $\mathbf{R_{AW}(x)}$ determined as short-term prognosis. Elements of the **x** set are: ship loading conditions [**a**], sailing regions [$\mathbf{b(b_1,b_2)}$] – seas ($\mathbf{b_1}$) and their regions ($\mathbf{b_2}$) and sea states – wave conditions (**c**) characteristic of those seas
- $\mathbf{p(x) = p(a) \cdot p(b) \cdot p(c|b_1)}$ is a probabilistic model of conditions **x**, or the combined probability function of discrete random variables **a, b,** and **c**.

➢ **The short-term prognosis of resistance increase $R_{AW}$ (x)** (prognosis of resistance increase in the stationary irregular wave conditions) were determined in accordance with the principle of superposition expressed as follows :

$$R_{AW}(x) = 2\int_{\omega} r_{AW}(\omega, a)\cdot S(\omega, b, c)\cdot d\omega \qquad (4)$$

where :

- $r_{AW}$ (**ω , a**) is a dimensionless resistance increase coefficient (resistance increase operator) determined from ship **model tests** carried out in regular head waves of different frequencies **ω**
- **S** (**ω**, **b, c**) is a model of the spectral density function of stationary irregular waves, the form of which was :
  - assigned to a sea region $\mathbf{b_2}$. For restricted (coastal) waters – the JONSWAP spectrum, for unrestricted (open) waters – the ISSC spectrum
  - identified each time by the sea state characteristics $\mathbf{c = c(H_{1/3}, T_1)}$ : significant wave height $\mathbf{H_{1/3}}$ and characteristic period $\mathbf{T_1}$.

➢ **The resistance increase prognosis** were prepared for :

- shipping routes in the **North Sea** and **Baltic Sea** regions, whose probabilistic model **p**(**b**) was based on the Author's knowledge
- random stationary waves, whose probabilistic model $\mathbf{p(c) = p(H_{1/3}, T_1)}$ was based on the information taken from the „Global Wave Statistics" atlas [4]
- three ships with two loading conditions [light ($\mathbf{a_1}$) and heavy ($\mathbf{a_2}$)] described by the model **p**(**a**) assumed by the Author
- three speeds **V** of each ship. The speed values were in the range $V \in < 0.65V_E ; 1.1V_E >$ where $\mathbf{V_E}$ is the nominal (design) service speed of each ship.

## *Determination procedure of the long-term $R_{AV}$ prognosis*

The determination procedure of a single (for a given ship and its constant speed **V**) $\mathbf{R_{AV}}$ prediction, as described by expressions (3) and (4), was the following :

→ In the **first stage** the functional relations: $\mathbf{r_{AW} = f\,(\varpi|_{Fn})}$ were determined, where : $\mathbf{\varpi = \omega(L/g)^{1/2}}$ is a dimensionless wave frequency determined in a stationary system.
The relations were based on the results of model tests carried out with **1:85** scale ship models towed in calm water and on ca. 15 regular waves, whose frequencies **ϖ** were in the $\mathbf{\varpi \in (1.5 , 5.0)}$ range.

→ In the **second stage** the $\mathbf{R_{AW}}$ prognosis were calculated in accordance with expression (4).
The $\mathbf{R_{AW}}$ values were calculated for stationary waves (sea states) $\mathbf{c\,(T_1 , H_{1/3})}$ :

- whose $\mathbf{T_1}$ and $\mathbf{H_{1/3}}$ characteristics were taken from the $T_1 \in < 4s ; 10s >$ , $H_{1/3} \in < 0.5m ; 6.5m >$ ranges
- whose spectral density function $\mathbf{S\,(\omega, T_1, H_{1/3})}$, depending on sea region, is modelled by :
  - for **coastal waters** – the ISSC spectrum in the form :

$$S(\omega, T_1, H_{1/3}) = \frac{173 \cdot H_{1/3}^2}{T_1^4 \cdot \omega^{-5}} \cdot \exp\left(-B \cdot \omega^{-4}\right)$$

  - for **open waters** – the JONSWAP spectrum in the form proposed by the Author :

$$S_J(\omega, T_1, H_{1/3}) = \frac{1}{2} S(\omega, T_1, H_{1/3}) \cdot$$
$$\cdot\, 3.3^{\exp\left[-80\cdot|0.2069\cdot\omega\cdot T_1 - 1|^{2.5}\right]}$$

where : $\mathbf{S\,(\omega, T_1, H_{1/3})}$ is a model of the ISSC spectrum.

→ In the **third stage**, using expression (1), the $\mathbf{R_{AV}}$ prognosis were calculated.
The $\mathbf{R_{AV}}$ values for each ship and for each selected speed **V** were calculated by means of a **probabilistic model of all-year sailing conditions**, whose elements are :

- ship loading condition probability function **p**(**a**) presented in Tab.1
- sailing area probability function **p**(**b**) presented in Tab.2
- sea state probability function $\mathbf{p(c|b_1) = p(T_1, H_{1/3}|b_1)}$ for the NORTH SEA presented in Tab.3
- sea state probability function $\mathbf{p(c|b_1) = p(T_1, H_{1/3}|b_1)}$ for the BALTIC SEA presented in Tab.4.

***Table 1***

| **Loading condition $p(a_i)$** | | |
|---|---|---|
| light $\mathbf{a_1}$ | heavy $\mathbf{a_2}$ | Σ |
| 0.800 | 0.200 | 1.000 |

***Table 2***

| **Sea areas** | **North Sea** | **Baltic** | Σ |
|---|---|---|---|
| Restricted | 0.150 | 0.100 | 0.250 |
| Open | 0.450 | 0.300 | 0.750 |
| Σ | 0.600 | 0.400 | 1.000 |

***Table 3***

| **Heights $H_{1/3}$ [m]** | **Periods $T_1$ [s]** | | | | | | | Σ |
|---|---|---|---|---|---|---|---|---|
| | **4** | **5** | **6** | **7** | **8** | **9** | **10** | |
| **0.5** | 0.019 | 0.084 | 0.092 | 0.041 | 0.010 | 0.002 | - | 0.248 |
| **1.5** | 0.003 | 0.048 | 0.120 | 0.096 | 0.037 | 0.010 | 0.002 | 0.316 |
| **2.5** | 0.001 | 0.017 | 0.061 | 0.071 | 0.039 | 0.013 | 0.003 | 0.205 |
| **3.5** | - | 0.006 | 0.027 | 0.038 | 0.024 | 0.010 | 0.003 | 0.108 |
| **4.5** | - | 0.002 | 0.011 | 0.018 | 0.014 | 0.006 | 0.002 | 0.053 |
| **5.5** | - | 0.001 | 0.004 | 0.009 | 0.007 | 0.004 | 0.001 | 0.026 |
| **6.5** | - | - | 0.002 | 0.004 | 0.004 | 0.002 | 0001 | 0.013 |
| Σ | 0.023 | 0.158 | 0.317 | 0.277 | 0.135 | 0.047 | 0.012 | 0.969 |

***Table 4***

| **Heights $H_{1/3}$ [m]** | **Periods $T_1$ [s]** | | | | | | | **Σ** |
|---|---|---|---|---|---|---|---|---|
| | **4** | **5** | **6** | **7** | **8** | **9** | **10** | |
| **0.5** | 0.093 | 0.140 | 0.075 | 0.020 | 0.003 | - | - | 0.331 |
| **1.5** | 0.033 | 0.130 | 0.130 | 0.060 | 0.014 | 0.003 | 0.001 | 0.371 |
| **2.5** | 0.006 | 0.045 | 0.068 | 0.040 | 0.013 | 0.003 | 0.001 | 0.176 |
| **3.5** | 0.001 | 0.011 | 0.020 | 0.015 | 0.007 | 0.002 | - | 0.056 |
| **4.5** | - | 0.002 | 0.006 | 0.005 | 0.002 | 0.001 | - | 0.016 |
| **Σ** | 0.133 | 0.328 | 0.299 | 0.140 | 0.039 | 0.009 | 0.002 | 0.950 |

## *Ships*

The resistance increase predictions were determined for three ships, with their main characteristics presented in Tab.5.

The $\mathbf{R_{AV}}$ values were determined in accordance with procedure described at the page 31, and the $\mathbf{R_{TL}}$ and $\mathbf{R_{TC}}$ resistance values were determined from model test results recalculated for a real ship by the three-dimensional extrapolation method.

***Table 5***

| Ship characteristics | Symbol | Tanker | | Container carrier | | Ro - ro ship | |
|---|---|---|---|---|---|---|---|
| | | light loaded ship | heavy loaded ship | light loaded ship | heavy loaded ship | light loaded ship | heavy loaded ship |
| Length between perpendiculars | L [m] | 132.00 | | 132.00 | | 147.75 | |
| Length on waterline | $L_W$ [m] | 131.77 | 132.18 | 129.28 | 132.00 | 145.24 | 147.50 |
| Breadth | B [m] | 22.50 | | 22.50 | | 24.80 | |
| Draught | d [m] | 8.00 | 8.70 | 7.60 | 8.55 | 6.00 | 6.50 |
| Depth | D [m] | 12.80 | | 11.20 | | 14.00 | |
| Displacement | $\nabla$ [m$^3$] | 17512 | 19375 | 15138 | 17528 | 13899 | 15940 |
| Block coefficient | $C_B$ [-] | 0.737 | 0.750 | 0.671 | 0.690 | 0.632 | 0.650 |
| Coefficient of fineness | $C_W$ [-] | 0.884 | 0.906 | 0.845 | 0.869 | 0.855 | 0.868 |
| Service speed | $V_E$ [kn] | 14.00 | | 16.50 | | 19.50 | |
| Dimensionless longitudinal radius of gyration | $k_{yy}/L$ [-] | 0.250 | | 0.250 | | 0.250 | |

It can be seen that in the NORTH SEA :

- annual mean wave parameters are : $\mathbf{T_1}$ = 6.35 s ; $\mathbf{H_{1/3}}$ = 1.95 m
- calm water occurs with probability **p** = 0.031 i.e. approx. 11 days in a year.

It can be seen that in the BALTIC SEA :

- annual mean wave parameters are: $\mathbf{T_1}$ = 5.36 s ; $\mathbf{H_{1/3}}$ = 1.43 m
- calm water occurs with probability **p** = 0.050 i.e. approx. 18 days in a year.

## *Results of the calculations*

Table 6 presents the values of :

- mean still water resistance $\mathbf{R_T = 0.8R_{TL} + 0.2R_{TC}}$ (where $\mathbf{R_{TL}}$ and $\mathbf{R_{TC}}$ are still water resistance values in the light and heavy loading condition, respectively)
- long-term prognosis of resistance in waves $\mathbf{R_{AV}}$
- total mean resistance in waves $\mathbf{R_{TV} = R_T + R_{AV}}$
- respective relative increases : $\dfrac{R_{AV}}{R_T}$ and $\dfrac{R_{AV}}{R_{TV}}$

determined for 3 speeds **V** of each ship characterised in Tab.5.

$$\text{where : } C_B = \frac{\nabla}{L \cdot B \cdot d} \; ; \; C_W = \frac{S_W}{L \cdot B}$$

***Table 6***

| | Tanker | | | Container carrier | | | Ro - ro ship | | |
|---|---|---|---|---|---|---|---|---|---|
| **V [w]** | 9.8 | 12.6 | 15.4 | 11.5 | 14.2 | 16.9 | 12.5 | 16.4 | 20.3 |
| **Fn [-]** | 0.140 | 0.180 | 0.220 | 0.164 | 0.203 | 0.242 | 0.169 | 0.222 | 0.274 |
| $\mathbf{R_T}$ **[kN]** | 180 | 360 | 635 | 270 | 420 | 680 | 215 | 390 | 810 |
| $\mathbf{R_{AV}}$ **[kN]** | 56 | 76 | 83 | 57 | 65 | 72 | 46 | 50 | 54 |
| $\mathbf{R_{AV}}$ **[kN]** | 236 | 436 | 718 | 327 | 485 | 752 | 261 | 440 | 864 |
| $\mathbf{\frac{R_{AV}}{R_T}}$ **[%]** | 31 | 21 | 13 | 21 | 15 | 11 | 21 | 13 | 7 |
| $\mathbf{\frac{R_{AV}}{R_{TV}}}$ **[%]** | 24 | 17 | 12 | 17 | 13 | 10 | 18 | 11 | 6 |

## *Long-term prognosis of the relative increase of resistance in waves - wave service margin*

Figure presents a coefficient function $k = \dfrac{R_{AV}}{R_T} = f\,(Fn)$ built from the respective values taken from Tab.6 :

- **points** (different for different ships) mark the $k = \dfrac{R_{AV}}{R_T}$ values taken directly from Tab.6
- **full line** marks a universal function for the ships in question and their speeds, of the following mathematical form identified here :

$$k = \frac{R_{AV}}{R_T} = \frac{0.0635}{Fn} - 0.157 \tag{5}$$

$$\text{where : } Fn \in \; < 0.12 \; ; \; 0.30 >$$

The expression (5), with the assumption that :

$$Fn \equiv Fn_E = \frac{V_E}{\sqrt{g \cdot L}}$$

may also be treated as a certain model of a **dimensionless coefficient (k) of the wave service margin**, i.e. coefficient being a **component** (part) of the coefficient $\mathbf{k_W}$ [see expression (1)], which determines the ship resistance and/or horsepower margin balancing the expected (in the long-term prognosis sense) increase $\mathbf{R_{AV}(V_E)}$ [or $\mathbf{P_{AV}(V_E)}$] caused exclusively by the sea waves.

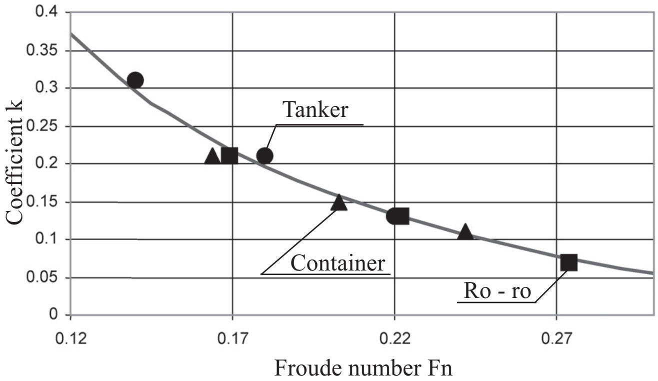


***Fig.*** *Expression* $k = \dfrac{R_{AV}}{R_T} = f\,(Fn)$

An alternative version of model (5) is the expression :

$$\boxed{k_1 = \frac{R_{AV}}{R_T} = 0.91 \cdot C_B - 0.50} \quad \text{for} \quad C_B \in < 0.50\,;\,0.85 > \tag{6}$$

where: $C_B = \frac{\nabla_E}{L \cdot B \cdot d_E}$ is the design block coefficient (corresponding to the design values of $\mathbf{\nabla_E}$ and $\mathbf{d_E}$ ).

Model (6) was identified here by substituting to model (5) an expression :

$$Fn_E = \frac{0.070}{C_B - 0.377}$$

devised by the Author and very close (qualitatively and quantitatively) to the well known Ayre's formula :

$$C_B = 1.075 - 1.68Fn_E$$

In order to illustrate the obtained results, Tab.7 presents percentage values of the **k** and $\mathbf{k_1}$ wave service margin coefficients, calculated :

- according to models (5) and (6)
- for ships characterised in Tab.5, where values of the $\mathbf{C_B}$ coefficients for those ships were determined as their long-term prognosis, i.e. as weighted averages of the $\mathbf{C_B}$ values corresponding to the light (80 %) and heavy (20 %) loading condition.

***Table 7***

| | **Tanker** | **Container carrier** | **Ro-ro ship** |
|---|---|---|---|
| $\mathbf{C_B}$ [-] | 0.740 | 0.675 | 0.636 |
| $\mathbf{V_E}$ [kn] | 14.00 | 16.50 | 19.50 |
| $\mathbf{Fn_E}$ [ - ] | 0.200 | 0.236 | 0.263 |
| **k** model (5) | **16 %** | **11%** | **8%** |
| $\mathbf{k_1}$ model (6) | **17%** | **11%** | **8%** |

## CONCLUDING REMARKS

It is fitting to underline the following aspects in the summary :

- The intention of this paper is to draw attention to the needs and possibilities of :
  - constant improvement of the quality and quantitative accuracy of the ship theory and design methods
  - more frequent application, in ship theory and in practical ship design, of the useful but still little used models of ship hydromechanics - the **seagoing qualities** in particular.
- It was particularly intended to remind that such an important design problem as prediction of optimum ship resistance and/or horsepower in real operating conditions still remains a **problem waiting for solution**.
- An important objective of the work was also to indicate a possibility of identifying more justifiable service margin formulae based on **generalisation of long-term prognosis of the relative additional ship resistance in waves**.
- The identified models (5) and (6) do not aspire to be complete and directly applicable design service margin formulae, as they :
  - are based on too scarce research material, and also
  - are modelling only the wave part of the problem.
- The above mentioned limitation of models (5) and (6) is to some degree compensated by the fact that their results are an **upper estimation** of the wave service margin. They were identified :
  - only for the **head sea conditions** when the resistance increase is the greatest
  - without taking into account the **voluntary speed (and horsepower) reduction** and/or change of the ship course in order to protect the ship against too intense (dangerous) motions - accelerations, vibrations, slamming, shipping of water etc.
- The form of models (5) and (6) shows clearly that the **k** coefficient depends either on the ship **design speed** $\mathbf{V_E}$ or on the **design values of block coefficient** $\mathbf{C_B}$ :
  - it is inversely proportional to $\mathbf{V_E}$
  - it is directly proportional to $\mathbf{C_B}$.

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a) Author of the sentence unknown

b) There is a strong conviction that the work of research and practical sciences may be appreciated only when it is concentrated on fulfilling the existing and not potential design needs

c) - „superposition principle" published in 1953 by St. Denis and Pierson and its application to the resistance-propulsion problems published in 1967 by Vassilopoulos
- standard spectral (spectral density function) wave models, e.g. the Pierson-Moskowitz model recommended in 1969 by ITTC
- atlases of annual wave distributions in the main shipping sea areas, e.g. the „Ocean Wave Statistics" atlas published in 1967 by Hogben and Lamb

d) The problem is that the service margin is meant to be a remedy not only for the increased ship resistance (horsepower) due to sea-way but also due to the uncontrolled and gradually increasing hull roughness. There is no doubt that with totally welded hull shell (and constantly improved welding methods) and with a constant progress in the quality, durability and anti-fouling effects of paint coatings, also the „roughness" component of the service margin should be revised (reduced)

# The predictions of resistance and propulsion for 4 types of ships

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ABSTRACT

*In this paper the predictions of resistance and propulsion for 4 types of ships: container carrier, oil product tanker, ro-ro vessel and river-sea vessel included in the EUREKA „Baltecologicalship" project E!2772 are presented. All the above ships were designed by the SINUS Enterprise Ltd. The scope of the research covers: experimental measurement of ship resistance in calm water, determination of resistance increment in irregular waves and computational powering prediction for POD propulsors. The experiments were conducted in the Laboratory of the Department of Underwater Technology, Hydromechanics and Design of Ships, Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology.*

**Keywords** : model tests, ship resistance in calm water, ship resistance in irregular waves, POD propulsors

## INTRODUCTION

Trustworthy determination of the resistance characteristics of the designed ships through model experiments is necessary for confirmation of the hydrodynamic quality of design. Determination of the full scale resistance enables realistic prediction of the propulsive characteristics of ships, assessment of the delivered power and propulsive parameters. Minimization of the engine power and ensuring correct conditions for engine-propeller interaction is an important problem both from the economic (operational cost) and ecological (level of pollution) points of view. The above mentioned problems refer to the task "Resistance Model Experiments for 4 new types of ship", included in the EUREKA project E!2772. The Project covers the following ship types :

- Container carrier
- Oil product tanker
- Ro-ro vessel
- River – sea vessel.

All the above ships were designed by the SINUS Enterprise. The scope of the research covers :

- experimental measurement of ship resistance in calm water
- experimental determination of resistance increment in irregular waves
- analysis of the ship-induced wave system
- computational powering prediction for POD propulsors.

The experiments were conducted in the Laboratory of the Department of Underwater Technology, Hydromechanics and Design of Ships, Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology.

## RESULTS OF EXPERIMENTS

The ship hull models for experiments were built in the model workshop on the basis of the theoretical hull lines supplied by SINUS. Resistance experiments were performed in the towing tank and visualisations of the flow on the hull were made in the circulating water channel. Ship models were tested for two displacements: design draught and scantlings draught. The resistance prediction for full scale was performed using three-dimensional extrapolation with form factor (1+k) calculated according to the ITTC-72 formula. The approximate computational powering prediction was based on experimentally determined resistance, on the propeller-hull interaction parameters determined by the Holtrop method, on the design POD propulsor data supplied by SINUS and on propeller design software based on lifting line/lifting surface theory. Prediction of ship resistance in irregular waves was done for the Baltic and North Sea wave spectrum. The results of all experiments and computations were the basis for assessment of the hydrodynamic quality of each ship design.

### *Container carrier – SINE 202*

The basic parameters of the container ship SINE 202 are as follows :

| | |
|---|---|
| ➢ Length overall | 138.10 m |
| ➢ Length between perpendiculars | 132.00 m |
| ➢ Breadth | 22.50 m |
| ➢ Depth | 11.20 m |
| ➢ Design draught | 7.60 m |
| ➢ Scantlings draught | 8.55 m |

The hull geometry is presented in the form of body lines in Fig.1. The model of the container ship was manufactured to the scale 1: 84.5. The resistance characteristics obtained for design draught and scantlings draught, recalculated into full scale are shown in Fig.2. The calculated effective engine power curves for both draughts are presented in Fig.3.

On the basis of experimental results it may be concluded that the ship will reach the speed of 17.80 knots on trials at design draught, 17.20 knots in service at design draught and 16.25 knots in service at scantlings draught. Further increase of ship speed by 0.2÷0.3 knots may be possible by increasing the propeller diameter by about 10%.

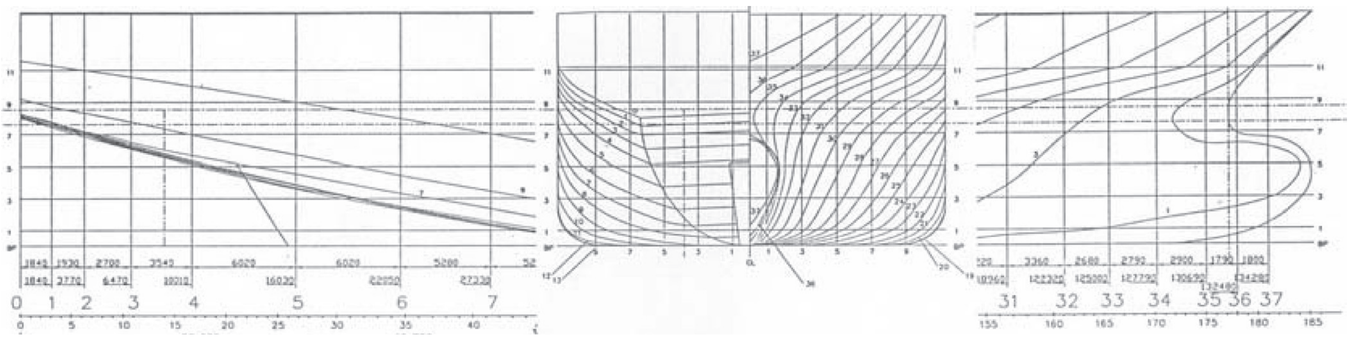

*Fig. 1. Body plan of the container carrier SINE 202*

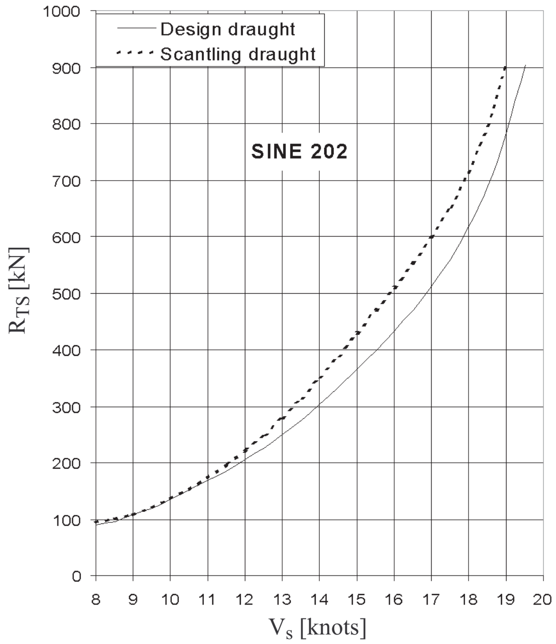


*Fig. 2. Total resistance of the container carrier SINE 202*

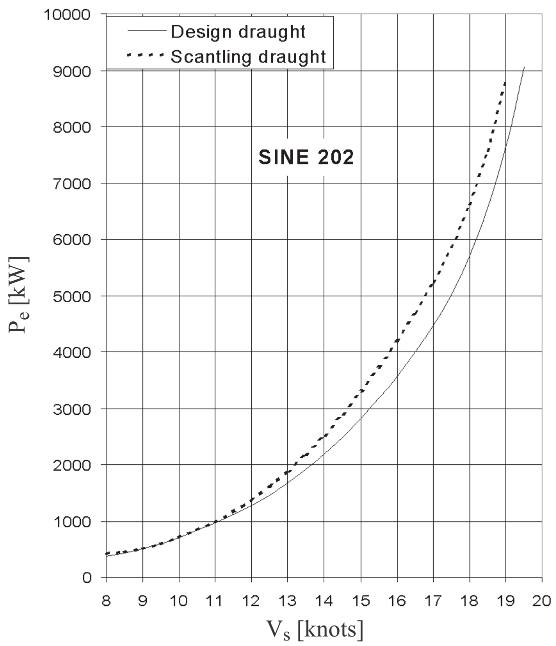


*Fig. 3. Effective engine power of the container carrier SINE 202*

It is assessed that the container ship SINE 202 has good sea-keeping quality. The increment of resistance in waves is relatively low both in the sense of long-term predictions (Fig.4) and in the sense of short-term predictions (Fig.5). It has been determined that the sea margin of 15%, applied in propulsive prediction, is correct for typical sea states in the Baltic and North Sea. Additionally, it has been determined that the resistance increment in waves is identical for the design and scantlings draught.

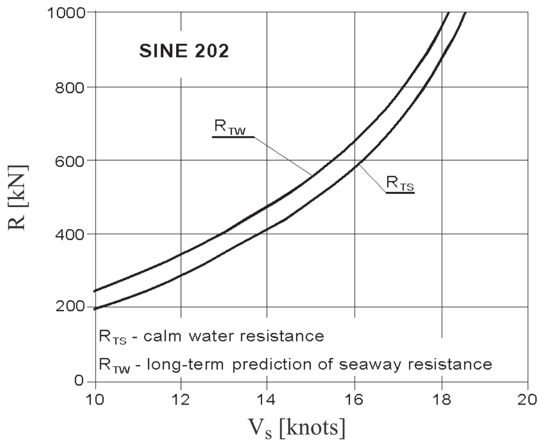


*Fig. 4. Resistance characteristics of the container carrier SINE 202*

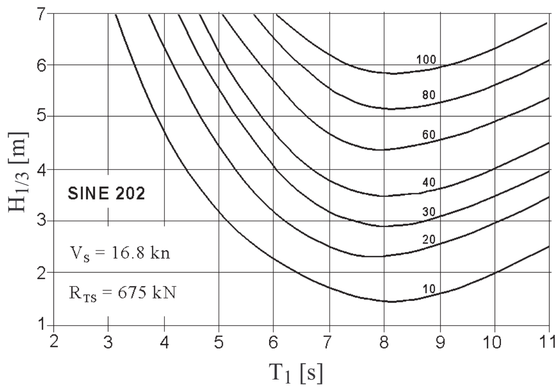


*Fig. 5. Relative added resistance* $\left(\frac{R_{AW}}{R_{TS}}[\%]\right)$ *contours in the* $T_1 - H_{1/3}$ *plane for SINE 202 ship. (set of the short–term predictions of the ship added resistance RAW)*

The detailed results of experiments and computational analyses are included in the report [1 , 2].

## *Oil product tanker - SINE 203*

The basic parameters of the oil product tanker are as follows :

| | |
|---|---|
| ➢ Length overall | 138.10 m |
| ➢ Length between perpendiculars | 132.00 m |
| ➢ Breadth | 22.50 m |
| ➢ Depth | 12.80 m |
| ➢ Design draught | 8.00 m |
| ➢ Scantlings draught | 8.70 m |

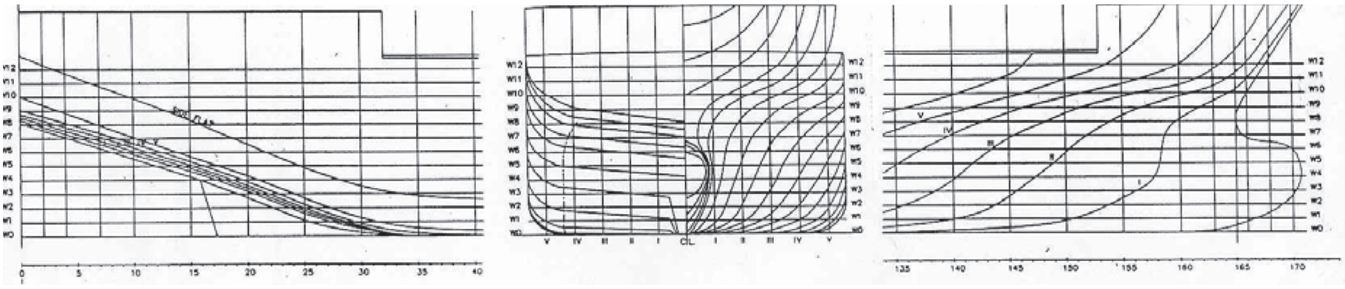

***Fig. 6.*** *Body plan of the oil product tanker SINE 203*

The hull geometry is presented in the form of body lines in Fig.6. The ship model was manufactured to the scale 1 : 84.5. The resistance characteristics obtained experimentally for design and scantlings draughts respectively are shown in Fig.7. The calculated effective engine power curves for both draughts are presented in Fig.8.

On the basis of experimental results it may be concluded that the ship will reach the speed of 12.80 knots in trials condition, 12.25 knots in service condition at design draught and 12.05 knots in service at scantlings draught.

Model experiments of resistance in waves have shown that the SINE 203 model has a small increase of resistance in waves both in the sense of a long-term prediction Fig.9 and in the sense of a short-term prediction Fig.10 for sea states typical of the Baltic and North Sea.

The detailed results of experiments and computational analyses are included in the report [3].

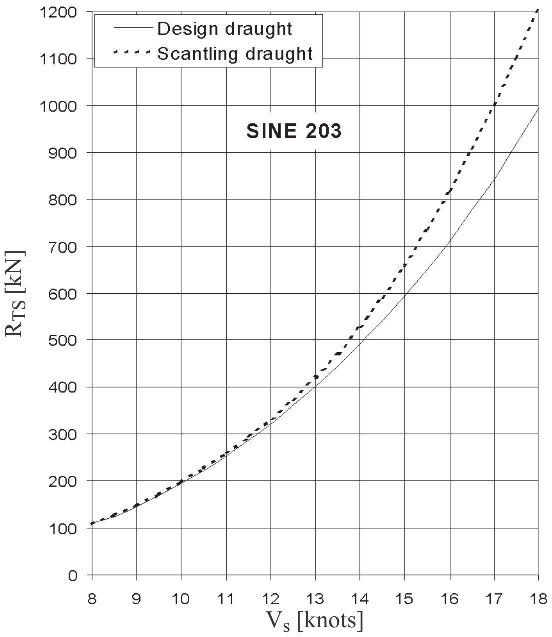


***Fig. 7.*** *Total resistance of the oil product tanker SINE 203*

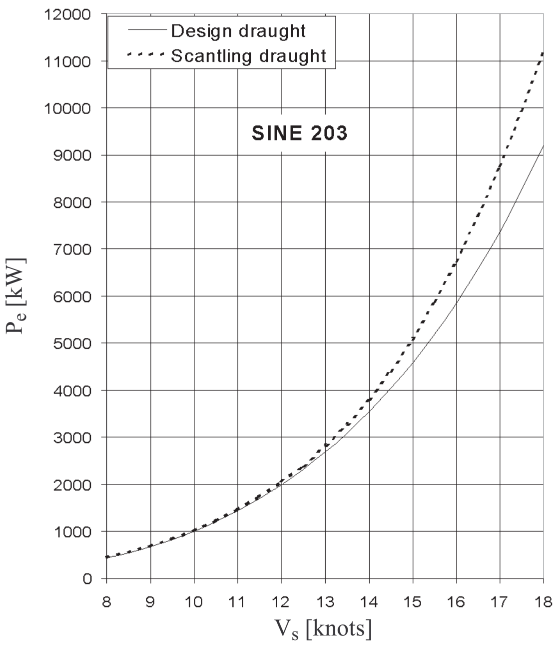


***Fig. 8.*** *Effective engine power of the oil product tanker SINE 203*

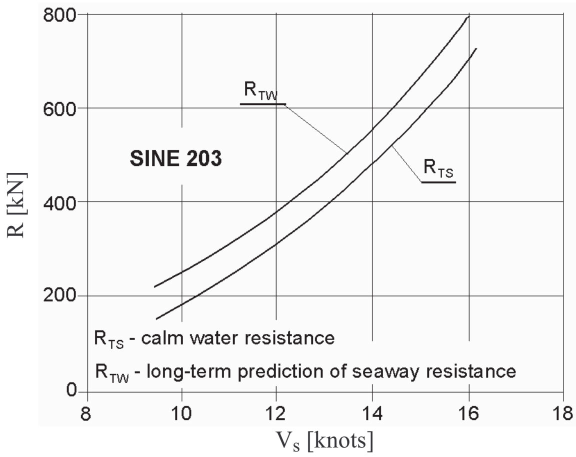


***Fig. 9.*** *Resistance characteristics of the oil product tanker SINE 203*

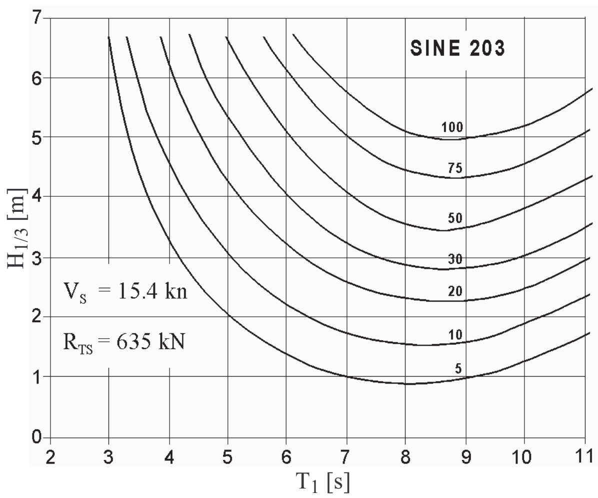


***Fig. 10.*** *Relative added resistance* $\left(\frac{R_{AW}}{R_{TS}}[\%]\right)$ *contours in the* $T_1 - H_{1/3}$ *plane for SINE 203 vessel. (set of the short–term predictions of the ship added resistance* $R_{AW}$*)*

## *Ro-ro vessel - SINE 204*

The basic parameters of the Ro-Ro vessel are as follows :

| | |
|---|---|
| ➲ Length overall | 156.72 m |
| ➲ Length between perpendiculars | 147.75 m |
| ➲ Breadth | 24.80 m |

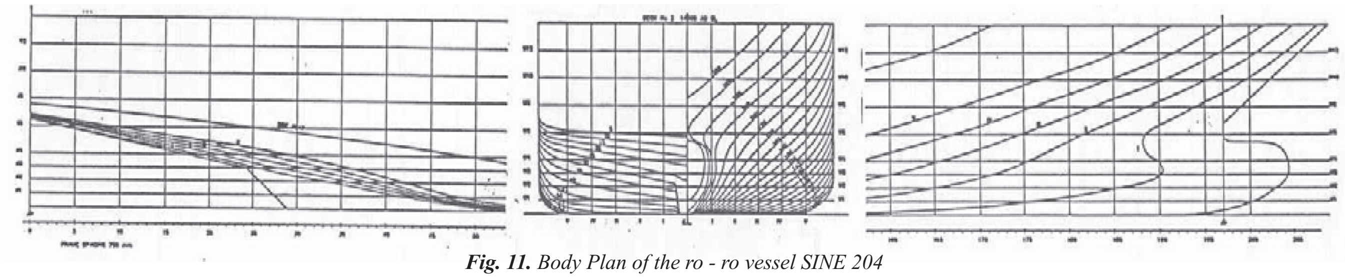

*Fig. 11. Body Plan of the ro - ro vessel SINE 204*

- Design draught 6.00 m
- Scantlings draught 6.50 m

The hull geometry is presented in the form of body lines in Fig.11. The ship model was manufactured to the scale 1:100. The resistance characteristics obtained experimentally for design and scantlings draught respectively are shown in Fig.12. The calculated effective engine power curves for both draughts are presented in Fig.13.

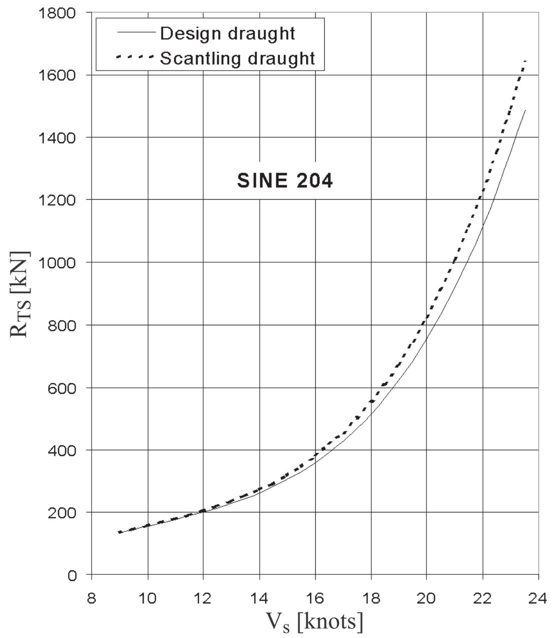


*Fig. 12. Total resistance of the ro – ro vessel SINE 204*

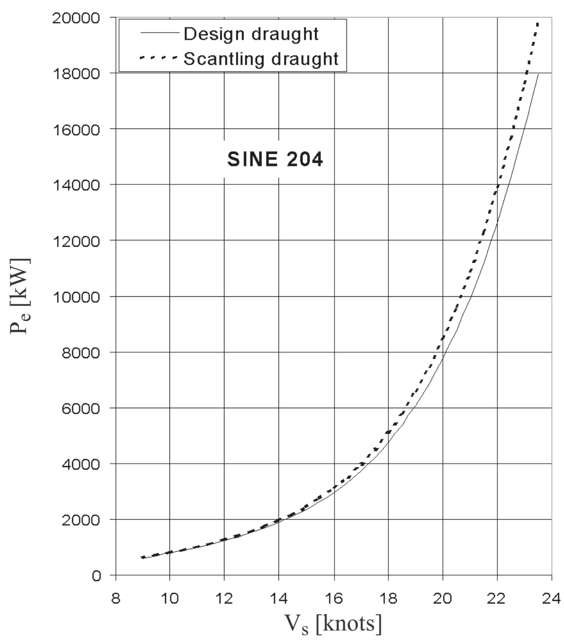


*Fig. 13. Effective engine power of the ro - ro vessel SINE 204*

On the basis of experimental results it may be concluded that the ship has favourable resistance characteristics. The planned value of delivered power is sufficient for attaining the planned design speed of 20.0 knots. A minor power deficit for scantlings draught with sea margin is meaningless from the practical point of view. The ship will reach the speed of 20.92 knots in trials condition, 20.45 knots in service condition at design draught and 19.90 knots in service at scantlings draught.

Model experiments of resistance in waves have shown that the resistance increment in waves is relatively low both in the sense of a long term prediction Fig.14 and in the sense of a short term prediction (Fig.15). It has been concluded that the sea margin of 15%, adopted in the propulsion analysis, is correct for sea states typical of the Baltic and North Sea.

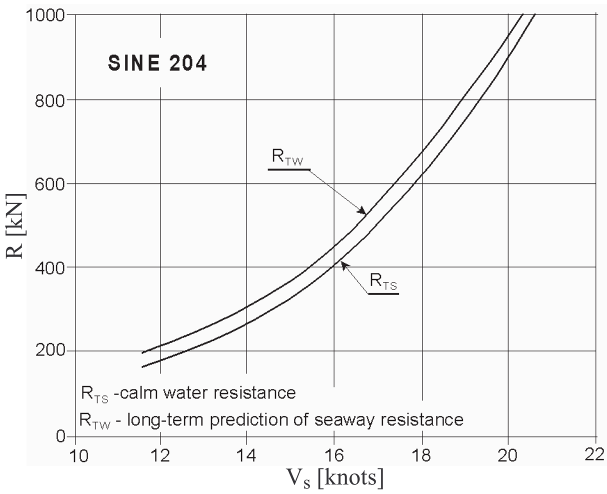


*Fig. 14. Resistance characteristics of the ro - ro vessel SINE 204*

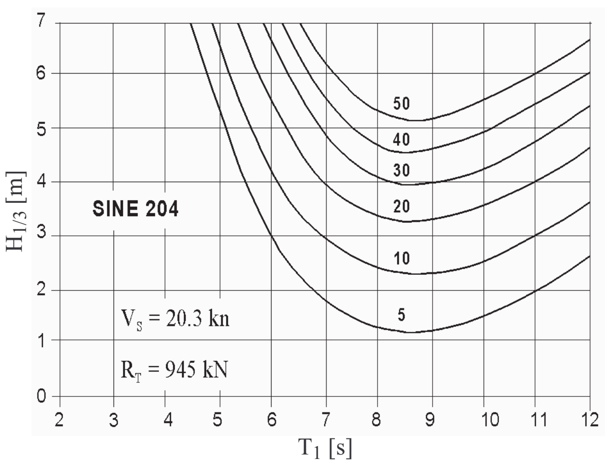


*Fig.15. Relative added resistance* $\left(\frac{R_{AW}}{R_{TS}}[\%]\right)$ *contours in the* $T_1 - H_{1/3}$ *plane for SINE 204 vessel. (set of the short–term predictions of the ship added resistance* $R_{AW}$*)*

The detailed results of experiments and computational analyses are included in the report [4].

## *River-sea vessel - SINE 205*

The basic parameters of the River-sea vessel are as follows :

- Length overall 89.45 m
- Length between perpendiculars 87.47 m
- Breadth 11.40 m
- Depth 5.45 m
- Sea draught 4.40 m
- River draught 2.80 m

The hull geometry is presented in the form of body lines in Fig.16. The ship model was manufactured to the scale 1: 50. The resistance characteristics obtained experimentally for sea and river draughts respectively are shown in Fig.17. The calculated effective engine power curves for both draughts are presented in Fig.18.

On the basis of experimental results it may be concluded that the ship will reach the speed of 11.40 knots in trials condition at sea draught, 10.90 knots in service condition at sea draught and 12.00 knots in trials at river draught.

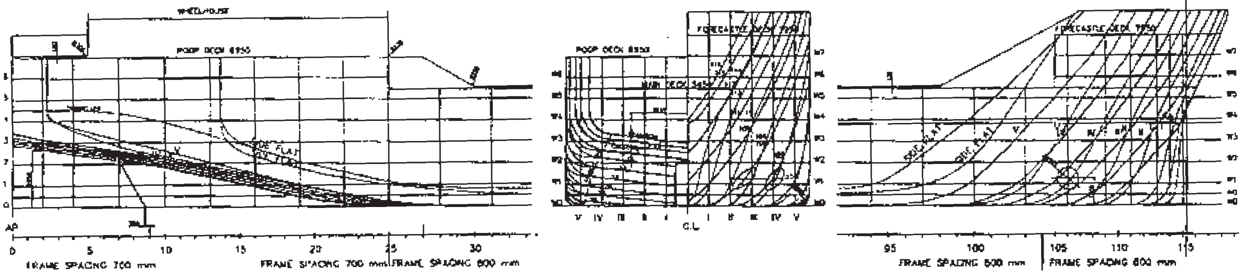

*Fig. 16. Body plan of the river - sea vessel SINE 205*

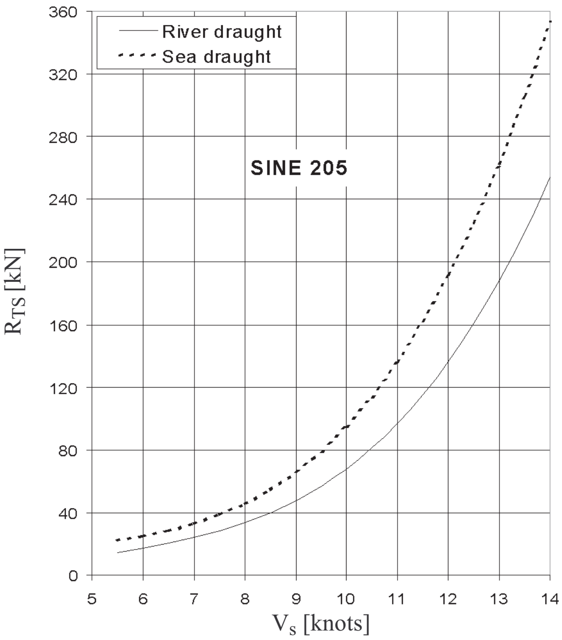


***Fig. 17.** Total resistance of the river - sea vessel SINE 205*

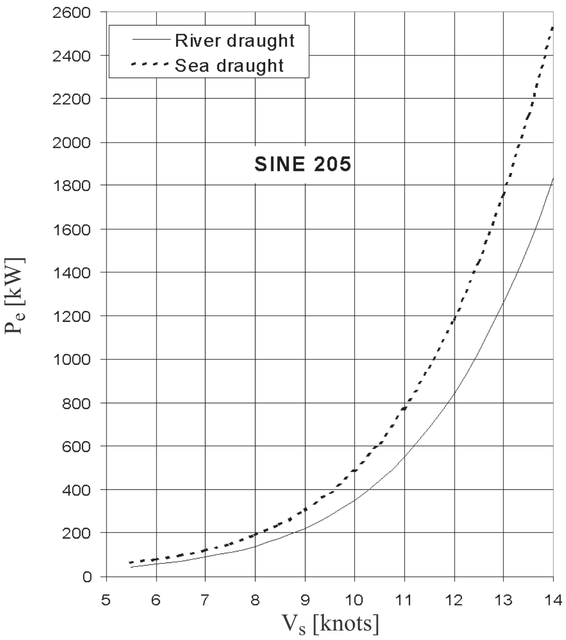


*Fig. 18. Effective engine power of the river - sea vessel SINE 205*

The detailed results of experiments and computational analyses are included in the report [5].

## CONCLUSIONS

The above presented predictions of resistance and propulsion for 4 types of ship obtained through model experiments enable realistic prediction of the power and propulsive characteristics. In the cases where the performance required by the EUREKA project was not achieved, the directions for necessary modification were indicated. The positive results of experimental investigations demonstrate that the shape of the hull geometry and basic parameters were well selected and they create a possibility of ship design with favourable hydrodynamic characteristics.

### NOMENCLATURE

$H_{1/3}$ - significant wave height [m]
$P_e$ - effective engine power [kW]
$R_{TS}$ - total resistance of ship [kN]
$R_{TW}$ - long-term prediction of seaway resistance [kN]
$T_1$ - characteristic wave period [s]
$v_S$ - ship speed [knots]

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# Stability criteria as constraints in a fleet of ships optimisation problem

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ABSTRACT

*The paper has been written within the European EUREKA Project E!2772, initiated and completed at the Faculty of Ocean Engineering & Ship Technology, Gdańsk University of Technology in the years 2001-2003. A problem has been solved concerning mathematical optimisation of a fleet of multipurpose sea-river vessels for European short-shipping regular lines, in the area of The North and Baltic Seas, on the level of marine transportation task, by the non-linear programming methods with constraints. A method is proposed which enables existing criteria of stability to be included as constraints in the optimisation model of a fleet. In the numerical examples, three typical criteria of intact stability: by IMO, PRS, and HSMB have been selected to demonstrate a post-optimisation feasibility analysis of principal parameters of ships.*

**Keywords** : maritime transportation, computer-aided ship design, optimisation, intact stability criteria

## 1. INTRODUCTION

Computer aided ship design methods used at present, while offering automation of the design process, require its rationalisation and formalisation. In consequence, adequate mathematical models of the design object must be created which affect the design process by introducing a structure and terminology which unavoidably bounds reasoning to the terms of the model.

In this case a fleet of ships at the stage of owner's study is assumed to be an object and the task of optimising its main parameters is an objective of the adequate mathematical model. In consequence, the global structure of the model (further called an "optimisation model") corresponds to that proposed by operational research methods in general and non-linear programming methods (NPM) in particular [3]. Within this structure, optimisation models consist of a set of sub-models of particular properties of the object which have been recognised as significant to the predictive features of the model. Optimisation models applied to fleet/ship design are definitely synthetic in nature. This feature requires the analytical representation of particular sub-models to be relatively simple. In consequence, sub-models usually neither become isomorphic with, nor conform to the physical structure of that part of object to which they are related. Such type of models is sometimes referred to as "non-structural" [19]. NPM require for all the sub-models concerned to be formulated as constraints. Among them there are always those concerning safety of an object. In ship design, a special interest in this group is focused on the stability of ships. In naval architecture today, the stability requirements are imposed in the form of legal regulations by such institutions as IMO, classification societies, governmental organisations and other bodies. An essential part of stability regulations are stability criteria.

The paper deals with the problem of incorporating stability criteria as constraints in the optimisation model of fleet/ship design. At the initial stages of the design the principal difficulty is that the full geometry of a hull, necessary for the stability criteria to be applied, is usually unknown. A standard solution was to take into account the initial stability only, represented by the initial metacentric height GM0 [2], [4], [7], [12], [14]. The paper proposes an alternative approach, based on an idea introduced by Wiśniewski [20] and developed by Kupras [10], [11]. In this concept the full stability of ship can be accounted for by using systematic standard series of hull forms, following the methodology developed in ship resistance and power prediction.

In order to accomplish the task, an attempt has been made to define all the stability-related geometrical characteristics of a ship analytically, based on the Series 60 body forms [19]. In consequence, an arbitrary criterion of intact stability can also be defined in an analytical way and so incorporated into optimisation model as a constraint.

Stability aspects in the computer-aided modelling of ship design have been addressed on the background of the optimisation problem concerning a fleet of multipurpose sea-river vessels for European short-shipping regular lines, in the area of The North and Baltic Seas, on the level of marine transportation task, by non-linear programming methods with constraints. The problem has been undertaken within the European EUREKA Project [13] based on predictions that a significant increase of cargo transportation in Europe over the next 10 years (or probably after this period) will take place between Western Europe and the Central and East European countries.

In the numerical examples, three criteria of intact stability: IMO [6], HSMB [5], and PRS [17] have been selected, as typical of contemporary stability regulations, to demonstrate the method in a post-optimisation, feasibility analysis of principal parameters of ships.

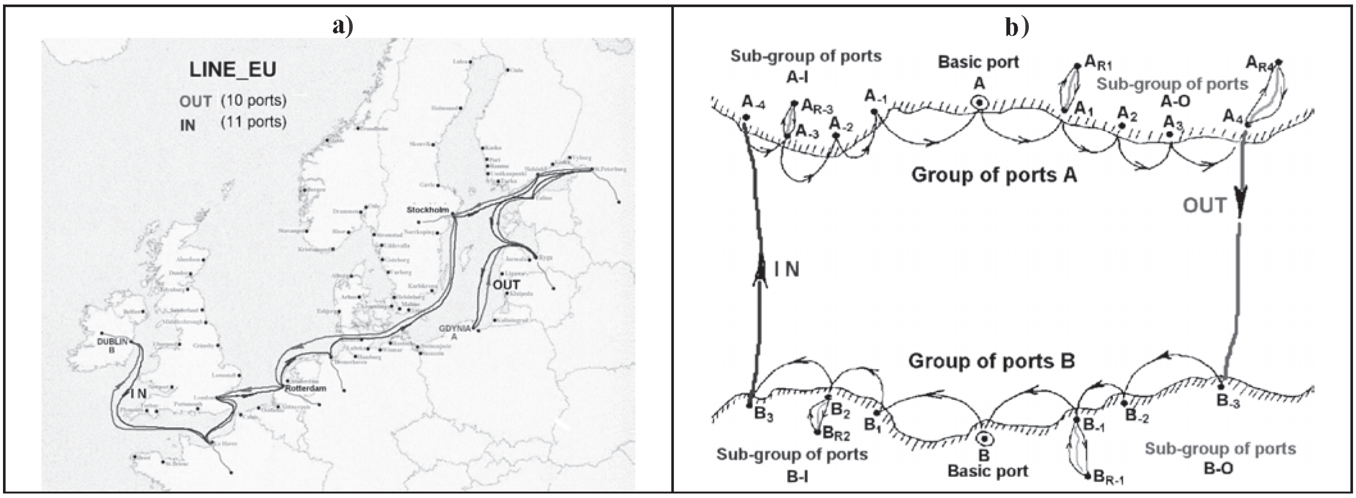
## 2. PROBLEM STATEMENT

A fleet of ships consists of a number of homogeneous ships operating as a maritime transportation system in a certain environment. The transportation task for a fleet of ships is to carry goods between ports during a prescribed period of time. An optimum fleet to perform this task, given the particular (owner's) data, is a general problem under discussion. A solution to this problem needs adequate functional and mathematical models.

## 2.1. ***Functional model of a fleet***

In the particular case (Tab.2.1), a (potential) shipping line connects the furthest Western and Eastern regions of Europe (**a**). A corresponding model of shipping (**b**) is called a multi-port route model linking two areas of operation A and B with the two groups of clustered sea and hinterland river ports. There are two streams of goods transportation in the model: from A to B (called OUT) and back, from B to A (called IN). Ports A-0 and B-0 are the home and destination ports. For more details about the functional model of a fleet - see [13].

***Tab. 2.1.*** *An example of a shipping line and its graphical model*



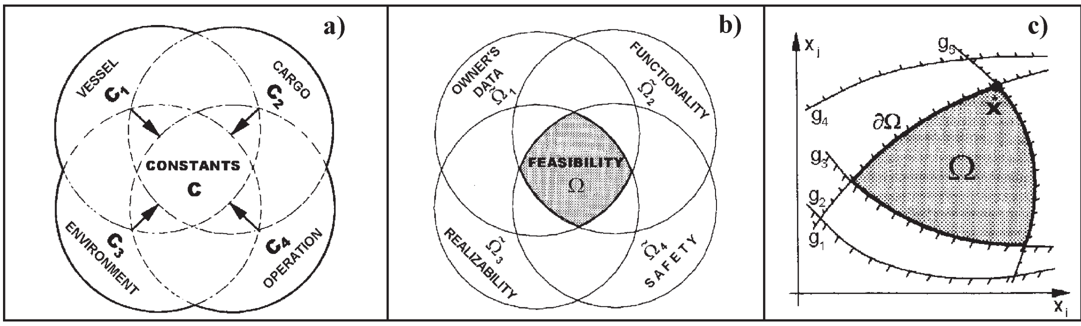

## 2.2. ***Optimisation model of a fleet***

The mathematical model chosen for the fleet optimisation problem can be described as deterministic, static, continuous, single level and single objective model, imposed and solved by non-linear programming methods. A standard formulation of the model, within NPM, is as follows: given a vector **c** (or a set **C**) of constants, find such a vector of decision variables **x** that minimises a single valued objective function Q(**x**,**c**) subject to a set of inequality constraints. An adequate mathematical form of the problem is :

$$\begin{cases} \min Q(\mathbf{x};\mathbf{c}); \quad Q: \mathbf{R}^n \times \mathbf{R}^N \to \mathbf{R}^1 \\ (\mathbf{x};\mathbf{c}) \in \mathbf{\Omega} = \left\{ (\mathbf{x};\mathbf{c}) : g_j(\mathbf{x};\mathbf{c}) \le 0 \quad {\scriptstyle j = 1,2,\ldots, m} \right\} \ne \varnothing \end{cases} \tag{2.1}$$

It is generally assumed that $Q(\cdot)$ and $g_j(\cdot)$ are all non-linear functions.
The conditions for existence and uniqueness of the (optimum) solution to (2.1) can be found in [3].

***Tab. 2.2.*** *Elements of an optimisation model of a fleet of ships*




$\mathbf{J}_B = \{2,3,5\}$

$\mathbf{J}_B^a = \{2,5\}$

$\mathbf{J}_B^p = \{3\}$

It is obvious that the optimum solution $\mathbf{x}^*$ of the problem (2.1) is, in fact, parametrised by *constants* **C** that can be classified according to different criteria (Tab.2.2a). In particular, the group $\mathbf{C}_3$ delivers constants for the stability criteria of a ship (legal environment). A crucial element of the optimisation model is a *feasible solution region* (FSR) **Ω**, because it eventually decides about an optimum solution to the problem. FSR is such a set of pairs (**x**,**c**) that all the constraints hold.

$$\mathbf{\Omega} = \{(\mathbf{x},\mathbf{c}) : g_j(\mathbf{x},\mathbf{c}) \le 0 \;, j \in \mathbf{J}\} \;, \mathbf{J} = \{1,2,...m\} \tag{2.2}$$

For further discussion, it is useful to classify FSR from the functional and formal points of view. In the *functional* classification, FSR defines, in fact, the notion of feasibility in ship design. Formally it can be thought of as a common part of the four groups of requirements imposed on a fleet/vessel by the environment (Tab.2.2b) and can then be written as the following product of four sets :

$$\Omega = \bigcap_{k=1}^{4} \Omega_k$$

All of them affect constraints of the model. For example, the SAFETY ($\tilde{\Omega}_4$) group contains, among other things, design restrictions concerning stability regulations.

In (2.2), an individual constraint is an inequality-type relation in which a function $g_j(\cdot)$ expresses a balance between certain (dependent) parameters of an object. In order to keep the same standard relation of inequality ($\leq$) for all constraints, the general form of $g_j(\cdot)$ has to be alternatively :

$$g_j(\mathbf{x};\mathbf{c}) = \begin{cases} p_j(\mathbf{x};\mathbf{c}) \;-\; \tilde{p}_j(\mathbf{x};\mathbf{c}) & \text{(a)} \\ \qquad\qquad \text{or} & \\ \tilde{p}_j(\mathbf{x};\mathbf{c}) \;-\; p_j(\mathbf{x};\mathbf{c}) & \text{(b)} \end{cases} \qquad j \in \mathbf{J} \tag{2.3}$$

where :

the $p_j(\cdot)$ parameters are those predicted by the model and $\tilde{p}_j(\cdot)$ are the corresponding ones required by $\tilde{\Omega}_k$.

In most cases the $\tilde{p}_j$ parameters are just constant figures such that $\tilde{p}_j = c_j \in \mathbf{C}$.

In the *formal* classification of FSR, attention will be focused on constraints forming the boundary of $\Omega$.

Figure in Tab.2.2c illustrates the problem. Let us define an FSR corresponding to a single $j^{th}$ constraints :

$$\Omega_j = \{(x;c) : g_j(x;c) \leq 0\}, \;\; j \in J$$

Its boundary is then:

$$\partial\Omega_j = \{(\mathbf{x};\mathbf{c}) : g_j(\mathbf{x};\mathbf{c}) = 0\}, \;\; j \in \mathbf{J}$$

For the whole feasible region one has, of course : $\mathbf{\Omega} = \bigcap \mathbf{\Omega}_j \;,\;\; j \in \mathbf{J}$

Similar relation does not, however, hold for boundaries :

$$\partial\mathbf{\Omega} \neq \bigcap \partial\mathbf{\Omega}_j, \neq \bigcup \partial\mathbf{\Omega}_j \quad \text{but} \quad \partial\mathbf{\Omega} \subset \bigcup \partial\mathbf{\Omega}_j, \; j \in \mathbf{J} \tag{2.4}$$

A $j^{th}$ constraint is called a *boundary constraint* if its own boundary contributes to the boundary of $\Omega$, otherwise the constraint is a *non-boundary constraint*. It is obvious that, in fact, the boundary constraints are those which determine $\Omega$ and, as such, affect the optimum solution of the problem :

$$\mathbf{\Omega} = \bigcap_{j \in \mathbf{J}} \mathbf{\Omega}_j = \bigcap_{j \in \mathbf{J}_B \subset \mathbf{J}} \mathbf{\Omega}_j \tag{2.5}$$

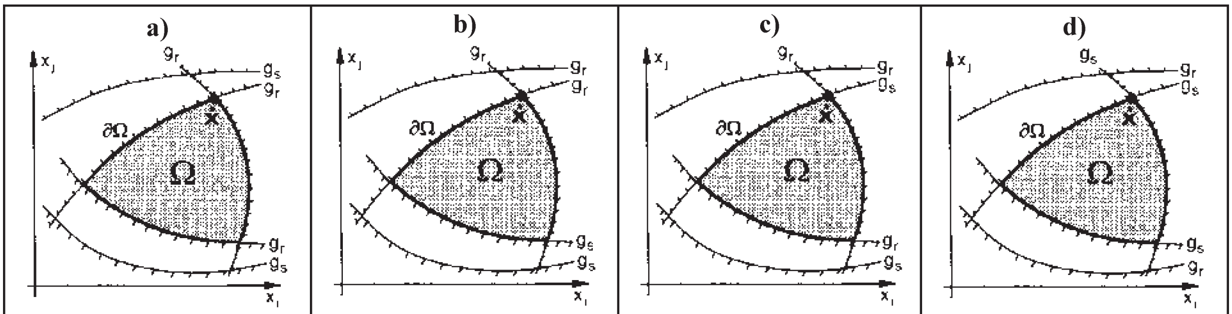
Relation (2.5) suggests an apparently equivalent formulation of (2.1) with the smaller number of constraints (boundary constraints only). Unfortunately, this is not the case because the set of boundary constraints $\mathbf{J}_B$ is not known in advance. Among the boundary constraints one can further distinguish *active* constraints and *passive* constraints. The classification refers to the location of the optimum $\mathbf{x}^*$ solution in $\mathbf{\Omega}$, which exclusively depends on the objective function $\mathbf{\Omega}(\cdot)$ used in the optimisation process. A $j^{th}$ constraint is active in $\mathbf{x}^*$ if $\mathbf{x}^* \in \partial\mathbf{\Omega}j$ ($\Rightarrow j \in \mathbf{J}_B^a$), otherwise the constraint is passive ($\Rightarrow j \in \mathbf{J}_B^p = \mathbf{J}_B \setminus \mathbf{J}_B^a$). If the optimum solution belongs to the interior of FSR ($\mathbf{x}^* \in \mathrm{int}\mathbf{\Omega}$), all the boundary constraints are passive and no constraint affects $\mathbf{x}^*$.

Let us now introduce another classification of $\mathbf{\Omega}$ and the corresponding constraints : *stability* constraints vs. *non-stability* (remaining) constraints.

$$\mathbf{\Omega} = \mathbf{\Omega}_S \cap \mathbf{\Omega}_R \quad \Leftrightarrow \quad \mathbf{J} = \mathbf{J}_S \cup \mathbf{J}_R \tag{2.6}$$

where : $\mathbf{\Omega}_S$ is a feasible solution region with regard to stability requirements, $\mathbf{J}_S$ is a set of indices of stability constraints, $\mathbf{\Omega}_R$, $\mathbf{J}_R$, correspond to the remaining constraints accordingly. The current classification is independent of the previous one; i.e. both stability and remaining constraints can become the boundary or non-boundary constraints. Let us investigate the status of stability constraints via the relations: $\mathbf{\Omega}_S$ vs. $\mathbf{\Omega}$, $\partial\mathbf{\Omega}_S$ vs. $\partial\mathbf{\Omega}_S$, and $\mathbf{J}_S$ vs. $\mathbf{J}_B$. It is obvious that the relation $\mathbf{\Omega}_S \supseteq \mathbf{\Omega}$ is always satisfied, but similar relation does not occur for boundaries. One can distinguish here four cases (see Tab.2.3).

***Tab. 2.3.** Stability constraints ($g_S$) vs. remaining constraints ($g_R$)*




| a) | b) | c) | d) |
|---|---|---|---|
| $\Omega \subset \Omega_S$ | $\Omega \subset \Omega_S$ | $\Omega \subset \Omega_S$ | $\Omega = \Omega_S$ |
| $\partial\Omega_S \cap \partial\Omega_B = \varnothing$ | $\partial\Omega_S \subset \partial\Omega$ | $\partial\Omega_S \subset \partial\Omega$ | $\partial\Omega_S = \partial\Omega$ |
| $\mathbf{J}_S \cap \mathbf{J}_B = \varnothing$ | $\mathbf{J}_S \in \mathbf{J}_B^p$ | $\mathbf{J}_S \in \mathbf{J}_B^a$ | $\mathbf{J}_S = \mathbf{J}_B$ |

ad. a) Stability constraints do not contribute to the FSR, so, for given stability regulations and a set of constants **C**, they are totally insignificant, regardless of the objective function Q.

ad. b) Stability constraints do contribute to the FSR, so, for given stability regulations and a set of constants **C**, they are currently insignificant but potentially significant, dependent on the objective function Q.
ad. c) Stability constraints do contribute to the FSR, so, for given stability regulations and a set of constants **C**, they are currently significant, but potentially insignificant, dependent on the objective function Q.
ad. d) Stability constraints form the boundary of the FSR, so, for given stability regulations and a set of constants **C**, they are currently and potentially significant, regardless of the objective function Q.

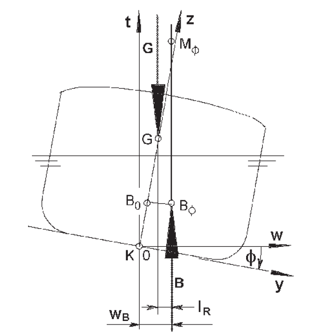
The foregoing discussion will be recalled in Chapter 5 to verify the status of stability constraints in the fleet optimisation model.

## 3. IDENTIFICATION OF INTACT STABILITY CRITERIA

In order to illustrate the thesis of the paper, three criteria of intact stability have been chosen, as recommended by the following institutions: International Maritime Organisation (IMO [6]), Polish Register of Shipping (PRS, [17]), and Hamburg Ship Model Basin (HSMB [5]). In the light of the discussion in [8], all these criteria can be regarded as a present day standard as far as intact stability regulations in naval architecture are concerned.

Tab.3.1 introduces the coordinate systems as well as basic notions and terminology which will be in use throughout the paper. In particular, (Y-Z) and (y-z) are, respectively, a dimensional [m] and non-dimensional [-] coordinate systems of the hull and (w-t) [-] is a non-dimensional coordinate system rotated around the x axis.

*Tab. 3.1. Coordinate systems and basic terminology of intact stability of ships*

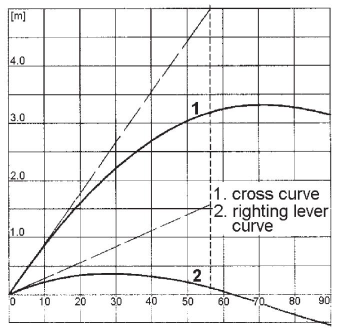



$$\begin{bmatrix} y \\ z \end{bmatrix} = \begin{bmatrix} \frac{1}{T} & \\ & \frac{1}{T} \end{bmatrix} \cdot \begin{bmatrix} Y \\ Z \end{bmatrix} \tag{3.1a}$$

$$\begin{bmatrix} w \\ t \end{bmatrix} = \begin{bmatrix} \cos\phi & \sin\phi \\ -\sin\phi & \cos\phi \end{bmatrix} \cdot \begin{bmatrix} y \\ z \end{bmatrix} \tag{3.1b}$$

$$L_R(\phi;Z_G) = l_R(\phi;z_G) \cdot T \ [m] \tag{3.2a}$$

$$l_R(\phi;z_G) = w_B(\phi) - z_G \cdot \sin\phi \ [-] \tag{3.2b}$$




A common feature of the criteria under discussion (in fact, all the contemporary criteria of intact stability of ships) is that, as far as the righting moment of a ship is concerned, they all are totally based on the righting lever curve calculated in a calm water as the function : $L_R(\phi;D,Z_G)$ (3.2a), where : $\phi$ is angle of inclination and **D** [t], $\mathbf{Z_G}$ [m] are a constant displacement and coordinate of the centre of gravity **G**, describing current loading condition of a ship. In further discussion, the only loading condition accounted for is the design condition. It follows from (3.2) that all the stability-related characteristics and parameters of a ship have been normalised with regard to the design draught T. A short overview of the criteria by IMO, PRS and HSMB in an analytical form has been shown in Table 3.2.

It can be seen that the criteria by IMO and PRS involve both righting $L_R(\cdot)$ and heeling $L_H(\cdot)$ lever curves („weather-type" criteria), whereas the criteria by HSMB are based on the righting lever curve exclusively („Rahola's-type" criteria).

It follows from Tab.3.2, that, from a mathematical viewpoint, all the criteria can be formulated as combinations of certain operations on the functions $L_R(\cdot)$ and $L_H(\cdot)$ : linear - such as calculating ordinates, first derivatives, and integrals or non-linear – – such as calculating characteristic angles, the weather parameter K, etc. Two examples illustrate the problem :

(i) Linear case - a dynamic righting lever curve $L_{R_2}(\cdot)$ (PRS) is defined by integration of $L_R(\cdot)$ :

$$L_{R_2}(\phi) = \begin{cases} \int\limits_{-\phi_0}^{-\phi} L_{R_1}(\phi)d\phi \quad + \quad \int\limits_{0}^{-\phi_0} L_{R_1}(\phi)d\phi & \text{for} \quad \phi \le 0 \\ \int\limits_{0}^{\phi} L_{R_1}(\phi)d\phi & \text{for} \quad \phi > 0 \end{cases} \tag{3.3}$$

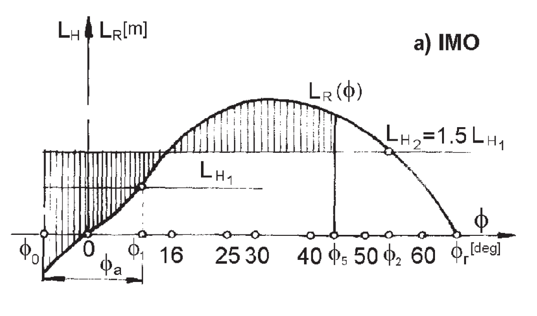
(ii) Non-linear case - angles of the first and second interception $\phi_1$, $\phi_2$ (IMO, PRS) are defined as the smaller and greater roots of the following non-linear equation :

$$F_1(\phi) \equiv L_{H_1}(\phi) - L_{R_1}(\phi) = 0 \ \Leftrightarrow \ \phi_1 = \min[F_1^{-1}(0)], \ \phi_2 = \max[F_1^{-1}(0)] \tag{3.4}$$

A formulation of the intact stability criteria in Tab. 3.2 differ from but are fully equivalent to those originally formulated in the referenced documents by IMO, PRS and HSMB. Among other things, a notation has been unified and sequences of the righting and heeling lever curves as well as characteristic angles have been introduced to emphasize the analytical aspects of the criteria.

In the subsequent chapters of the paper, the intact stability criteria will be examined with a special attention to those parts of them which concern the righting lever curve of a ship. This is because determination of the righting lever curve $l_R(\cdot)$, via cross curves $w_B(\cdot)$ involves the full geometry of the hull and as such decides on the reliability of the whole model.

***Tab. 3.2.*** *Intact stability criteria by IMO [6], PRS [17] and HSMB [5]*

<table>
<tr><th>Righting / heeling lever curves</th><th>Stability criteria</th></tr>
<tr><td>
</td><td>

1. $L'_R(0) \ge 0.15$ m
2. $L_R(25) \ge 0.20$ m
3. $L_R(30) \ge 0.20$ m
4. $\int_0^{30} L_R(\phi)\,d\phi \ge 0.055$ mrad
5. $\int_0^{\phi_3} L_R(\phi)\,d\phi \ge 0.090$ mrad
6. $\int_{30}^{\phi_3} L_R(\phi)\,d\phi \ge 0.30$ m
7. $\phi_1 \le \phi_4$
8. $\phi_K \ge \phi_5$
9. $\phi_r \ge 60^o$
10. $K_{IMO} \ge 1.0$

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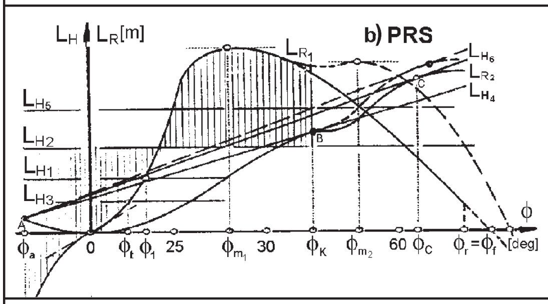
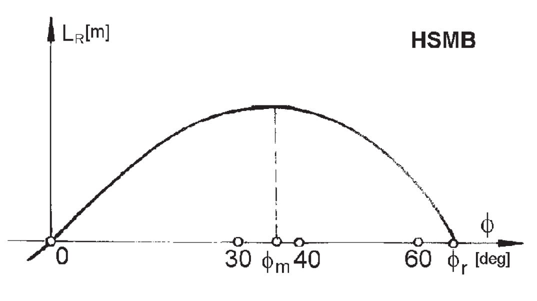
<tr><td>
</td><td>

1. $L'_R(0) \ge 0.20$ m
2. $L_R(\phi_m) \ge 0.20$ m
3. $\phi_6 \le \phi_7$
4. $\phi_{m_1} \ge 25 - 0.5 \cdot \Delta\phi_r$
5. $\phi_{m_2} \ge 0.30$
6. $\phi_K \le \phi_C$
7. $\phi_r \ge 60 - \Delta\phi_r$
8. $K_{PRS} \ge 1.0$

</td></tr>
<tr><td>
</td><td>

1. $L_R(30) \ge \frac{0.04}{C}$
2. $L_R(\phi_m) \ge \frac{0.05}{C}$
3. $\int_0^{30} L_R(\phi)\,d\phi \ge \frac{0.12}{C}$
4. $\int_0^{40} L_R(\phi)\,d\phi \ge \frac{0.020}{C}$
5. $\int_{30}^{40} L_R(\phi)\,d\phi \ge \frac{0.007}{C}$
6. $\int_0^{\phi_r} L_R(\phi)\,d\phi \ge \frac{0.035}{C}$

</td></tr>
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**Legend**

**Characteristic angles**

$\phi_0$ - initial heeling angle, $\phi_a$ - rolling amplitude, $\phi_f$ - ship flooding angle, $\phi_d$ - deck immersion angle

$\phi_t$ - turning angle, $\phi_s$ - superstructure flooding angle, $\phi_c$ - capsizing angle, $\phi_C = \min(\phi_f, \phi_c)$ - critical angle

$\phi_K$ - the angle for calculating K, $\phi_{m_1}, \phi_{m_1}$ -1st and 2nd maximum of $L_R$ angles, $\phi_m$ maximum of $L_R$ angle $\phi_m = \phi_{m_1}$ if $L_R(\phi_{m_1}) \ge L_R(\phi_{m_2})$ otherwise $\phi_m = \phi_{m_2}$, $\phi_1$, $\phi_2$ - first and second interception angles, $\phi_3 = \min(\phi_f, 40)$

$\phi_4 = \min(0.8 \cdot \phi_d, 16)$, $\phi_5 = \min(\phi_f, \phi_2, 50)$, $\phi_6 = \max(\phi_1, \phi_t)$, $\phi_7 = \min(0.5 \cdot \phi_d, 15)$

**Weather criteria indices**: $K_{IMO} = A_R / A_H\,[-]$, $K_{PRS} = L_{H_5} / L_{H_2}\,[-]$

**HSMB form factor**

$C = \frac{T \cdot H'}{B^2} \cdot \sqrt{\frac{T}{z_G}} \cdot \left(\frac{C_B}{C_W}\right)^2 \cdot \sqrt{\frac{100}{L}}$, where $H' = H + h \cdot \frac{2b - B}{B} \cdot \frac{2l_h}{L}$ - a corrected depth, h - height of a hatch above deck [m], b - breadth of a hatch $(b \ge B/2)$ [m], $l_h$ - length of hatches within 0.5 L

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</table>

## 4. ANALYTICAL FORM OF STABILITY-RELATED CHARACTERISTICS OF A SHIP

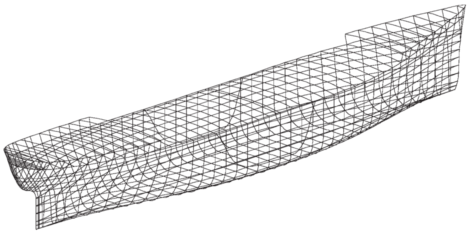
Analytical definition of the stability constraints needs the analytical definition of righting and heeling lever curves. For the reasons given earlier, the attention will focus on the righting lever curve only. This will be done by approximation or interpolation of a series of stability-related characteristics of a ship.

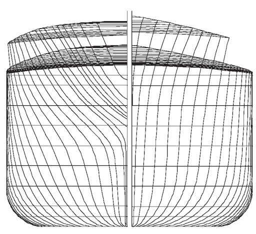
The geometrical data for the task were prepared in [1], based on the systematic calculations of cross curves for Series 60 (S60) [19] for three different block coefficients $C_B$ and four h = H/T ratios. Three parent models of S60 had been chosen and then modified by: (i) extrapolating sections to obtain different h ratios (beyond the basic $h^0 = 1.50$), (ii) adding standard superstructures (a poop and forecastle), (iii) extracting the data concerning merely the design draught T (displacement), and (iv) making them dimensionless. The ratio $b = B/T = b^0 = 2.50$ was kept constant for all the models. Table 4.1 presents the resulting twelve models and, as an example, geometry of the M7072 model.

Table 4.2 shows a full list of ship characteristics that have to be defined analytically in order to form the stability constraints in the fleet optimisation model. All of them are dimensionless (related to T) and classified from the analytical point of view as one - (1D), two - (2D), three - (3D), and four-dimensional (4D) characteristics, taking into account a number of variables of corresponding interpolating or approximating functions.

***Tab. 4.1.*** *The Series 60 hull models used for the approximation of cross curves and geometry of the M7072 model*

| **Series 60** | **Series 60 Modified Models** (B/T = 2.50) | | | | |
|---|---|---|---|---|---|
| **Parent Models** | **$C_B$ \ H/T** | **1.610** | **1.726** | **1.813** | **1.900** |
| **4210W** | **0.60** | M6061 | M6072 | M6081 | M6090 |
| **4212W** | **0.70** | M7061 | M7072 | M7081 | M7090 |
| **4214WB-4** | **0.80** | M8061 | M8072 | M8081 | M8090 |





***Tab. 4.2.*** *Stability-related geometrical characteristics of a hull form and their analytical representation*

| No | Characteristics | Dimension | Generation process | Function |
|---|---|---|---|---|
| 1 | Block coefficient | 2D | Interpolation | $\delta = f_1(C_B, z)$ |
| 2 | Vertical coordinate of the centroid B | 2D | Interpolation | $z_B = f_2(C_B, z)$ |
| 3 | Transversal metacentric radius | 3D | Interpolation | $r_0 = f_3(C_B, b, z)$ |
| 4 | Deck immersion angle | 3D | Interpolation | $\phi_D = f_4(C_B, h, b)$ |
| 5 | Cross curves in the (w-t) system | 4D | Approximation | $w_B = f_5(\phi; C_B, h, b)$ |
| 6 | Righting lever curve | 4D | Approximation | $l_R = f_6(\phi, C_B, h, b; z_G)$ |
| 7 | Righting lever curve | 1D | Approximation | $\widetilde{l}_R = f_7(\phi)$ |

Fig.4.1 presents three examples of 2D/3D geometrical characteristics of a hull form based on the S60 data defined by the interpolation method.

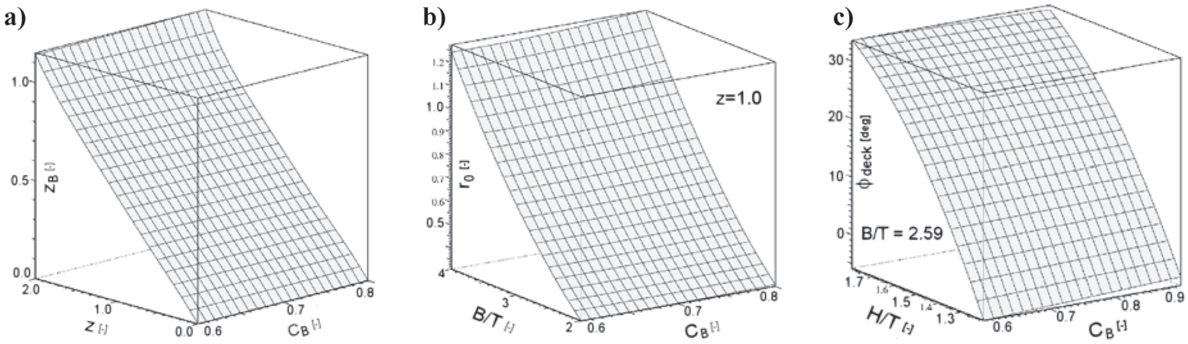



***Fig. 4.1.*** *Graphs of analytically defined 2D / 3D geometrical characteristics of Series 60 by the interpolation method*

In the analytical definition of stability constraints, the basic characteristic is righting lever curve $l_R(\phi;\cdot)$. A fleet optimisation model requires for the righting lever curve to be defined as the following 4D function :

$$l_R(\phi, C_B, h, b; z_G) = w_B(\phi, C_B, h, b) - z_G \cdot \sin\phi \tag{4.1}$$

The process of generation of the righting lever function $l_R(\phi;\cdot)$ (4.1) has been divided into five steps :

(i) Approximation of cross curves based on the twelve models of S60, resulting in a 3D function $w_B^0(\phi, C_B, h)$ for $b = b^0 = \text{const}$.

(ii) Correction of the $w_B^0(\cdot)$ function for the regions of small h (below the available S60 data), resulting in the $w_B^{cor}(\cdot)$ function for these special regions. This step was necessary to cover H/T ratios typical of sea-river vessels and expected when optimising a fleet of such ships.

(iii) Affine transformation of the function $w_B^0(\cdot) / w_B^{cor}(\cdot)$ with regard to $b \neq b^0$, resulting in a 4D function $w_B^1(\phi, C_B, h, b)$

(iv) Definition of the righting lever curve as a 4D function: $l_R(\phi, C_B, h, b) = w_B^1(\phi, C_B, h, b) - z_G \cdot \sin\phi$

(v) Approximation of the 4D function $l_R(\phi, C_B, h, b; z_G)$ by a 1D function (of $\phi$ only): $\widetilde{l}_R(\phi; C_B, h, b, z_G)$ for $C_B, h, b, z_G = \text{const}$.

In all the steps listed above, the only problem that appears twice is that of approximation of multivariable (i) or single-variable (v) functions. The problem will be then addressed first as a separate numerical problem formulated in a compact matrix notation.

### *4.1. **Approximation problem***

Let $f(\mathbf{u})$ be a function to be approximated and $\tilde{f}(\mathbf{u})$ be its approximation, both assumed to be single valued, multi-variable functions of $\mathbf{u} = (u_1, u_2, ....u_M)$. The approximation problem is formulated as :

$$\rho(f, \tilde{f}) \to \min \tag{4.2}$$

where : $\rho(\cdot,\cdot)$ is a metric of approximation. Let the $f(\cdot)$ function be given in a discrete manner by the following sequence of data : $\{\mathbf{u}_k ; f_k\}_{k=1}^{N}$ and the $\tilde{f}(\cdot)$ function has the following linear representation :

$$\tilde{f}(\mathbf{u};\alpha) = \sum_{i=1}^{n} \alpha_i \cdot e_i(\mathbf{u}) = \alpha^T \cdot \mathbf{e}(\mathbf{u}) \quad , \quad n < N \tag{4.3}$$

where : $\alpha = (\alpha_1, \alpha_2, ....\alpha_n)^T$ a vector of unknown coefficients, $\mathbf{e}(\mathbf{u}) = [e_1(\mathbf{u}), e_2(\mathbf{u}), ....e_n(\mathbf{u})]^T$ - a basis.

For given arguments one has: $\tilde{f}_k = \tilde{f}(\mathbf{u}_k, \alpha),\ k = 1,2,..., N$ and $\tilde{\mathbf{f}} = (\tilde{f}_1, \tilde{f}_2, ...., \tilde{f}_N)^T$

Using the representation (4.3), one gets the following equation :

$$\tilde{\mathbf{f}} = \mathbf{E}^T \cdot \alpha \tag{4.4}$$

A matrix $\mathbf{E} = [E_{ij}] = [e_i(\mathbf{u}_j)]_{nxN}$ will be later called the *basis matrix*. As the metric of approximation $\rho(\cdot\,,\cdot)$ a weighted norm $\|\cdot\|_{2w}$ of an Euclidean distance between f and $\tilde{f}$ has been chosen, then :

$$\rho(f, \tilde{f}) = \left\| f - \tilde{f} \right\|_{2w} \tag{4.5}$$

In consequence, the approximation problem (4.2) can be formulated as the following quadratic optimisation problem :

$$\min Q(\alpha) \ , \ Q : \mathbf{R}^n \to \mathbf{R}^1 \tag{4.6a}$$

with the objective function :

$$Q(\alpha) = \left\| f - \tilde{f} \right\|_{2w}^2 = (\mathbf{f} - \mathbf{E}^T \cdot \alpha)^T \cdot \mathbf{W} \cdot (\mathbf{f} - \mathbf{E}^T \cdot \alpha) \tag{4.6b}$$

where : $\mathbf{W} = \mathrm{diag}(\mathbf{w})$ , $\mathbf{w} = (w_1, w_2, ..., w_n)^T$ – a vector of weighting factors ($w_i \geq 0,\ i = 1,2,...., N$).

Now let us assume that some linear interpolatory constraints are imposed on the $\tilde{f}(\cdot)$ function :

$$I_\mu(\tilde{f}) = c_\mu \ , \ \mu = 1,2,...., m < n = \dim \alpha \tag{4.7}$$

where $I_\mu(\cdot)$ is a functional of $\tilde{f}(\cdot)$ in the general form :

$$I_\mu(\tilde{f}) = L_\mu(\tilde{f})(\mathbf{u}_\mu; \boldsymbol{\alpha}) \quad , \quad \mu = 1, 2, ...., m \tag{4.8a}$$

and $L_\mu(\cdot)$ is a linear operator. For instance, $L_\mu(\cdot)$ can be a differential operator, as below :

$$L_\mu(\tilde{f}) = \frac{\partial^{\lambda_\mu} \tilde{f}(\cdot)}{\partial u_{i_\mu}^{\lambda_\mu}} \quad , \quad \lambda_\mu = 0, 1, 2... , \quad i_\mu = 1, 2, 3, .... \tag{4.8b}$$

where the sequences of indices: $\{\lambda_\mu\}$, $\{i_\mu\}$, $\mu = 1,2,...., m$ have to be defined separately.

Let $\mathbf{I}(\tilde{f}) = (I_1(\tilde{f}), I_2(\tilde{f}), ...., I_m(\tilde{f}))$ be a vector of functionals of $\tilde{f}(\cdot)$.

The interpolatory constraints can then be written as the following matrix equation :

$$\mathbf{I}(\tilde{f})(\alpha) = \mathbf{c}^T \tag{4.9}$$

where : $\mathbf{c} = (c_1, c_2, ....c_m)^T$ is a vector of given values of interpolatory constraints. Thanks to the linearity of $I(\cdot)$, one can write :

$$\mathbf{I}(\tilde{\mathbf{f}}) = \mathbf{I}(\alpha^T \cdot \mathbf{e}(\mathbf{u})) = \boldsymbol{\alpha}^T \cdot \mathbf{I}(\mathbf{e}(\mathbf{u})) = \boldsymbol{\alpha}^T \cdot \mathbf{C}^T \tag{4.10}$$

A matrix $\mathbf{C} = [c_{\mu\nu}] = [L_\mu(e_\nu)(\mathbf{u}_\mu)]$ will be later called the *constraint matrix*.

Putting (4.10) into (4.9), one obtains the interpolatory constraints as the following matrix equation :

$$\mathbf{H}(\alpha) \equiv \mathbf{C} \cdot \boldsymbol{\alpha} - \mathbf{c} = \mathbf{0} \tag{4.11}$$

The approximation problem with the additional interpolatory constraints can now be formulated as the following quadratic optimisation problem with constraints :

$$\begin{cases} \min Q(\alpha) \equiv (\mathbf{f} - \mathbf{E}^T \cdot \boldsymbol{\alpha})^T \cdot \mathbf{W} \cdot (\mathbf{f} - \mathbf{E}^T \cdot \alpha) \\ \alpha \in \Omega = \{\alpha : \mathbf{H}(\boldsymbol{\alpha}) = \mathbf{0}, \quad \dim \mathbf{0} = m < n\} \subset \mathbf{R}^n \end{cases} \tag{4.12}$$

It has been proven [3], that both problems (4.6) and (4.12), as the convex problems, do have unique solutions. Moreover, in both cases the solutions can be found analytically. In particular, for the problem (4.12), this can be done either by Lagrange's multipliers or by the penalty function method. The latter approach has been applied in the case. Let the penalty function $P(\cdot)$ be defined as a square of the norm $\|\mathbf{H}(\alpha)\|_{2\omega}$ in the form :

$$P(\boldsymbol{\alpha}) = \mathbf{H}^T(\boldsymbol{\alpha}) \cdot \omega \cdot \mathbf{H}(\alpha) \tag{4.13}$$

where: $\boldsymbol{\omega}$ = diag(ω), ω = ($\omega_1, \omega_2, ..., \omega_n$) - a vector of weighting factors for the penalty function, $\omega_i \geq 0.,\ i = 1,2,....,m$.
The optimisation problem with constraints (4.12) can now be replaced by the equivalent optimisation problem without constraints, with the modified objective function F(·) :

$$\begin{cases} \min F(\boldsymbol{\alpha}) \equiv Q(\boldsymbol{\alpha}) + P(\boldsymbol{\alpha}) \ , \ \ P : \mathbf{R}^n \to \mathbf{R}^1 \\ \boldsymbol{\alpha} \in \mathbf{R}^n \end{cases} \tag{4.14}$$

The solution to the problem (4.14) follows from the necessary condition of a stationary point in $\mathbf{R}^n$ :

$$\nabla_\alpha F(\boldsymbol{\alpha}) = \mathbf{0} \quad \Leftrightarrow \quad \nabla_\alpha Q(\alpha) + \nabla_\alpha P(\alpha) = \mathbf{0} \tag{4.15}$$

Finally, one arrives at the equation :

$$\underbrace{(\mathbf{E} \cdot \mathbf{W} \cdot \mathbf{E}^T + \mathbf{C}^T \cdot \boldsymbol{\omega} \cdot \mathbf{C})}_{\mathbf{B}} \cdot \boldsymbol{\alpha} - \underbrace{(\mathbf{E} \cdot \mathbf{W} \cdot \mathbf{f} + \mathbf{C}^T \cdot \boldsymbol{\omega} \cdot \mathbf{g})}_{\mathbf{b}} = \mathbf{0} \tag{4.16}$$

or, in short :

$$\mathbf{B} \cdot \boldsymbol{\alpha} - \mathbf{b} = \mathbf{0} \tag{4.17}$$

Existence of the solution to the problem (4.12) assures that the matrix **B** is non-singular (det **B** ≠ 0).
So one eventually gets the desired definition of the approximating function $\tilde{f}(\cdot)$ :

$$\boldsymbol{\alpha} = \mathbf{B}^{-1} \cdot \mathbf{b} \tag{4.18}$$

One can observe that: (i) the solution α is parametrized by the weighting coefficients: α = α(w;ω) and (ii) putting ω = 0, one gets the solution of an unconstrained problem (4.6a).

From the numerical point of view, solution α to the approximation problem (4.2) can be found either by solving the system of linear equations (4.17) or by solving a single matrix equation (4.18) (inverting B). The first approach has been applied in approximation of cross curves (Section 4.2) and the second one in approximation of righting lever curve (Section 4.4).

## *4.2. **Approximation of cross curves***

In this case the approximated function is a cross curves function ($f = w_B^0$) determined in a discrete way for the twelve S60 models listed in Tab. 4.1. Given 11 ordinates of the function per one model, one gets N = 132 ordinates to be approximated. The following polynomial three-linear form has been chosen, as a particular representation of the approximating function $\tilde{f}(\cdot)$.

$$\tilde{f}(\mathbf{u};\boldsymbol{\alpha}) = \boldsymbol{\alpha}^T \cdot \mathbf{e}(\mathbf{u}) = \sum_{l=1}^{n} \alpha_l \cdot e_l(\mathbf{u}) = \sum_{\mu=1}^{n} \alpha_\mu \cdot e_\mu(\phi, C_B, h) = \sum_{\mu=1}^{n} \alpha_\mu \cdot \underbrace{(\phi^{i_\mu} \cdot h^{j_\mu} \cdot C_B{}^{k_\mu})}_{e_\mu(\mathbf{u})} \equiv \qquad \text{(a)}$$

$$\equiv \sum_{i=0}^{I} \sum_{j=0}^{J} \sum_{k=0}^{K} a_{i,j,k} \cdot \phi^i \cdot h^j \cdot C_B{}^k \qquad \text{(b)} \tag{4.19}$$

The equivalency of the two notations introduced follows from the correspondence of both indications. After some trials and errors, the upper limits of the indices in (4.19b) have been established as follows : $I = 5, J = 2, K = 2$. It gives the following dimension of the basis $\mathbf{e}(\cdot)$ in (4.3) : $\dim \mathbf{e}(\cdot) = n = (I+1) \cdot (J+1) \cdot (K+1) = 54$, so the condition n < N holds. The basis matrix **E** (4.4) can now be readily calculated according to the definition :

$$\mathbf{E} = [e_\mu(\mathbf{u}_\nu)] = \left[(\phi^{i_\mu})_\nu \cdot (h^{j_\mu})_\nu \cdot (C_B^{k_\mu})_\nu\right] \quad \text{for} \quad \begin{cases} \mu = 1,2,....,n \\ \nu = 1,2,....,N \end{cases} \tag{4.20}$$

The constraint imposed on the approximated function $w_B^0(\phi,\cdot)$ follows from the known property of cross curves :

$$\frac{d}{d\phi} w_B(\phi)\Big|_{\phi=0} = z_B + r_0 \equiv z_M \tag{4.21}$$

It imposes the following interpolatory constraints on the approximating function $\tilde{f}(\cdot)$ :

$$\frac{\partial}{\partial\phi} \tilde{f}(\phi\,; C_B, h)\Big|_{\phi=0} = z_B(C_B) + r_0(C_B) \tag{4.22}$$

Eq. (4.22) should hold for all the models given, so the number of imposed constraints is m = 12.
The determination of the weighting coefficient matrices **W** and ω in (4.12 and 4.13) was done by the „trial and error" method.

The best results obtained are as follows :

$$w_i = \begin{cases} 1000.0 & \text{for} \quad \text{i corresponding to } \phi = 0 \\ 1.0 & - \quad \text{otherwise} \end{cases} \qquad \omega_i = 1 \ \text{ for } \ i = 1,2,...,m \tag{4.23}$$

Results of the approximation of cross curves $w_B^0(\cdot)$ for the S60 models (Tab.4.1) is shown in Fig.4.2.

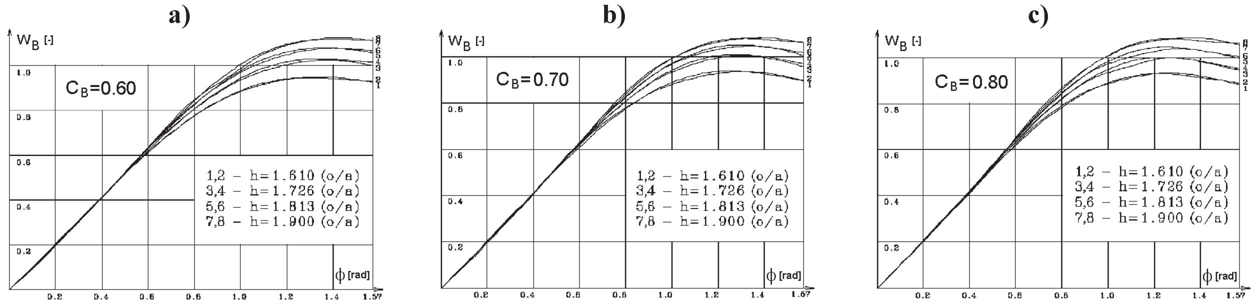
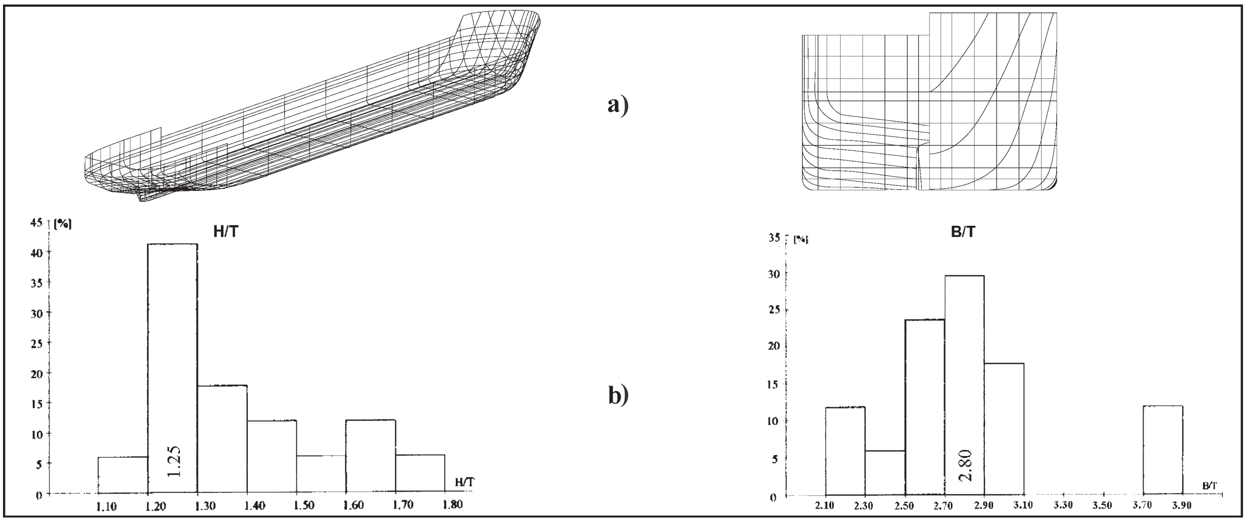



***Fig. 4.2.*** *Comparison of the original (o) and approximated (a) cross curves of S60*

## *4.3.* ***Correction of the cross curves function $\mathbf{w_B^0}$ for small H/T ratios***

The function $w_B^0(\phi, C_B, h)$ approximating the cross curves of the S60 data is formally valid for the form parameters from the region : $(C_B, h) \in [0.60, 0.80] \times [1.61, 1.90]$ , b = $b^0$ = 2.5. Such a range of data evidently corresponds to the standard hull form parameters of sea-going ships but does not cover the regions typical of the sea-river vessels where the block coefficient $C_B$ reaches values as large as 0.90 and h as small as 1.1 (Tab.4.3).

***Tab. 4.3.*** *Hull form of a sea-river vessel [18]. a - body lines, b - a sample of statistics [13]*




It turned out that a natural extrapolation of the cross curves function (4.19) obtained for S60 towards large $C_B$ and small h=H/T is acceptable as far as $C_B$ and not acceptable as far as h is concerned. Fig.4.3 illustrates the problem. So a special extrapolation was necessary for the function $w_B^0$ to be used in an optimisation model of sea-river ships. As a result, a corrected cross curves 2D function $w_B^{cor}(\phi, h)$ has been defined (by an interpolation technique), based on the boundary data of S60 (for h = 1.61) and a cross curve data for the SINE-205 sea-river vessel [18] (for h = 1.239) in the following tensor product form :

$$w_B^{cor}(\phi, h) = \mathbf{d}^T(\phi) \cdot \mathbf{g}(h) = \mathbf{e}^T(\phi) \cdot \mathbf{D} \cdot \mathbf{g}(h) \tag{4.24}$$

where : $\mathbf{d}(\phi)$ – interpolatory characteristics, $\mathbf{D}$ - interpolatory matrix, $\mathbf{e}(\phi)$, $\mathbf{g}(\phi)$ – bases, dim($\mathbf{d}$) = dim($\mathbf{g}$) = 3, dim($\mathbf{e}$) = 5. For details - see [13].

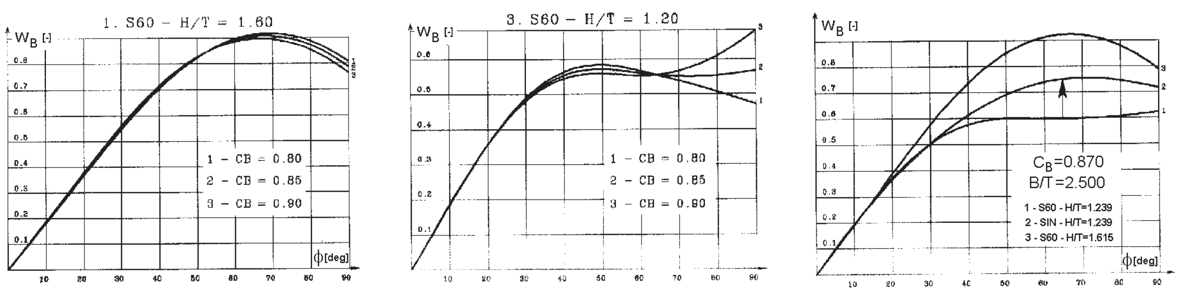


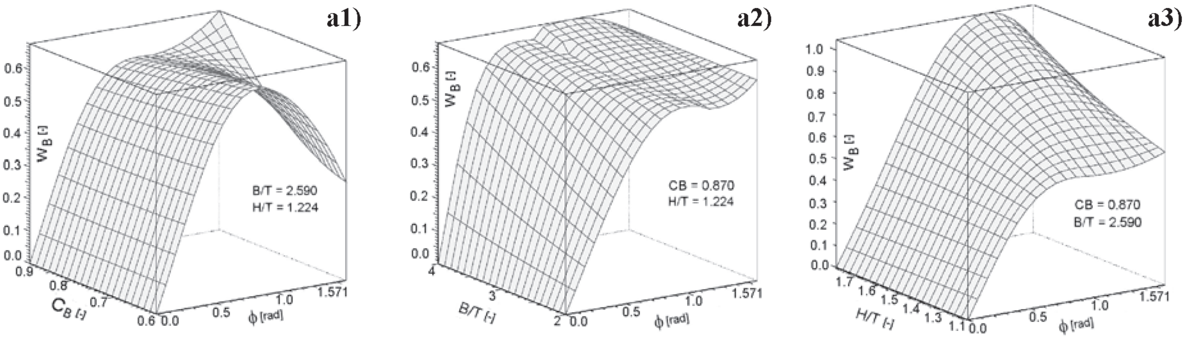

***Fig. 4.3.*** *A natural extrapolation of the cross curves function $w_B^0(\cdot)$ of S60 towards the region of small H/T ratio*

After correction, a definition of the basic (B/T = $b^0$ = 2.5) cross curves function is :

$$w_B^0(\phi, C_B, h) = \begin{cases} w_B^0(\phi, C_B, h) & \text{for} \quad h \geq 1.610 \\ w_B^{cor}(\phi, h) & \text{for} \quad h < 1.610 \end{cases} \tag{4.25}$$

Fig.4.4 show comparison of the basic cross curves functions before and after correction with regard to the small H/T ratio.

**A. Before correction**




**B. After correction**

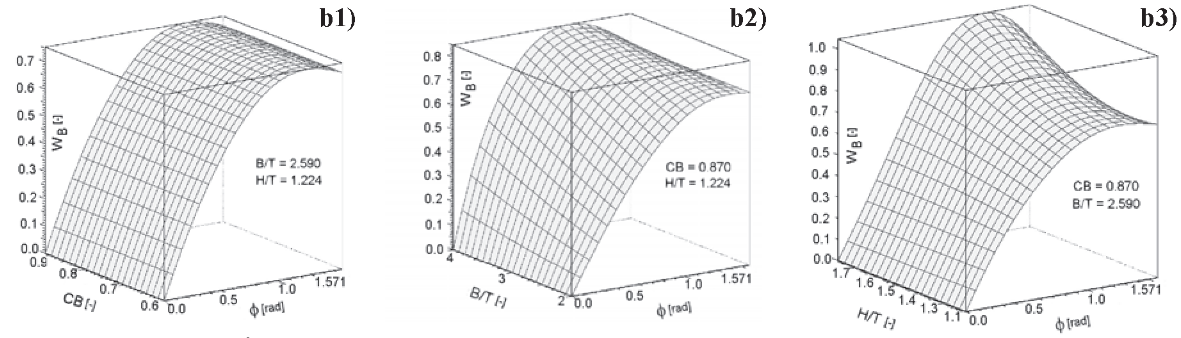
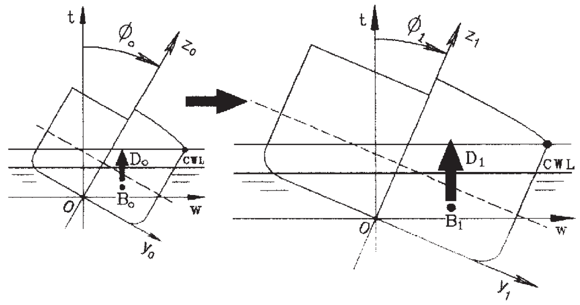



***Fig. 4.4.*** *Basic (b = b⁰=2.5) cross curves function before and after correction with regard to the small H/T ratio*

## 4.4. *Affine transformation of the cross curves function* $\mathbf{w_B^0(\cdot)}$ *with regard to B/T*

The definition of cross curves function $w_B^0(\cdot)$, obtained as the result of approximation of the Series 60 data and then corrected for small H/T ratio is only valid for B/T = $b^0$ = 2.5. Accounting for a new B/T ratio = $b^1 \neq b^0$ can be done by the way of affine transformation of a hull form. The situation is illustrated in Fig.4.5. A system of hull coordinates is transformed when the hull is heeled. As a result, for *equivalent* waterlines (e g. CWL or deck immersion waterline), one obtains, among other things, a changed angle of heel ($\phi_0 \rightarrow \phi_1$), changed coordinates of B ($B_0 \rightarrow B_1$) and, in consequence, the desired transformation of cross curves $w_B^0(\phi, C_B, h\,;\, b = b^0) \rightarrow w_B^1(\phi, C_B, h, b)$.




$$\begin{bmatrix} y_1 \\ z_1 \end{bmatrix} = \begin{bmatrix} \lambda_b & 0 \\ 0 & 1 \end{bmatrix} \cdot \begin{bmatrix} y_0 \\ z_0 \end{bmatrix} \qquad \lambda_b = b^1 / b^0$$

***Fig. 4.5.*** *Affine transformation of coordinate system*

In order to derive a particular form of this transformation,
let us recall the inverse relation between the (y-z) and (w-t) systems (3.1b) :

$$\begin{bmatrix} y \\ z \end{bmatrix} = \begin{bmatrix} \cos\phi & -\sin\phi \\ \sin\phi & \cos\phi \end{bmatrix} \cdot \begin{bmatrix} w \\ t \end{bmatrix} \tag{4.26}$$

Using the formula (4.26) for the points $B_0$ and $B_1$ one has accordingly
the following (w-t) $\rightarrow$ (y-z) transformation of coordinates :

$$\begin{bmatrix} y_B^0 \\ z_B^0 \end{bmatrix} = \begin{bmatrix} \cos\phi_0 & -\sin\phi_0 \\ \sin\phi_0 & \cos\phi_0 \end{bmatrix} \cdot \begin{bmatrix} w_B^0 \\ t_B^0 \end{bmatrix} \tag{4.27a}$$

$$w_B^1 = y_B^1 \cdot \cos\phi_1 + z_B^1 \cdot \sin\phi_1 = (\lambda_b \cdot y_B^0) \cdot \cos\phi_1 + z_B^0 \cdot \sin\phi_1 \tag{4.27b}$$

After substitution of (4.27a) into (4.27b) one arrives at the formula :

$$w_B^1 = \cos\phi_0 \cdot \cos\phi_1 \cdot \left[ w_B^0 \cdot (\lambda_b + tg\phi_0 \cdot tg\phi_1) + t_B^0 \cdot (-\lambda_b \cdot tg\phi_0 + tg\phi_1) \right] \tag{4.28}$$

Further algebraic manipulations lead to the final transformation function (the indices 1 were dropped) :

$$w_B^1(\phi, \lambda_b) = \begin{bmatrix} f_w(\phi, \lambda_b) \\ \hline f_t(\phi, \lambda_b) \end{bmatrix}^T \cdot \begin{bmatrix} w_B^0[\phi_0(\phi)] \\ t_B^0[\phi_0(\phi)] \end{bmatrix} \tag{4.29}$$

where the auxiliary functions involved in (4.29) are :

$$\begin{cases} f_w(\phi, \lambda_b) & = & \lambda_b \cdot \tau(\phi, \lambda_b) & \text{(a)} \\ f_t(\phi, \lambda_b) & = & 0.5 \cdot (1 - \lambda_b^2) \cdot \sin 2\phi \cdot \tau(\phi, \lambda_b) & \text{(b)} \\ \tau(\phi, \lambda_b) & = & \dfrac{1}{\sqrt{\cos^2\phi + \lambda_b^2 \cdot \sin^2\phi}} & \text{(c)} \\ t_B^0(\phi, \lambda_b) & = & z_B^0 - \displaystyle\int_0^{\phi_0(\phi)} w_B^0(\varphi) d\varphi & \text{(d)} \\ \phi_0(\phi, \lambda_b) & = & \mathrm{arc\,tg}(\lambda_b \cdot tg\phi) & \text{(e)} \end{cases} \tag{4.30}$$

Fig. 4.6 presents graphs of a heeling angle function (4.30e) for different $\lambda_b \in [0.25\ ,\ 2.00]$ and the functions $w_B^1(\cdot)$ (4.29) and $t_B^0(\cdot)$ (4.30 d).

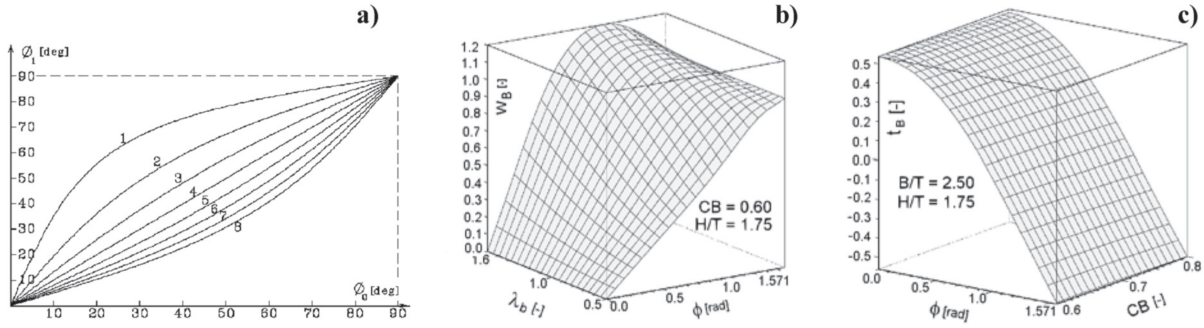



***Fig. 4.6.*** *Affine transformation of cross curves with regard to the B/T ratio* ($\lambda_b = b/b^0$)

## 4.5. *Analytical definition of the righting lever curve $l_R(\cdot)$*

An analytical definition of the righting lever function for the hull form parameters **g** = ($C_B$ , h , b) :

$$l_R(\phi, \mathbf{g}; z_G) = w_B(\phi, \mathbf{g}) - z_G \cdot \sin\phi \tag{4.31}$$

can instantly be obtained based on the analytical definition of the cross curves function (4.29) :

$$w_B(\phi, \mathbf{g}) \equiv w_B^2(\phi, \mathbf{g}) = w_B^1(\phi, C_B, h) \cdot f_w(\phi, b) + t_B^1(\phi, C_B, h) \cdot f_t(\phi, b) \tag{4.32}$$

When looking at the formulae above it becomes obvious that the righting lever function so defined would be too complicated for performing various analytical operations (with regard to $\phi$) when generating the stability constraints. In order to cope with this problem, a 4D function $l_R(\phi, \mathbf{g}; z_G)$ has been approximated by an 1D function $\tilde{l}_R(\phi; \mathbf{g}, z_G)$ (for **g** , $z_G$ = const.) using the known solution to the approximation problem (4.18). The idea is based on the observation that the matrix **B** depends only on the given nodes (matrices **E**, **C**), the types of constraints (matrix **C**), the weighting factors (matrices **W**, $\boldsymbol{\omega}$) and does not depend on ordinates of the approximated function or values of constraints (**c**). It follows from the above that for all the mentioned elements kept fixed, the matrix **B** and its inversion $\mathbf{B}^{-1}$ can be generated once and be used many times for different combinations of the **g** and $z_G$ parameters, when running the optimisation procedure. Bearing in mind the need for simplicity, a linear polynomial representation has been assumed for the approximating function :

$$\tilde{f}(\mathbf{u}; \boldsymbol{\alpha}) = \boldsymbol{\alpha}^T \cdot \mathbf{e}(u) = \sum_{i=1}^{n} \alpha_i \cdot e_i(\mathbf{u}) \quad \Leftrightarrow \quad \tilde{l}_R(\phi; \boldsymbol{\alpha}) = \sum_{i=1}^{6} \alpha_i \cdot \phi^{i-1} \tag{4.33}$$

where : $\alpha = (\alpha_1, \alpha_2, ....\alpha_6)^T$ - is a solution of the approximation problem and the definition of $\tilde{l}_R(\cdot)$. Note, that $\alpha = \alpha(\mathbf{g}, z_G)$.

In analogy to approximation of cross curves, the only constraint (m = 1) imposed on the lever righting function is :

$$\frac{d}{d\phi}(\tilde{l}_R)(\phi = 0) = z_B(C_B) + r_0(C_B, b) - z_G \tag{4.34}$$

The resulting righting lever function has the desired simple form :

$$\tilde{l}_R = \tilde{l}_R(\phi; C_B, h, b; z_G) = \sum_{i=1}^{6} \alpha_i(C_B, h, b; z_G) \cdot \phi^{i-1} \tag{4.35}$$

Fig. 4.7 presents some diagrams of the righting lever curve as a 4D function $l_R(\phi , C_B , h , b ; z_G)$ according to the definition (4.31).

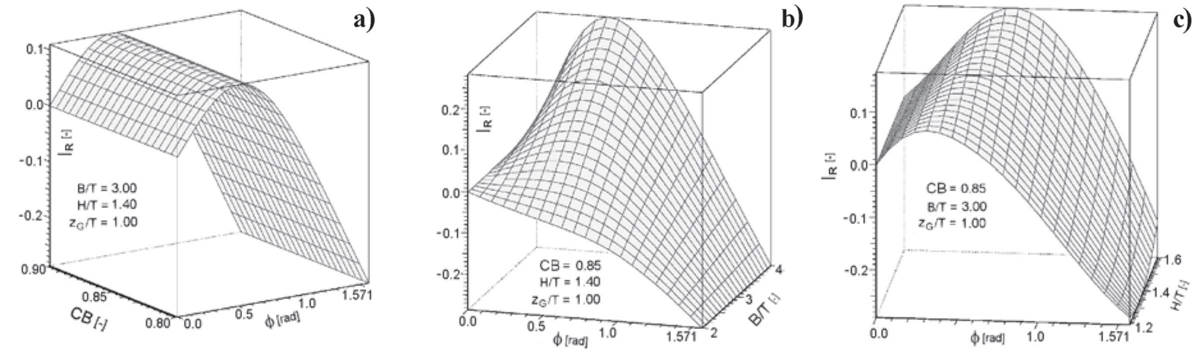



***Fig. 4.7.*** *Righting lever curve as a multivariable function of $\phi$ and hull form parameters of a ship*

## 5. **STABILITY CONSTRAINTS IN POST-OPTIMISATION STUDIES**

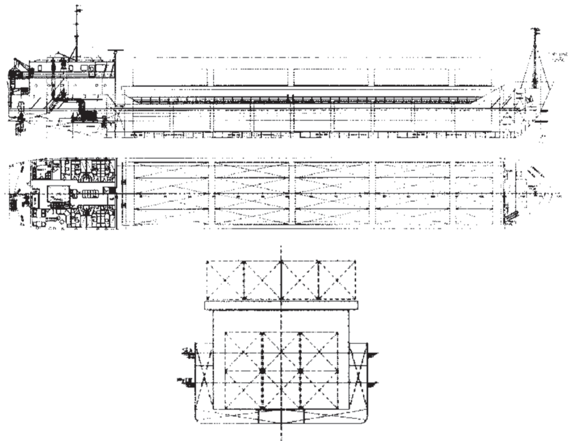
An example of post-optimisation stability-oriented study is presented on the background of the optimum solution of the fleet of ships optimisation problem. To perform the calculations, the optimisation model of a fleet has been implemented in the EUROS computer program [13].

### *5.1. **The data***

A typical sea-river vessel and a sample of the main fleet transportation data are shown in Table 5.1.

***Tab. 5.1.*** *Fleet transportation problem :*

Typical sea-river vessel for short shipping in the North and Baltic Seas [18] and main transportation data [13]

| **Sea-river vessel** | **A sample of main transportation data :** |
|---|---|
|  | Period of operation : N = 15 years<br>No. of ports : OUT / INP / total = 10 / 11 / 20<br>No. of river ports = 12<br>Mass of cargo OUT = 110 000.0 tons / year<br>Mass of cargo IN = 103 000.0 tons / year<br>Distance : OUT / IN / total: 2827 / 2959 / 5786 [NM]<br>Max. draught / velocity in river : 2.8 m/ 14 km/h<br>Period of calling at ports : 6 - 14 days<br>General cargo stowage factor : 1.0 - 2.0 [$m^3$/t]<br>Relative quotas of cargo type :<br>qcc / qgc /qbc OUT = 0.8 / 0.2 / 0.0 [-]<br>qcc /qgc / qbc IN = 0.9 / 0.1 / 0.0 [-]<br>qcc - containerised c., qgc - general c., qbc - bulk c. |

Sea-river ships, as multi-purpose vessels, carry different kind of commodities, such as containers, general cargo and bulk cargo. In the transportation data, these are accounted for by fixing relative quotas of cargo types for the whole period of fleet operation.

Referring to the terminology introduced in Table 2.2a, the overall quantities of numerical data used in the model (including those in Tab.5.1), in the four categories of constants, are: VESSEL - 154, CARGO 10, ENVIRONMENT - 374, OPERATION - 326. This sums up to 864 constants in total.

### *5.2. **The model and NPM algorithms***

The implemented model for fleet optimisation problem contains 12 decision variables (Tab.5.2), 70 constraints in total (including 26 stability constraints) and 3 alternative objective functions: Net Present Value (NPV), Internal Rate of Return (IRR) and Required Freight Rate (RFR). As a non-linear programming method of optimisation, a combined, double level, algorithm for constrained problems has been applied, including a penalty function shifting method by Powell and Wierzbicki, for the constrained part, and Rosebrock's method for the unconstrained part of the problem ([3],[15]).

### *5.3. **Numerical results***

Numerical examples of optimisation are presented in Table 5.2. The calculations have been carried out separately for three criteria of stability: IMO, PRS, and HSMB with the same data, the same objective function: (NPV) and the same starting point $\mathbf{x}^0$. The optimisation process has been repeated twice : the results of the first optimisation were used as a starting point to the second optimisation and then convergence was reached. It turned out that, from the viewpoint of the final value of the objective function, such an approach improved the results remarkably. Apart from the NPV, the results in Table 5.2 show the corresponding values of two remaining measures of merit: IRR and RFR. It can be seen from Table 5.2 that the results for the stability criteria by IMO and HSMB are identical. This will become obvious when one analyses the status of the stability constraints with regard to the feasible solutions region $\mathbf{\Omega}$.

### *5.4. **Feasibility analysis - the case study***

Feasibility analysis is an element of post-optimisation studies (together with parametric study and sensitivity analysis) that aims at investigating the status of a certain group of constraints (in the case - the stability constraints) versus boundary

***Tab. 5.2.*** *Results of optimisation of a fleet of multipurpose 'sea-river' ships for three intact stability criteria. Objective function: NPV*

| | | | | Intact Stability Criteria | | |
|---|---|---|---|---|---|---|
| **x** | **Description** | **Unit** | **Start** | **IMO** | **PRS** | **HSMB** |
| $x_1$ | Number of ships in a fleet | [-] | 4.000 | 3.927 | 4.433 | 3.927 |
| $x_2$ | Deadweight of a ship | [t] | 2941.000 | 3178.873 | 3112.931 | 3178.873 |
| $x_3$ | Speed of a ship at sea | [kn] | 12.000 | 11.075 | 11.059 | 11.075 |
| $x_4$ | Total container capacity of a ship | [TEU] | 90.000 | 117.499 | 110.336 | 117.499 |
| $x_5$ | Length b.p. of a ship ($L_{PP}$) | [m] | 87.470 | 99.106 | 97.592 | 99.106 |
| $x_6$ | Breadth of a ship (B) | [m] | 11.400 | 13.422 | 12.996 | 13.422 |
| $x_7$ | Draught of a ship (T) | [m] | 4.400 | 4.252 | 4.170 | 4.252 |
| $x_8$ | Depth of a ship (H) | [m] | 5.450 | 7.010 | 6.833 | 7.010 |
| $x_9$ | Block coefficient ($C_B$) | [-] | 0.874 | 0.796 | .827 | 0.796 |
| $x_{10}$ | No of container columns under deck | [-] | 9.000 | 11.296 | 11.257 | 11.296 |
| $x_{11}$ | No of container rows under deck | [-] | 3.000 | 4.345 | 3.000 | 4.345 |
| $x_{12}$ | No of container tiers under deck | [-] | 2.000 | 2.302 | 2.259 | 2.302 |
| NPVI | Net Present Value / Investment Cost | [%] | -10.000 | 28.800 | 18.400 | 28.800 |
| IRR | Internal Rate of Return | [%] | 8.23 | 14.540 | 12.934 | 14.540 |
| RFR | Required Freight Rate | [$/t] | 121.49 | 102.937 | 111.784 | 102.937 |

***Tab. 5.3.*** *List of stability constraints*

| | | | | | Constraints' index in the model | |
|---|---|---|---|---|---|---|
| **No** | **Criterion** | **Type** | **Unit** | **Institution** | **container ship** | **general cargo ship** |
| 1 | GM0 at river | min | [m] | - | 45 | - |
| 2 | GM0 at sea | min | [m] | IMO, PRS | 46 | 59 |
| 3 | GZ (30°) | min | [m] | HSMB | 47 | 60 |
| 4 | Max GZ | min | [m] | IMO, PRS, HSMB | 48 | 61 |
| 5 | Max GZ angle | min | [deg] | IMO, PRS | 49 | 62 |
| 6 | GZ range | min | [deg] | PRS | 50 | 63 |
| 7 | Critical heeling lever | min | [m] | PRS | 51 | 64 |
| 8 | Weather criterion | min | mrad | IMO | 52 | 65 |
| 9 | Turning heel angle | max | [deg] | PRS | 53 | - |
| 10 | Static heel angle | max | [deg] | IMO | 54 | 66 |
| 11 | Area under GZ [0, 30] | min | [mrad] | IMO, HSMB | 55 | 67 |
| 12 | Area under GZ [0, 40] | min | [mrad] | IMO, HSMB | 56 | 68 |
| 13 | Area under GZ [30, 40] | min | [mrad] | IMO, HSMB | 57 | 69 |
| 14 | Area under GZ [30, range] | min | [mrad] | HSMB | 58 | 70 |

constraints on the background of the remaining constraints, significant in a vicinity of the optimum solution. It can also be thought of as a preparatory, qualitative part of sensitivity analysis. Table 5.3 presents a list of stability constraints together with their indices in the optimisation model implemented in the EUROS program.

Sea-river ships, as multipurpose craft, operate in their life as different types of vessels, both from the viewpoint of their functionality and the corresponding stability regulations. To cope with the problem three extreme situations have been recognised in the model as crucial ones :

(i) A ship operates as a „pure" container sea-going vessel. She then carries containers both in holds and on deck and is allowed to take water ballast into the double bottom and wing tanks.

(ii) A ship operates as a „pure" general cargo sea-going vessel. As such, she does not load cargo on deck but, on the other hand, she is not allowed to take water ballast into her tanks.

(iii) A ship operates as a „pure" container, river vessel. She then carriers containers both in holds and on deck but, as the total mass of ship must be smaller than that at sea due to restricted draught, the ship is not allowed to take a water ballast.

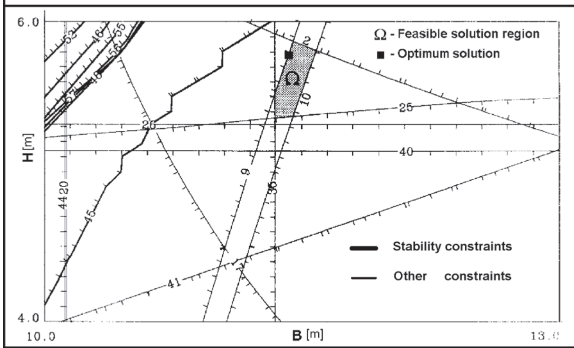
All the three situations have been accounted for by formulation of a triple set of corresponding stability constraints „acting" simultaneously. Such an approach assures that stability requirements will also be satisfied in all the intermediate loading conditions which can occur in operation. A feasibility analysis has been done in a graphical form using the method and software worked out in [9]. The results are presented in Tab. 5.4. The constraints are presented via visualisation of their contour lines in the vicinity of the optimum solution. The arguments of the graphs are ship's dimensions: B vs. H (1) and hull form ratios: B/T vs. H/T (2), the most essential ship parameters as far as the intact stability is concerned. The description in Tables 5.4 refers to the terminology introduced in Chapter 2. The stability constraints recognise three potential functional ship categories : a sea-going container carrier, a sea-going general-cargo ship and a container river ship. The feasible solution region **Ω** is shown as a darkened area and the „tufts" on the contour lines point in the unfeasible direction. The parameters of a fleet, which are not arguments of the graphs, are those in Table 5.2. The feasibility analysis is summarised in Table 5.5 where the attributes of stability constraints refer to those in Tab. 2.3.

It follows from Tab.5.5 that in the case under investigation the most restrictive stability criteria are those by PRS, where the criterion No. 50 (GZ range) appears as a boundary and active constraint.

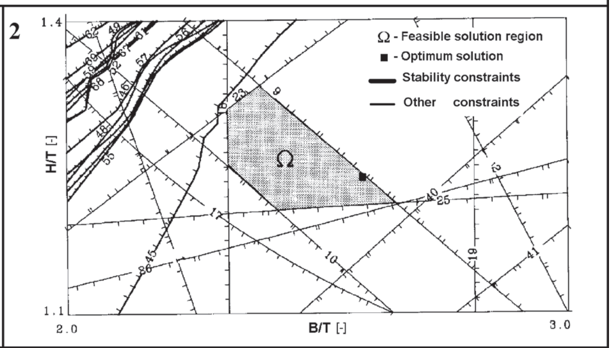
***Tab. 5.4.*** *Feasibility analysis - the case study*

**IMO**

1



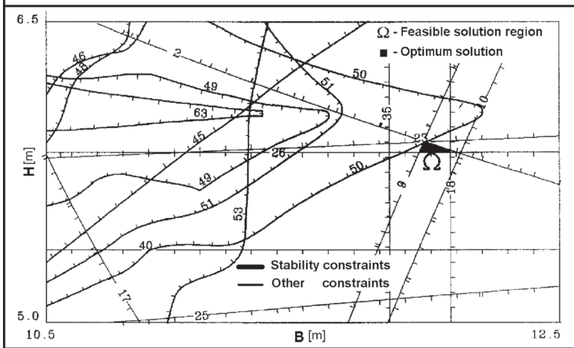

2



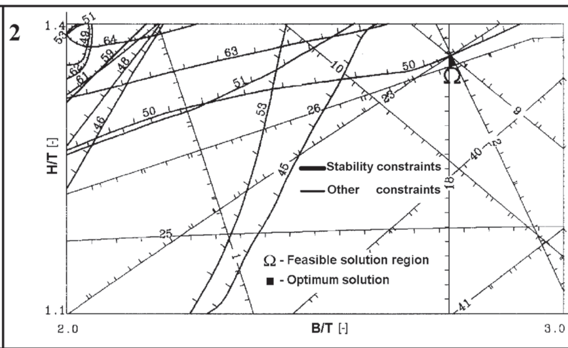

| Boundary constraints | Stability constraints | Boundary constraints | Stability constraints |
|---|---|---|---|
| 9<br>2, 10, 25, 35 | **cs**: 46, 48, 52,55,56, 57<br>**gs**: -<br>**cr**: 45 | 9<br>10, 18, 23, 25 | **cs**: 46, 48, 49, 52, 56, 57<br>**gs**: 59, 62, 67, 68, 69, 81;<br>**cr**: 45 |

**PRS**

1




2

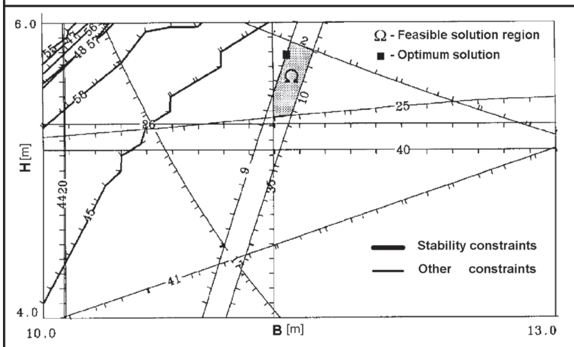



2

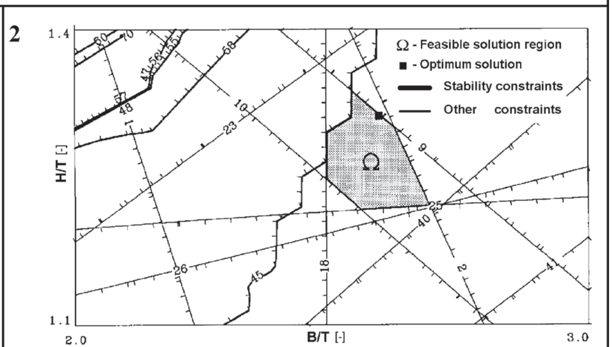
| Boundary constraints | Stability constraints | Boundary constraints | Stability constraints |
|---|---|---|---|
| 2, 9, **50**<br>23,26 | **cs**: 46, 48, 49, **50**, 51,53, 63<br>**gs**: -<br>**cr**: 45 | 2, 9, **50**<br>23, 26 | **cs**: 46, 48, 49, **50**, 51, 53<br>**gs**:59, 61, 62, 64, 63<br>**cr**: 45 |

**HSMB**

1




2




| Boundary constraints | Stability constraints | Boundary constraints | Stability constraints |
|---|---|---|---|
| 9<br>2, 10, 25, 35 | **cs**: 47, 48, 55,56,57<br>**gs**: -<br>**cr**: 45 | 9<br>2, 10, 18, 25, **45** | **cs**: 47, 48, 55, 56, 57, 58<br>**gs**: 60, 70<br>**cr**: **45** |

**Legend :** a - active constraint, p - passive constraint
cs - container ship at sea, gs - general cargo ship at sea, cr - container ship in river

**Boundary constraints :**

| | | | | | |
|---|---|---|---|---|---|
| Annual mass of cargo (INP) | [t] | (min) | Vertical container clearance | [m] | (max) |
| Mass reserve factor | [-] | (min) | Freeboard | [m] | (min) |
| Mass reserve factor | [-] | (max) | Height of bow | [m] | (min) |
| Length of forepeak ratio | [-] | (min) | L/B ratio | [-] | (max) |

***Tab. 5.5.*** *The status of stability constraints versus boundary constraints*

| | B vs. H | B/T vs. H/T |
|---|---|---|
| IMO | **a** | **a** |
| PRS | **c** | **c** |
| HSMB | **a** | **b** |

## 6. SUMMARY AND CONCLUSIONS

A post-optimisation study – stability-oriented feasibility analysis has been demonstrated on the background of a marine engineering design problem – optimisation of a fleet of sea-river ships for a regular shipping line in the area of the North and Baltic Seas. Two questions can be addressed based on the results obtained: (i) why the full intact stability criteria (such as IMO or similar) should be included in the optimisation model, and (ii) how to prepare such a model to accomplish the task.

As far as the first question is concerned,
the following justifications seem to be of importance :

- designers and potential customers (owners) express specific and growing interest in the safety of ships
- the stability criteria do exist as formal and legal design restrictions so they can not be neglected
- their presence in the model improves its quality, thereby making it more credible
- their incorporating into the model enables post-optimisation studies to be undertaken, such as parametric study, feasibility analysis and their quantitative extension - sensitivity analysis, with special emphasis on the economic aspects of the solution („shadow prices" and risk assessment).

As to the second question, an attempt has been made to define arbitrary stability criteria as constraints based on a complete analytical definition of all the necessary geometrical characteristics of hull form. A method has been proposed to use a systematic series of ship body forms. As an example, approximation of cross curves and accompanying characteristics of Series 60 has been demonstrated which makes it possible to determine a righting lever curve as a multi-variable function of the heeling angle and ship parameters – decision variables in the fleet optimisation model.

For a particular case under study – a fleet of sea-river ships, Series 60 turned out to be not suitable for the whole range of form parameters expected of such type of ships (very large $C_B$ and very small H/T). Consequently some corrections have been made to fit the available characteristics of SINE-205 sea-river vessel. Such an approach, however, must be regarded as a temporary solution. A research project is under way to work out a stability-oriented, analytically defined series of hull forms for specific functional types of ships such as full cellular container ships, tankers and so on, to be used in the corresponding fleet optimisation models.

**NOMENCLATURE**

| | | |
|---|---|---|
| B / b | - | breadth of ship / B/T ratio |
| **c** / **C** | - | vector / set of constants |
| $C_B$ | - | block coefficient |
| $g_j(\cdot)$ | - | function of a $j^{th}$ constraint |
| H / h | - | depth of ship / H/T ratio |
| $\mathbf{J_B}$ / $\mathbf{J_S}$ / $\mathbf{J_R}$ | - | sets of indices of boundary / stability / remaining constraints |
| $L_H$ | - | heeling lever curve |
| $l_R$ / $L_R$ | - | non-dimensional / dimensional [m] righting lever curve |
| m | - | number of constraints |
| n | - | number of decision variables |
| Q | - | objective function |
| $\mathbf{R}^n$ | - | n-dimensional arithmetic space |
| $r_0$ | - | non-dimensional transversal metacentric radius |
| T | - | design draught of a ship |
| $w_B$ | - | cross curves function |
| **x** | - | decision variable vector |
| $z_G$ | - | non-dimensional centre of gravity of a ship |
| $\phi$ | - | heeling angle |
| $\Omega$ | - | feasible solution region |

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# Hull strength assessment of the designed ships

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ABSTRACT

*The paper presents the method and scope of analysis of the hull strength of four ships designed in the Eureka project. Criteria of the Det Norske Veritas rules for classification and construction of ships were used in the analysis. Specific features of the structure of each hull have been described and the resulting problems connected with ensuring adequate strength.*

**Keywords** : hull strength, FEM calculations

## INTRODUCTION

The strength analysis was performed
for the following four ships :

- product tanker (length overall $L_{OA}$ = 138.10 m, breadth B = 22.50 m, depth D = 12.80 m, draught T = 8.70 m)
- ro-ro ship ($L_{OA}$ = 156.72 m, B = 24.80 m, D = 19.60 m, T = 6.50 m)
- „river-sea" ship ($L_{OA}$ = 89.45 m, B = 11.40 m, D = 5.45 m, T = 4.40 m – at sea, T = 2.80m – in river)
- container carrier ($L_{OA}$ = 138.10 m, B = 22.50 m, D = 11.20 m, T = 8.55 m).

The design requirement of those ships was their ecological cleanness, i.e. a CLEAN DESIGN symbol given in the Det Norske Veritas class description [1].

The ecological cleanness of a ship is first of all connected with its installations and equipment. It does not impose any specific requirements on the internal subdivision or structure of the hull. A sufficient precaution is the use of classic solutions, i.e. double bottom and double side in the case of a product tanker. However, double bottom and double side have been used in all the designed ships as they are classic solutions for those ship types. They protect the sea against pollution in the emergency situations, e.g. grounding, collision with another ship etc. The ecological cleanness feature does not mean, however, that the hull strength standards have to be raised.

The structure strength assessment was performed
in accordance with the rule requirements [1, 2, 5].

The following structure strength levels had to be checked :

- overall strength
- zone strength
- local strength.

Additionally, the criteria of total normal stresses in the longitudinal girders and in longitudinal stiffeners of side shell, decks etc. from the general, zone and local bending have to be fulfilled.

The overall strength involves the longitudinal bending of the hull in the vertical plane. In the case of ships with wide hatch openings, also torsion and bending in the horizontal plane should be analysed. The calculations are relatively simple as the rules require a bent beam model to be used in the analysis of bending in the vertical or horizontal plane. In the torsion analysis, the use of constrained torsion theory of thin-walled bars is admissible. However, it is recommended to use a shell-bar FEM (Finite Element Method) model of the whole hull and then the calculations are time-consuming.

The zone strength involves bending of the hull girder system. Calculations are often time-consuming as in general it is required to use a shell-bar FEM model. The rules [2] generally allow to use also simpler models – space frames, grids, flat frames, continuous beams. Such calculations are still relatively time-consuming.

The simplest are local strength calculations. The respective formulae for plate thickness or stiffener section modulus are given in the rules.

In the above mentioned analyses the stress level is checked, by the permissible stress criterion. The structure should also fulfil the criteria given in the rules [1] for the structure element stability and fatigue life.

Specific hull strength problems of individual ships
are described below.

## PRODUCT TANKER

The most labour consuming problem of the hull strength
assessment was the zone strength analysis.

The cargo section of the hull is divided with transverse bulkheads into 6 compartments. Compartments no.2 to 6 (counting from the bow) are additionally divided with a longitudinal bulkhead in the hull plane of symmetry.

An FEM model in the form of space frame was created comprising a hull fragment of the length of ½ + 1 + ½ cargo compartment (Fig.1). The possibility of asymmetric liquid cargo distribution made it necessary to develop a full hull breadth FEM model.

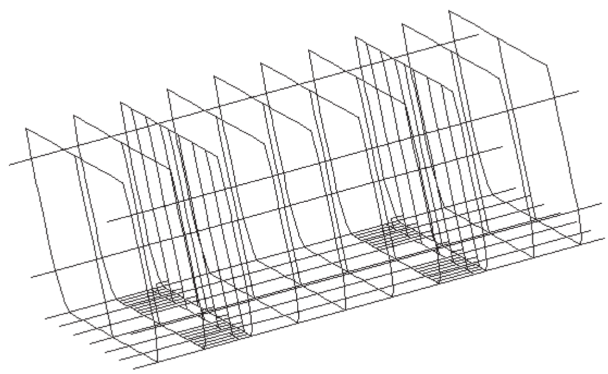

*Fig. 1. FEM model of a product tanker hull module frame*

Calculations revealed a high level of the reduced stresses in floors and transverse web frames coming from shear in the area of lightening and communication holes. It was necessary to correct the original version of the structure. The holes were made smaller and thicker plates were used. Also the original transverse corrugated bulkheads appeared not strong enough. They had to be made thicker.

Difficulties were met with ensuring local strength in the specific product tanker conditions of dynamic loads in partially filled tanks (sloshing). The problem was connected with the deck structure and upper parts of bulkheads in the first fore cargo tank. It was necessary to correct the original plate thicknesses and stiffener sizes as well as the main girders in that region.

A characteristic feature is that the elements of a product tanker hull structure (total length of 138.1 m) with double ship side shell and double bottom, without hatch openings in the deck, dimensioned according to the local and zone strength criteria, ensure automatic fulfilment of the general longitudinal bending criteria.

## RO-RO SHIP

Characteristic solutions of the ship are the following :

- three loading levels (inner bottom, main deck, upper deck with the loading ramp hatch openings)
- no transverse bulkheads in the cargo section of the hull
- double bottom, double side from bottom to main deck and single side above.

Assessment of the zone strength of such structure was performed by means of simple FEM models in the form of flat frames and grids loaded with the container or trailer weight. Additionally a shell FEM model was created of a typical web frame (Fig.2) in order to verify the structure correctness in the bottom and side joint region with communication openings.

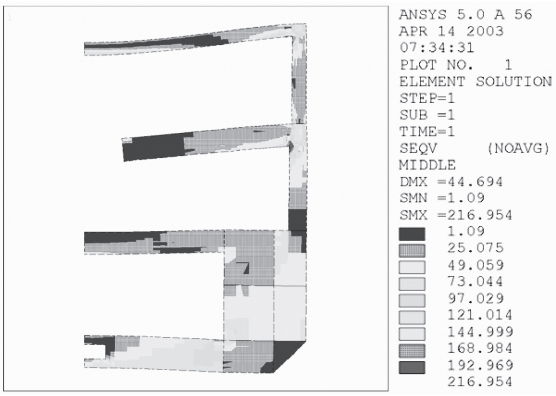


***Fig. 2.*** *Transverse web frame shell FEM model*

Calculations indicated the necessity of strengthening that structure region in relation to the original version. Also the upper deck web beams in the side region appeared too weak. Their section modulus had to be increased.

In the case of a ro-ro ship, the deck shell and stiffener dimensioning load is that of the vehicle wheel load. The use of 80 t and 60 t roll-trailers and 30.5 t carrying capacity container stackers was assumed. That gives relatively great wheel load values. Increase of the originally proposed cargo deck shell thickness and beam strength appeared necessary.

As in the case of the product tanker, the structure dimensioned according to the local and zone strength criteria automatically fulfilled the general overall strength criteria. Important is here the relatively large depth of the ship and lack of hatch openings in the middle part of the hull.

## RIVER-SEA SHIP

This is the smallest
of the four designed ships ($L_{oa}$ = 89.45 m).

Characteristic features of the hull are: wide hatch opening as long as 62 m over the only hold of the ship, strong double bottom adjusted to relatively large local loads from cargo, double ship side and high hatch coamings on the main (single) deck.

The structure dimensioning loads
are those during the operation at sea.

It is interesting that the overall strength problems occurred in the case of a small ship. Normal stresses in the upper fibres of hatch coamings reached, in the general bending conditions, values near the admissible level.

In the case of that ship type, important is the problem of hull torsion on a course oblique to sea waves. Torsion analysis was performed with the use of thin-walled bar constrained torsion theory [5]. Torsional displacements of the hull appeared small but stresses reached significant values.

The greatest normal torsional stresses occurred in the hatch rear end area, in the upper edge of hatch longitudinal coamings (75 MPa). Total normal stresses in that place, from torsion, hull bending in the vertical and horizontal plane and from deck strake bending in the horizontal plane (due to bottom and side loads) reached the admissible stress level, i.e. 195 MPa.

In the zone strength analysis, it was necessary to use a FEM model in the form of a space frame, containing main structural elements in the whole cargo part of the hull and across the whole breadth of the hull, in order to analyse e.g. the structure stresses in the conditions of ship heel with containers on deck (Fig.3).

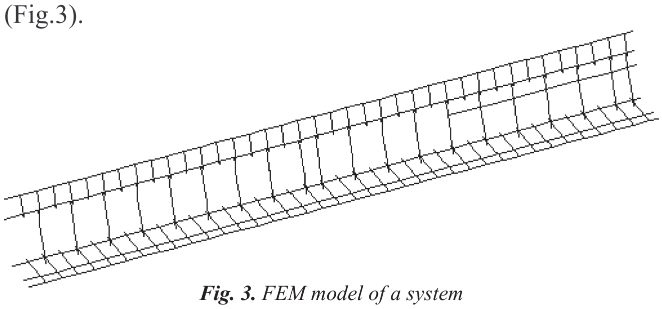

***Fig. 3.*** *FEM model of a system of structure elements in the river-sea ship hold*

With the use of such computational model a relatively great deformability of the hull was demonstrated. Convergence of the ship sides in certain loading conditions was estimated at 65 mm, which may require special hatch covers to be used acting as hatchway beams.

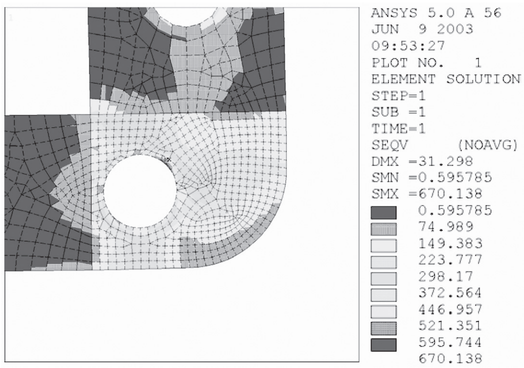


***Fig. 4.*** *Shell FEM model of the bilge region of river-sea ship hull*

Computations showed also exceeded admissible stresses in the bilge region floors. Therefore, a more precise (shell) FEM model of that region was prepared (Fig.4). Calculations have shown that holes are inadmissible in that region.

Standard local strength calculations were also performed in the hull strength analysis. The design requirement that the ship should be able to transport iron ore, with its approximately 100 kPa static load on the inner bottom, resulted in a relatively massive structure of that hull fragment.

## CONTAINER CARRIER

A characteristic feature of that ship hull
are wide hatch openings and narrow double sides.

There are 5 hatch openings in the deck, connected with batten plates. Such hull configuration causes the cross section neutral axis to be positioned much closer to the base plane than to the deck.

There were some problems with adequate general bending strength in the vertical plane. In the case of a small ship ($L_{oa} \approx \approx 140{,}0$ m), it appeared effective to provide high tensile steel (315 MPa yield point) girders in the middle part of the hull in the deck region.

The rules [1] require that torsion analysis be performed for ship hulls of the above mentioned characteristics. The calculations were carried out with a twisted bar model. Bending of a frame created by the side deck strakes and the batten plates (webs of the frame beams) was taken into account. The beam flanges are hatch coamings together with the internal and external side fragments or transverse bulkheads below deck. The calculations were performed in accordance with the procedure required in [6].

Normal stresses from bending of the deck strakes due to deformation of the above mentioned frame (hull cross section torsional deplanation) reached the 30 MPa level. This is a significant value.

The total normal stresses from hull bending in the vertical and horizontal plane, torsion (due to constrainment of the deplanation), deck bending (as described above) and deck bending from sea water pressure on ship sides, reached a 231.5 MPa level, which is as much as 95% of the admissible value. Those stresses occurred on the top of the no. 5 hatch rear end coaming.

Container carrier torsion is therefore
a significant problem, even in the case of a small ship.

The zone strength analysis was performed with a space frame model of the structural elements of halves of two adjacent holds (Fig.5).

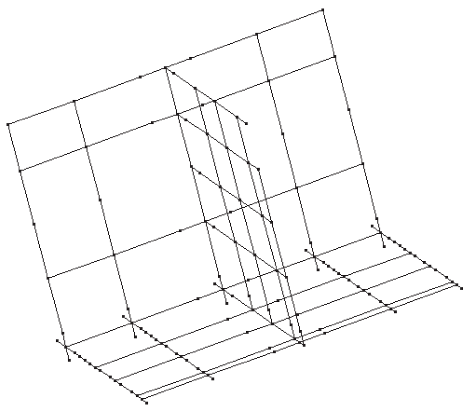

*Fig. 5. FEM model of the container carrier structure elements*

Also an FEM shell model of a single web frame was developed (Fig.6) in order to check the stress level in the web communication opening areas.

Standard local strength calculations were carried out in the same way as for the other ships. In the case of container carrier, it was necessary to check the strength of the bow section bottom structure subjected to slamming loads. This is connected with the relatively high service speed of those ships.

The calculations showed that the bottom plating
in the original version of the ship design was too thin.

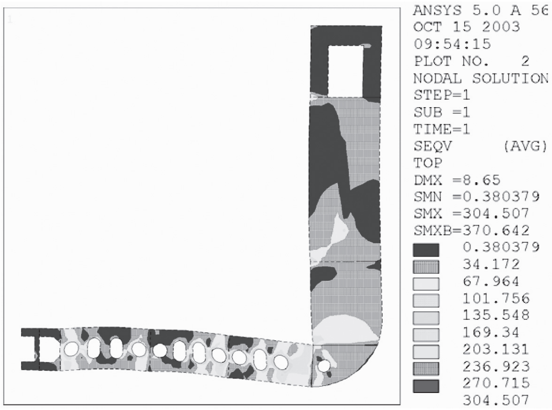


*Fig. 6. FEM shell model of a container carrier frame*

## FINAL REMARKS AND CONCLUSIONS

- Four different ships have been designed in the EUREKA project. They are intended for different service and therefore their hulls are subject to specific loads.
- In the case of a product tanker, this is the sloshing pressure having a significant impact on the shell plate thickness as well as on the size of stiffeners and structural elements of ship sides, bulkheads and deck, particularly in the fore part of the ship.
- In the case of a ro-ro ship, characteristic are the vehicle wheel loads on decks.
- In the case of a relatively small river-sea ship, problems occurred with providing adequate general strength in the bending and torsion condition, with bottom strength under considerable local loads from cargo, and with the hull stiffness.
- The container carrier hull is significantly strained in the overall bending and torsion condition. In this case, the use of high tensile steel for girders in the upper part of the hull appeared effective.
- A common feature of all the hulls is double bottom and double sides – at least below the main deck.
- The ship hull design process proceeded in a rational way. The preliminary scantlings proposed by the designers were corrected „upwards" as a result of the performed strength analyses.

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# Designing engine rooms of new generation ships realized within the framework of European research projects EUREKA - chosen questions

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ABSTRACT

*Importance of reliability and safety in the operation phase of engine rooms has been proven. A concept of designing engine rooms taking into consideration their reliability, safety of functioning as well as ecological features has been proposed. The need for as well as the main principles of carrying out empirical research in particular ship design stages have been introduced. Possibilities of the use of the semi-Markov theory in designing engine rooms have been justified. Possibilities of formulation and importance of hypothesis in scientific research related to engine rooms of sea going and inland ships depending on their particular specific features have been introduced. Examples of such hypothesis and their verification methods have been given. Possibilities of the use of semi-Markov processes in optimization of the ship operation have been signalled.*

**Keywords** : ship power plant, sea-going ship, reliability and safety, design, semi-Markovian process

## INTRODUCTION

Ship engine rooms should be designed so that, assuming that the ship carries her tasks, she could be operated with in the highest in these conditions (if possible – optimum) rentability but keeping proper reliability, required safety of motion as well as meeting ecological measures. This is especially important for transport ships operating in relatively restricted sea areas such as the Baltic Sea. Thus problems of economic functioning of engine rooms of ships taking into consideration their reliability should be judged as the most important.

Design solutions used in engine rooms can be different but the general way of forming their reliability (understood as the choice of types of their activities as well as the choice of possible resources ensuring reaching needed level of reliability) can in principle be the same. Building the reliability of each ship engine room in the design phase is based on consecutive solution of problems resulting from the need to obtain answers to the following questions :

- What criteria of creation of reliability should be adopted ?
- What is the risk of excessive increase of the cost of designing in case of starting an original design of engine room ?
- What could be the cost of obtaining the required level of reliability of an engine room similar to an already designed and operated engine room ?
- How to divide the machines and equipment of the engine room in order to shorten the design phase and in the same time to obtain the required reliability ?
- Against which requirements should the reliability be considered ?

This is why scientific research is indispensable in the design stage of ship engine rooms. Such research is also indispensable in further stages of existing of an engine room, namely during its building period as well as operation. Such scientific research enables to create scientific knowledge, i.e. such a knowledge which is justified using scientific methods of its creation. In result of such research information of cognitive as well as useful value can be obtained. In the first case new knowledge is created in process of verification of hypothesis and/or proving thesis explaining relations between new facts, ascertained during research carried out during operation of engine rooms of particular ships, used by different shipwoners, with the existing scientific knowledge. The second case refers to the sphere of scientific knowledge which is used in practice as to designing engine rooms as well as other structural nods e.g. ship hulls. This way the number of true information creating useful knowledge but checked using scientific methods is increased and in the same time the number of so called "common sense knowledge" information is decreased. This way useful knowledge becomes a more precise tool which can be used in the design processes and later on in the phases of building and operation of engine room [4,8,9,17,18 and 19].

The importance of scientific research in designing engine rooms is based on the fact that their introduction enables to explain facts ascertained during designing, building and operation of already existing engine rooms and using desirable solutions in the phase of consecutive designing. The importance of such research is also based on the fact that during repair of engine rooms they should be modernized using knowledge of designers who followed up the construction as well as the operation of the engine rooms.

To show this importance one should first signal the contemporary design concept of ships and navy vessels.

## OUTLINE OF THE DESIGN CONCEPT OF MODERN SHIPS VERSUS SCIENTIFIC RESEARCH

Deterministic design methods are used in contemporary design of engine rooms (as well as of ships treated as a whole) [1,8,15]. Although the use of these methods enables to prepare designs leading towards building engine rooms with a level of reliability judged intuitively as high (in principle the measure of such assessment is subjective or psychological probability) but does not allow to present such reliability in form of reliability factors. Such indicators even in form of probabilities (logical or statistic) are indispensable for rational planning and later on control of the process of operation of engine room. This is why there is a need to develop probabilistic methods shaping the reliability of engine rooms and to use them in the engine room design phase [6,7,17,18 and 19].

Probabilistic methods enabling shaping reliability and functional safety including proper ecological safety can be developed using the contemporary theory of reliability and safety of complex technical systems (engine rooms obviously can be treated as such systems), probability calculation methods, mathematical statistics, the theory of machines (mainly thermal machines) installed in engine rooms as well as using the theory of semi-Markov processes. In turn in order to prepare a set of reliability conditions of engine rooms one should use technical diagnostics as well as the results of damages of machines and equipment used in similar engine rooms. The knowledge of reliability and safety indicators of particular engine rooms enables to take rational decisions related to operation.

During operation of sea going ships decisions are taken always in stochastic decision situation i.e. in conditions of uncertainty (statistic risk conditions). This means that the probability and induction (mathematic) statistic rules are to be used when making decision. Decisions concerning operation are taken before commencement of the operation as well as during operation of the mentioned ships. This means that at least once such decisions are taken basing on initial information (obtained for example from research of reliability of engine rooms as well as of their particular machinery and equipment or basing on databanks of similar engine rooms). Such decisions can be named "a priori" decisions. Following decisions are based on information obtained during operation of the technical systems (e.g. in result of diagnostics, not only of technical diagnostics) and can be described as "a posteriori" information.

Decisions taken at the beginning of the operation are indispensable to plan the process of operation and maintenance of the said ship engine rooms. When taking decisions one should consider the statistic risk estimated as probability of taking wrong decision resulting from [2,7,9,19] :

- ⊙ impossibility of precise assessment of unknown parameters of the distribution of random variables, especially of such variables, which represent conditions of the operation process of the engine room and of particular machinery and equipment
- ⊙ lack of possibility to prepare full and/or reliable enough information indispensable to take the right decision.

The first case generates stochastic type of errors being subject of the so called **statistic precision of conclusions** while the second – both random mistakes as well as such which cannot be described as random (systematic). Determining the latter errors is a subject, which I suggest to name as **problems of precision or scrupulous way of concluding**. In total determining such errors is a question described as **accuracy of statistical conclusions. Accuracy of conclusions** results from the present level of scientific and applied knowledge, while **scrupulous way of concluding** results from insufficient appreciation or neglecting some information by the decision maker, who first could have checked that the information was really not so important. On the other hand decisions made during ship engine room operation are erroneous or irrational due to difficulties in preparation of full diagnosis and lack of proper credibility of information related to the technical conditions of these engine rooms, their machinery and equipment as well as to the foreseen external conditions (atmospheric and sea conditions), which could be expected during the ship's voyage [3,5,10,12].

In the presented decision situation taking rational decisions is possible in case of use of statistic theory of decision [4,5,18,19]. However determining the set of decisions, which can possibly be taken in accordance with the assumed criterions of optimization, requires identification of problems of assessment of reliability of the engine room, at least of its main energetic systems.

Different reliability and safety indicators of engine rooms, its machinery and equipment can be considered in designing engine rooms.

The most useful reliability indicators are [6,19] :

- ★ probability of correct work until the first damage
- ★ probability of correct work between two consecutive damages
- ★ single-dimensional distribution of process (instantaneous distribution) the elements of which are $P_k(t)$functions describing the probability that in moment *t* the process would attain condition *k*
- ★ limit distribution of the process $P_j = \lim_{t\to\infty} P\{Y(t) = j\}$
- ★ conditional probability described as probability of transition of the process from *i* to *j* condition, $P_{ij}(t) = P\{Y(t) = j / Y(0) = i\}$ (probability of transition)
- ★ distribution of the time of the first transition from *i* to subset of A $[\Phi_{iA}(t)]$ conditions, if such subset is a one element set – to condition *j* , i.e. condition $\Phi_{ij}(t)$
- ★ distribution $\Phi_{jj}(t)$ of the time of return of the process back to condition *j*
- ★ asymptotic distribution of the renewal process $\{V_{ij}(t) : t \geq 0\}$ generated by the distance in time of the return of the stochastic process (to condition *j* obtained from condition *i* , which in the moment *t* takes the value equal to the number of "entries" of the process into condition *j*
- ★ approximate distribution of the total time of the presence of the process Y(t) in condition *j* provided that condition *i* is the initial condition
- ★ expected value $E(T_i)$ of the time $T_i$ of i-duration of the process condition independently to which condition the transition is being effectuated in moment $\tau_{n+1}$
- ★ variance $D^2(T_i)$ of the time $T_i$ of the duration of i-condition
- ★ expected value $E(T_{ij})$ of the time $T_{ij}$ of i-duration of the process condition provided that condition *j* will be next condition
- ★ expected value $E(\Theta_{jj})$ of the random variable $\Theta_{jj}$ describing the time of return of the process to condition *j*
- ★ expected value $E\{V_{ij}(t)\}$ of random variable $V_{ij}(t)$ describing the number of "entries" of the process into condition *j* in the range [0, t]
- ★ variance $D^2\{V_{ij}(t)\}$ of the random variable $V_{ij}(t)$
- ★ average number of "entries" $\lambda_{ij}(t)$ of the process into condition *j* occurring per time unit provided that the initial condition of the process is condition *i* (i.e. intensity of "entries" of the process into condition *j* provided that Y(0) = i)
- ★ limit intensity of "entries" of the process into condition *j* i.e. intensity $\lambda_{ij} = \lim_{t\to\infty} \lambda_{ij}(t)$ .

Reaching numerical values mentioned in characteristics is possible only in case when the following conditions are met :

- ❋ Appropriate statistics have been gathered. The set includes values of assessed probabilities of transition $p_{ij}$, of the expected value $E(T_i)$ etc.
- ❋ Construction of a stochastic model of the process of operation of technical objects having small number of conditions

and not much complicated number (in mathematical sense) of transitions (changes) from one condition into other condition.

All the above mentioned types of reliability indicators are important for the operator but the most useful and in the same time the most easy to determine are the first two indicators.

Among the most useful safety indicators one can mention :

- expected value of random variable $N_s(t)$, i.e. the number of damages of the machinery and equipment of the engine room within the time range [0, t], causing a complicated situation describer with symbol $E\{N_s(t)\}$
- expected value of random variable $N_n(t)$, i.e. the number of damages of machinery and equipment of engine room in range [0, t], causing a dangerous situation described with symbol $E\{N_n(t)\}$
- expected value of random variable $N_a(t)$, i.e. the number of damages of machinery and equipment of engine room within the time range [0, t], causing a emergency situation described with symbol $E\{N_a(t)\}$
- expected value of random variable $N_k(t)$, i.e. the number of damages of machinery and equipment of engine room within the time range [0, t], causing a catastrophic situation, described with symbol $E\{N_k(t)\}$
- expected value of sail outs of the ship per one sea accident, described with symbol $E\{N_w(t)\}$
- expected value of sail outs of the ship per one sea catastrophe, described with symbol $E\{N_t(t)\}$
- expected value of death casualties per one sea catastrophe, described with symbol $E\{N_d(t)\}$
- expected value of death casualties per one sea mile, described with symbol $E\{N_m(t)\}$
- probability of safe ship (engine room) movement at sea $P_b(t)$, within the time range [0, t]
- probability of non-occurrence of a complicated situation of the engine room at sea $P_s(t)$, within the time range [0, t]
- probability of non-occurrence of a dangerous situation of the engine room at sea $P_n(t)$, within the time range [0, t]
- probability of non-occurrence of an emergency situation of the engine room at sea $P_a(t)$, within the time range [0, t]
- probability of non-occurrence of a catastrophic situation of the engine room at sea $P_k(t)$, within the time range [0, t].

It is obvious that in order to estimate the mentioned expected values of particular random variables $N_i(i = s, n, a, k)$ one should apply interval and not point estimation because only then the error of estimation can be determined. Determining the mentioned expected values is relatively simple. During investigations within the time period [0, t] one can determine the average (arithmetic) value $\overline{n}_i$ being the observed statistics $\overline{N}_i$, which as it is well known has a normal asymptotic distribution

$$N\left(m_i;\frac{\sigma_i}{\sqrt{n_i}}\right)$$

where :

$m_i$ - expected value (average)
$\sigma_i$ - average (standard) deviation of the random variable
$N_i$ , $n_i$ - number of registered events [1,4,16,18].

Reliability and safety requirements concerning functioning of the given engine room are as important as the technical and economic requirements formulated by the shipowner ordering a new ship in order to obtain the highest possible rentability of the ship as well as efficiency of its propulsion. Considering reliability and safety factors of a ship's engine room at the stage of design enables to influence its readiness to start up in any moment of operation, which should be maintained using possibly lowest operation cost. Thus economic calculations should be considered in the design stage in such a way as to control the process of engine room operation when in the same time maximizing the income. However this economic goal cannot be met without considering in the design process all the conditions of construction as well as operation and service of the engine room. This means that there are two staged which should be differentiated in the design of engine rooms [1,8,14] :

- ★ The first stage encompassing activities enabling designing of the process of fulfilling the need for production of energy indispensable to ensure the movement of the vessel as well as social needs of the crew and passengers, proper organization and control of work process proper for the design of engine rooms.
- ★ The proper design stage i.e. the stage of creation of new values in result of which is created a more or less original design of the engine room is created. Said design stage is needed to realize the above mentioned process of meeting the need for energy production.

The second design stage is finished with constructing, which is leading to creation of the final form of the designed engine room, namely to the creation of the final ship structure.

In both mentioned design stages one should consider such properties of the engine room as: functionality (functional correctness), safety, efficiency, reliability, durability, easiness to make diagnosis, renewability, ability to be controlled, life expectancy, ergonomics, ecological aspects, low level of noise and vibrations and toxicity of exhaust gases.

In the so understood design process one should assume that the basic information for design of engine room is the information which is related to the performed targets and conditions in which it can be realized. This means that one should first design the process of the engine room operation. The process will be a discreet process in conditions and continuous in time. It results from the hitherto considerations that a semi-Markov process, especially of controlled semi-Markov can be a model for of such a process [3,4,5,6,7,17,19]. Therefore the theory of semi-Markov processes, can be used in the process of designing ship engine rooms.

## SEMI-MARKOV PROCESSES AS MODELS FOR REAL PROCESSES CONSIDERED IN THE PHASE OF ENGINE ROOM DESIGN

The use of the theory of semi-Markov processes to create semi-Markov models of real processes as semi-Markov processes can take place only when [4,6,19] :

- ✷ the Markov condition is met, what means that the evolution of conditions of the investigated objects (e.g.engine rooms) in the future, for which the semi-Markov models have been built, should depend only on the present condition of the given object (from the condition in the present moment) and not from the functioning of the object in the "future", what means that the future of the object should not depend on its "past" and only on its "present condition".
- ✷ random variables describing the time intervals when the subjects of investigations remain in particular conditions reveal different distributions than exponential.

Thus when modelling, which has to lead to preparation of a semi-Markov model of the engine room operation of any ship, one should consider the change of real conditions of the process of the given subject of investigations.

In case of engine room of each vessel its operation process can be interpreted as a process of simultaneous changes of technical and service conditions [4,6,7,19]. Designing such process of engine room of any ship requires to solve using scientific methods the problem of forecast of particular conditions of the process. This requires to assume the following hypothesis (H1) : **prognosis of the condition of the process of operation of any ship or marine vessel in moment $\tau_n + \tau$, when it is known in the moment $\tau_n$ is known because its condition considered in any moment $\tau_n$ (n=0,1,…,m; $\tau_0 < \tau_1 < \ldots < \tau_m$) depends in an important way directly from the previous condition and not from the conditions which occurred earlier and time intervals when the condition occurred.**

It has to be noticed that the formulated hypothesis does not include any contradictions, which would classify it from the logical point of view even before its checking.

The consequences of this hypothesis
are as follows [4,6,19] :

- ❖ probabilities ($p_{ij}$; $i \neq j$; $i,j \in N$) of the transition of the process of operation of engine room or any of its machinery or equipment from any conditions *i*, in which the process is at the moment, to any other condition *j* do not depend from the fact in which conditions the process used to be in the past
- ❖ intervals of the unconditional time of durations of particular conditions of ship engine room operation process *i* are stochastically independent random variables ($T_i$; $i \in N$)
- ❖ intervals of the duration of each possible occurrence of conditions i of the engine room process are random stochastically independent variables ($T_{ij}$; $i \neq j$; $i,j \in N$), provided that the next condition is one of the remaining processes *j* of the process.

The mentioned consequences show the probabilistic law of changes of conditions of the mentioned operation process. They are not internally contradictory and their logic truth does not allow for any doubts. Thus the condition of no contradiction of consequences is met. Therefore there is no reason not to understand the named consequences as one joint consequence K1 and to use it to check empirically the introduced basic hypothesis H1. This means to verify it in order to accept it or classify. The verification of the introduced hypothesis is based on experimental checking of the truth of the joint consequence K1. The verification of the introduced hypothesis H1 through experimental checking of the truth of the mentioned consequence K1 requires to assume the truth of the following syntactic implication :

$$H1 \Rightarrow K1 \tag{1}$$

In such a case one can apply noninduction (induction)
conclusion carried out according to the following scheme:

$$(K1,\ H1 \Rightarrow K1) \vdash H1 \tag{2}$$

The logical interpretation of the scheme (2) is as follows :
**if the experimental checking of the consequence K1 proved its rightness and if implication (1) is right, then hypothesis H1 is also right and can be accepted**. Concluding made in accordance with scheme (2) is a means concluding of reduction type, which is one of the types of induction concluding and thus does not lead to sure conclusions and is only probable.

In case when for any structural solution of ship's engine room, its machinery or equipment operated in any operation system it would result that empirical checking of the said consequence (K1) may contradict the formulated hypothesis (H1), then the conclusion about the truth of hypothesis (H1) can take place in accordance with the "modus tollens" scheme :

$$(\sim K1,\ H1 \Rightarrow K1) \vdash \sim H1 \tag{3}$$

The scheme of concluding defined
by the relationship (3) has the following interpretation :

**if experiment does not confirm the correctness of the sequence K1, then in case of correctness of implication H1 ⇒ K1, the hypothesis H1 is not correct and therefore cannot be accepted**.

Both ways of concluding (2) and (3)
have to be supported with statistical conclusions [6,7,19].

In service practice it is important to plan the preventive maintenance of particular engine rooms as well as of particular machinery and equipment with regards to the undertaken tasks and the need for gathering appropriate resources to carry out indispensable operations.

The application of semi-Markov processes as models of real operation processes of ship's engine rooms enables, already in its design stage, optimization of the time interval after elapsing of which one should carry out particular preventive operations. In this case one can consider two variants resulting from the necessity and possibility of considering two functions [4] :

- ❒ function $k_g : T_p \rightarrow G$, which describes the dependence between the readiness factor of the investigated object (engine room or any of its machinery or equipment) and time $T_p$ elapsing from the moment of the start of the use of the object, being in condition of ability to the moment of the start of the preventive operation
- ❒ function $k_d : T_p \rightarrow D$, which describes the dependence between the average income (or cost) per unit of time and time $T_p$ .

The first case results from the need for maximization of the readiness factor of the object (ship's engine room, its main engine or any other machinery or equipment), while the second – – from maximization of income and minimalization of cost. In these variants G and D mean in turn : readiness of the object to start carrying tasks and average income per unit of time.

The correctness of hypothesis H1 can be confirmed in case of verification and acceptance of a more detailed hypothesis. Among such hypothesis is the hypothesis H2 explaining the fact observed in practice that wearing of the trybologic sliding systems of marine machinery (e.g. diesel engines) is weakly correlated with time [2,9,16]. This observation allowed to forecast the technical condition of machinery basing only on its current condition with omission of earlier conditions. Explanation of this fact can be presented in form of the following hypothesis (H2): **the condition of any trybologic sliding system as well as the time of its duration considerably depend on the earlier condition and not on earlier conditions and time intervals of their duration because its load and both implied speed and increase of wear are processes having asymptotically independent values**.

The statement given in this hypothesis that **load and both implied speed and increase of wear are processes having asymptotically independent values** results from two obvious facts :

- there exists an exact dependence between load of the sliding trybologic systems and their wear [2,9]
- there is a lack of monotonous changes of the load of the sliding trybologic systems of machinery in longer periods of operation. Thus it can be assumed that the load of such systems is stationary [10,12,13].

Verification of the introduced thesis H2 requires to determine (foresee) the consequences, the occurrence of which can confirm the correctness of the formulated hypothesis. The consequences K2, which can be drawn from the hypothesis H2 are as follows [2,9,16] :

- irregular wear of particular sliding trybologic systems
- interlacing of realized wearing processes of the sliding trybologic systems
- the course of correlation function for a given sliding trybologic system such that with the increase of distance the function first quickly decreases and then oscillates around zero with a relatively small but gradually smaller amplitude in function of distance
- nearly normal distribution of increase of wear of the sliding trybologic systems in appropriately long time period of proper operation
- linear dependence between the variance of the process of wear of sliding trybologic systems and the time of operation.

The mentioned consequences show the probabilistic law governing the wear of sliding trybologic systems. The verification of hypothesis H2 can be done in a similar way as in case of verification of hypothesis H1. However in this case one should accept the correctness of the following syntactic implication :

$$H2 \Rightarrow K2 \tag{4}$$

and then to apply noninductive (inductive) concluding in accordance with the following scheme :

$$(K2, H2 \Rightarrow K2) \vdash H2 \tag{5}$$

what is known as reduction conclusion.

## FINAL REMARKS AND CONCLUSIONS

Carrying scientific research in order to prepare semi-Markov models of engine rooms adequate for operation processes already in the design phase of these objects gives the possibility to present to shipowners the concept of control of the above mentioned processes in accordance with optimization criterions, which are important for them. Among these criterions one can mention: maximum income, minimum operation investments, required (also maximum) technical readiness factor etc. To prepare this concept one can apply decision (controlled) semi-Markov processes i.e. such semi-Markov processes in which realization depends on the decisions taken in particular moments of changes of process conditions. The processes are subject of considerations of semi-Markov decision (controlled) process theory.

The control of the processes of engine room operation in the phase of operation enables maintaining proper durability, reliability and safety of functioning of the objects and of their ecologic features.

Preparation of the above mentioned models of operation processes is equal to determination of a set of information, which should be at the disposal of designer, builder and operator so that they could ensure rational control of real processes of such engineer rooms. Without doubt it will influence the progress in operation of engine rooms and thus the progress in shipbuilding technology.

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### *Acronyms*

| | |
|---|---|
| SPE KBM PAN | - Section on Exploitation Foundations, Machine Building Committee, Polish Academy of Sciences. |
| MCNEM | - TInterministerial Scientific Centre on Exploitation of Fixed Assets |
| ITE | - Operation Technology Institute |
| PWN | - State Scientific Publishing House |
| WN-T | - Scientific-Technical Publishing House |
| WM | - Maritime Publishing House |
| WOiO PG | - Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology |
| WSM | - Maritime University of Szczecin |

# A probabilistic model of environmental safety of ship power plant

**Roman Liberacki**, M.Sc., Eng.


ABSTRACT

*In the paper a probabilistic model of environmental risk of ship power plant is presented. This is a linear strategy model with an additional restraint. It can be used for assessing risk to marine environment, which results from ship power plant operation. The risk is a sum of component probabilities of exceedance of the limits assigned - by MARPOL convention - to particular kinds of pollution discharged from ships, multiplied by weight factors. The factors determine a degree of harmfulness of a discharge for the environment. The restraint concerns the case of exceedance of the limits assumed unacceptable by the convention. A risk value is be contained within (0,1) interval. Moreover a criterion for environmental safety of ship power plant was propsed. This is the criterion of the ALARP class, in which an intolerable risk level and acceptable one is distinguished. Suggestions concerning determination of the levels are submitted.The considerations are illustrated by results of example computer investigations of influence of reliability of technical elements of the systems responsible for environmental safety of a hypothetical ship power plant on environmental risk value. Three qualification levels of ship's crew were accounted for : high, average and low.*


**Keywords** : environment protection, environment safety, risk, reliability, ship power plant

## INTRODUCTION

Ships are technical objects which produce serious hazards to the seas and atmosphere. Importance of the problem is supported by the fact of establishment of IMO MEPC Committee which deals with problems of pollution from ships to the seas. It has been estimated that maritime economy and shipping share in 20% in total pollution amount discharged to the marine environment [3].

When analyzing safety of a technical object one assumes that the object is a component of the man-technology-environment system, and that losses which may be generated by the object within the system, are considered. The losses are unavoidable, however one can influence a level of risk of their appearance. It should be reduced to a size which one would be willing to tolerate or even to accept [1]. Different safety criteria which define permissible risk levels for different categories of losses, are proposed. It also concerns the ships and risk involved to the marine environment. The risk mainly arises due to harmful substances released from ships to the seas and atmosphere. It is obvious that environmental safety of a ship should be shaped when designing the ship. During its operation it can be only maintained at a certain level or corrected.

International conventions adopted by IMO and ratified by member states, as well as other legal instruments of a lower rank impose definite limits upon harmful substances discharged from ships. They impose necessity of installing special anti-pollution devices and complying with appropriate procedures of action to be undertaken by ship crews. The main legal instrument in this respect is the 73/78 MARPOL Convention [11].

Norway is especially active in the area of marine environment safety. This state elaborated a procedure called Environmental Indexing of Ships, submitted to IMO and implemented in the waters of its state responsibility [9]. The procedure makes different kinds of dues paid by a ship, dependent on pollution-preventing systems installed on it.

Both the requirements of the MARPOL Convention and the Norwegian procedure have one important drawback: they do not account for a random nature of risk created by ships to marine environment. The pollution-preventing systems are anthropotechnical, and their technical and human components have definite reliability. The fact that a ship is equipped with devices complying with the standards imposed by the Convention does not show itself that the standards are also fulfilled during ship's operation because the devices can be unservicable, wrongly used or simply switched off. Therefore a proposal is to supplement the deterministic requirements with probabilistic methods for assessing reliability and environmental risk of ships. The probabilistic model of environmental safety of ship power plant, elaborated by this author and presented in the paper, may serve as such a tool.

## PROBABILISTIC MODEL OF ENVIRONMENTAL SAFETY OF SHIP POWER PLANT

In the environmental safety model of ship power plant the kinds of pollutions which are dealt with by the MARPOL Convention standards, were taken into account. These are : oil pollutants, sewage, noxious components of exhaust gas emitted by piston diesel engines and waste incinerators, as well as cooling media ( freons).

The environmental safety of ship power plant is expressed by risk level it creates for the environment. The lower risk the greater safety. The risk measure associates probabilities of causing hazards to environment and their consequences into a single numerical value. A caused hazard is the emission of such pollution amount which exceeds its limit value imposed by the MARPOL Convention.

Noxious substances released to the environment during ship's power plant operation are of different forms of harmful influence on the environment. Due to complexity of the problem, to use weight factors for determining the consequences of released pollutions is most suitable. To each kind of pollution a weight factor reflecting seriousness of environmental losses due to its discharge, was assigned.

The probabilities and weight factors were joined together into a single value by means of the linear strategy model [5], to which a restraint was added. The strategy is of a compensating character. A high value of one component of the model is capable to compensate a low value of the other. Whereas the restraint has to select the ship power plants not complying with the Convention's requirements as those hazardous. An additional advantage of the assumed linear strategy is the simplicity resulting from its additive character. The environmental **risk measure** of ship power plant can be expressed as follows :

$$\begin{cases} R = \sum_{i=1}^{m} w_i \cdot P_i(X_i > x_{il}) \ , \ \sum_{i=1}^{m} w_i = 1 \\ \exists i \ \ P_i(X_i > x_{il}) = 1 \Rightarrow R = 1 \end{cases} \tag{1}$$

where :

- $X_i$ – amount of discharge of i-th kind of pollution (random variable)
- $x_{il}$ – limit amount of discharge of i-th kind of pollution, imposed by the Convention
- m – number of kinds of pollution, taken into account
- $w_i$ – weight factor for i-th kind of pollution
- $P_i(Xi > x_{il})$ – probability that the discharge amount of i-th kind of pollution will exceed the yearly limit amount of discharge permitted by the Convention.

It is worth observing that :

$$\exists i \ \ P_i(X_i > x_{il}) = 1 \Rightarrow R = 1$$
$$\forall i \ \ P_i(X_i > x_{il}) = 0 \Rightarrow R = 0$$

The first relationship corresponds with the situation when a ship does not comply with any of the Convention's requirements, i.e. it is not fitted with appropriate devices and systems. The maximum risk value is then equal to R = 1.

The second relationship corresponds with the situation when a ship and its crew are beyond reproach regarding environmental safety. The situation is obviously not real; then R = 0. Therefore the risk measure R is contained within the interval between 0 and 1.

The probabilities $P_i(X_i > x_{il})$, i = 1,2,...,m, which appear in (1) can be determined on the basis of singular models of environmental safety of ship power plant. 12 such models were elaborated :

- ★ for operation of fuel oil delivery to ship
- ★ for operation of fuel oil pumping between tanks onboard the ship
- ★ for operation of lubricating oil delivery to ship
- ★ for operation of lubricating oil pumping between tanks onboard the ship
- ★ for operation of transferring oily substances out of the ship
- ★ for packing (sealing) of stern tube,controllable pitch propeller boss, and controllable pitch propeller boss of thruster
- ★ for bilge water deoiling system
- ★ for ship sewage system
- ★ for ship diesel engines regarding their $NO_x$ emission
- ★ for ship diesel engines regarding their $SO_x$ emission
- ★ for ship incinerator
- ★ for ship refrigerating system.

For every kind of pollution a review of design, features, structure and modes of operation of a relevant technical system, was performed with taking into account crew's actions. Next, appropriate unserviceability and event trees were elaborated. This way cause - effect models were obtained. Each of the models contains – in its structure comprising the above mentioned trees - description of a scenario of events leading to triggering a hazard. A triggered hazard is the emission of such pollution amount which exceeds its limiting value imposed by the MARPOL Convention. The specific models make it possible to determine probabilities of violating the Convention conditions, i.e. of triggering the considered hazards. In the models any conscious, deliberate, unlegal pollution discharges by ship crews were not taken into account. Input values to the singular models are the unserviceability events of technical elements and human errors. The model tree structures are compliant with recommendations of the FSA method being developed within IMO activity [2, 12, 13].

To determine numerical values of the weight factors the Norwegian report on the Environmental Indexing of Ships was used [9]. However the there given values of weight factors were not applied in this work, because such pollutions as bilge water, lubricating oils, and oil residues were not accounted for in it. The report's authors asssumed that if the discharging to the sea a given kind of pollution is prohibited then it is not discharged in fact. They also assumed that the pollutions subjected to treatment (bilge water) are always cleaned up to a sufficient degree. They do not take also into account oil leakage from stern tube as it should be prevented by packing. This approach results from a deterministic viewpoint on the environment protection systems, without accounting for reliability of technical elements and human relability.

However in the Norwegian report in question, a very detail analysis of influence of different pollutants on the environment, was performed. Their impact to the environment, which results from their destructive mechanisms to the environment, range of influence and potential amount of discharge, was there linguistically described.

Values of the weight factors of detrimental influence of pollutants on the marine environment are given in Tab.1. The factors were determined by changing the linguistic variables characterizing the environmental impact into appropriate numerical values (col. 4 and 5). Next, for each kind of pollution the assigned numerical values (col.5) were summed up for particular destructive impacts as shown in col. 6. Finally, the values of col. 6 for particular kinds of pollution were divided by the total sum of the numerical values determined for all kinds of pollution ($\Sigma$ =72). This way the searched values of the weight factors, $w_i$ , (col. 7) were found. The sum of all weight factors equals one in accordance with the formula (1).

Incinerators are only used at high sea, far from land. Hence their influence is very limited. Amount of their exhaust gases is small relative to that emitted from the combustion engines.

## INVESTIGATION OF THE MODEL

The investigating subject was a hypothetical ship power plant of a typical technical structure complying with contemporary solutions. Its operational process also complied with that presently applied. To perform calculations the following assumptions were made :

➲ **Regarding risk of $NO_x$ excessive emission** :
in the power plant 4 combustion piston engines are installed : 1 main engine, and 3 auxiliary engines driving independent electric generating sets; no systems for purifying exhaust gas from $NO_x$ are applied.

***Tab.1.*** *Values of weight factors of harmful influence of polluting agents on marine environment Σ = 72 Σ = 1*

| Polluting agent | Pollution symbol i | Environmental exposure | Environmental impact | Numerical value of environmental impact | Sum of numerical values | Weight factor $w_i$ |
|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 |
| $NO_x$ | I | Climatic changes<br>Acid rains<br>Production of ozon<br>Limitation of plant vegetation | Mean<br>High<br>Very high<br>High | 3<br>4<br>5<br>4 | 16 | **0.223** |
| $SO_x$ | II | Acid rains | Very high | 5 | 5 | **0.069** |
| CFC and HCFC | III | Destruction of ozone layer<br>Climatic changes | Very high<br>Very high | 5<br>5 | 10 | **0.139** |
| Exhaust gases from incinerators | IV | Local air pollution | Low | 1 | 1 | **0.014*** |
| Heavy fuel oil | V | Pollution of beaches<br>Pollution of sea | High to very high<br>High to very high | 4 to 5 (4.5)<br>4 to 5 (4.5) | 9 | **0.125** |
| Fuel oil (diesel) | VI | Pollution of beaches<br>Pollution of sea | Mean to high<br>Mean to high | 3 to 4 (3.5)<br>3 to 4 (3.5) | 7 | **0.097** |
| Lubricating oil | VII | Pollution of beaches<br>Influence on sea organisms | Moderate<br>Moderate | 2<br>2 | 4 | **0.056** |
| Oil residues | VIII | Pollution of beaches<br>Influence on sea organisms | Low to high<br>Low to high | 1 to 4 (2.5)<br>1 to 4 (2.5) | 5 | **0.069** |
| Oily bilge water | IX | Pollution of beaches<br>Influence on sea organisms | Low to high<br>Low to high | 1 to 4 (2.5)<br>1 to 4 (2.5) | 5 | **0.069** |
| Sewage | X | Pollution of beaches<br>Influence on sea organisms | Very high<br>Very high | 5<br>5 | 10 | **0.139** |
| | | | | | **Σ = 72** | **Σ = 1** |

* Harmfulness of exhaust gases from incinerators was not determined by the Norwegian authors. For needs of the model a low value of environmental impact was assumed.

- **Regarding risk of $SO_x$ excessive emission** :
the ship in question 24 times a year enters a protected region in which using a fuel oil having sulphur mass content lower than 1,5% is required; beyond the region a fuel oil of sulphur content exceeding 1,5% is used on the ship; no systems for purifying exhaust gas from $SO_x$ are applied; ship's fuel system is common for all the engines (a change of fuel oil kind for one engine simultaneously applies to the remaining engines); hence the set of engines of the power plant in question should be considered as a single engine.
- **Regarding risk of freon release to the atmosphere** :
in the ship's refrigerating system a freon of HCFC group is used; repair and maintenance operations are carried out once a year, during which a release of freon may happen.
- **Regarding risk of of excessive emission of noxious substances resulting from the use of a waste incinerator** :
waste combusting operations by means of the incinerator are performed 15 times a year.
- **Regarding risk of heavy fuel oil release to the sea** :
the heavy fuel oil bunkering operations are carried out 12 times a year on the ship; the heavy fuel oil pumping operations between the ship's tanks are performed 300 times a year.
- **Regarding risk of (diesel) fuel oil release to the sea** :
the fuel oil bunkering operations are carried out 2 times a year on the ship; the fuel oil pumping operations between the ship's tanks are performed 24 times a year.
- **Regarding risk of lubricating oil leakage to the sea** :
the lubricating oil bunkering operations are carried out 4 times a year by means of oil supply piping; the filled tank is fitted with an air-escape pipe reaching over the deck; the oil pumping operations between the ship's tanks do not involve any risk of oil leakage to the sea; the ship is equipped neither with a controllable pitch propeller nor thrusters; oil leakage may only occur through stern tube packing.
- **Regarding risk of leakage of oil residues to the sea** :
the operations of transferring oil derivatives out of the ship are carried out 4 times a year; the ship is fitted with the oil/bilge water separator NEPTUN made by Warma, Grudziądz (Poland); the separated oil collected in the separator may be released to the sea due to a failure of oil discharge system of the 2$^{nd}$ chamber of the separator.
- **Regarding risk of release of bilge water containing more than 15 ppm of oil** :
the ship is fitted with the bilge water/oil separator NEPTUN; in the engine room cleaning operations with the use of detergents are performed 6 times a year.
- **Regarding risk of sewage pollution of coastal waters** :
the ship is fitted with LK biological-chemical sewage treatment plant, Warma, Grudziądz; the probability of ship's sailing over sea waters up to 12 NM from the nearest land amounts to 30 % ; the seawage treatment plant is operated in accordance with DTR recommendations.

## *Way of performing the model investigation*

Three levels of crew qualification were assumed: high, average and low. Probabilities of making human errors were determined by using tables elaborated on the basis of the THERP method [8] as well as calculations performed by using ASEP – – HRAP method [4] and TESEO method [6], [14].

The same reliability level $R_{te}$ was assumed for all technical elements of the investigated systems. For elements of the singular operation models of : bunkering, pumping, transferring out fuel oils, lubricating oils and oil derivatives, of combusting wastes as well as of entering a region protected against $SO_x$ emission, the reliability level concerns a single operation only. For technical elements of singular models of : packing, bilge water treatment system, sewage system, combustion engines regarding $NO_x$ emission, as well as refrigerating system, the yearly reliability level is assumed. For elements of automation and signalling systems the so called **reliability on demand** is assumed.

The investigation consisted in simultaneous changing reliability levels of all technical elements within the range from 0,9 to 1. On the basis of relevant singular models as well as values of the weight factors of harmfulness of pollution impact onto the environment, values of yearly environmental risk were calculated in function of the reliabilities of technical elements and crew qualification levels. Results of the calculations are shown in Fig.1.

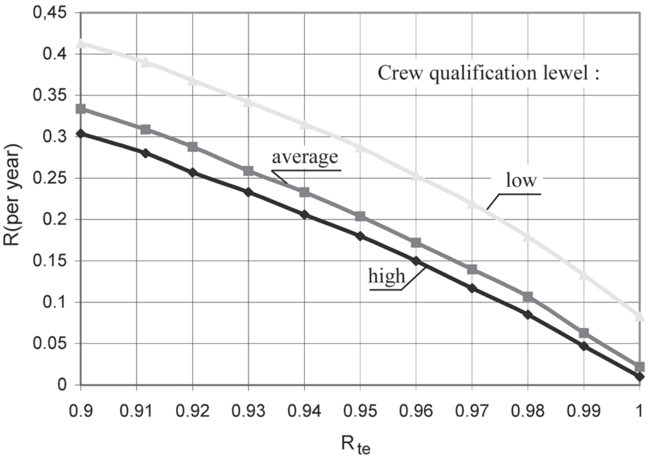


***Fig. 1.*** *Influence of the reliability level of technical elements, $R_{te}$ , and crew qualification levels on the yearly environmental risk of ship power plant, R.*

## *Conclusion derived from the example model investigation*

- ✶ The environmental risk of ship power plant, defined by the expression (1), obviously depends on the reliability of relevant technical elements as well as on the crew qualification level – the reliability of human operator.
- ✶ The greater reliability of technical elements and human reliability the lower environmental risk from the side of ship power plant, i.e. the higher level of environmental safety.

## A PROPOSED CRITERION FOR ENVIRONMENTAL RISK OF SHIP POWER PLANT

The proposed risk criterion follows the proposal elaborated in the frame of FSA method used by IMO. The entire risk range is divided into three categories :

- the intolerated risk
- negligible risk
- the so called ALARP (As Low As Reasonably Practicable) range.

The intolerated risk cannot be accepted in any case. The negligible risk is acceptable and no way of its mitigation is searched for. The risk within the ALARP range is subjected to an analysis. Possible lowering the risk level and costs associated with the possibility are considered. If lowering the risk level is practically possible and its cost is reasonable then actions aimed at its reduction are undertaken [7,13].

The division of the risk range into three categories requires two limit risk values : upper $R_u$ and lower $R_l$ , to be established. IMO carries out work on establishing such limits in the case of human life. They reflects risk produced by ships to human life. As far as the risk to the environment due to ship power plant is concerned such limits still have to be determined. A graphical interpretation of the risk criterion is shown in Fig.2.

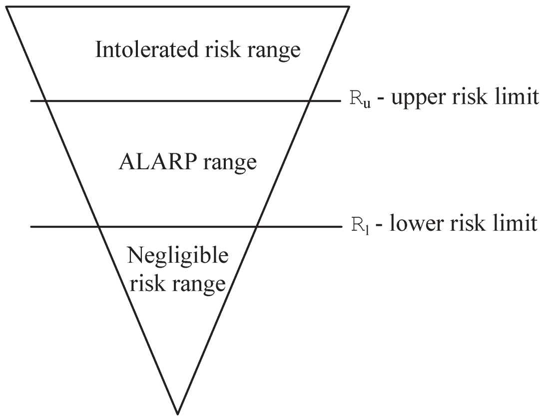


***Fig. 2.*** *A graphical interpretation of the risk criterion*

Here it is proposed the upper risk limit $R_u$ to be a value somewhat higher than the average risk according to the formula (1) for power plants of ships in service, under assumption of the average qualification level of ship crews. The proposal is based on the author's conviction that one should strive for improving present state of environmental safety. On the other hand it cannot be said that the risk level due to existing ship power plants complying with the international requirements, is not admissible.

A value of the lower risk limit $R_l$ can be also determined by using the formula (1), in this case for a ship power plant designed with taking into account possibly proecological solutions and ways of its operation, following the Norwegian ship's class notation Clean Design [10]. The notation does not take into account reliability of technical elements which should be on the level as high as possible. The human reliability would be assumed for the high qualification level.

In this case a proecological ship represents the ship the stern tube of which is filled with water instead of lubricating oil, on which oily bilge water and sewage are collected in tanks and transferred out to land – based stations, only diesel fuel oil containing less than 1.5% of sulphur is used, and freons are not applied to its refrigerating system.

## FINAL REMARKS AND CONCLUSIONS

- ❖ Randomness of the event of inadmissible pollution release from ship power plant is obvious. It implicates applying probabilistic approach to its environmental safety. The safety is a feature of the power plant which has to be assessed by using the notion of risk and safety criterion defined in safety engineering.
- ❖ Application of the elaborated models is closely associated with data on reliability of elements of environment protection systems; however they are unavailable in Poland. Perhaps they are saved in databases of Det Norske Veritas or Lloyds Register. Also experts – experienced ship engineers – have not sufficient experience to base on their opinion in

formulating credible assessments. Ship environmental protection systems are considered by ship engineers as auxiliary ones only, moreover they subject to failures rather rarely.

- ❖ The elaborated models make it possible to formulate reliability requirements for elements of environmental protection systems. The requirements can be put forward to producers of the elements, which are able to appropriately shape reliability of the elements as well as to verify it by means of laboratory tests or investigations in service. The problem is covered by PN-IEC 60300-3-4 standard. It consists only in formulating appropriate requirements and, certainly, in associated costs of their implementation.
- ❖ The models make it also possible to design reliability of the power plant systems responsible for releasing noxious substances – to a given amount only, as well as the entire power plant – to a given level of environmental risk.
- ❖ The environmental safety of ship power plant is a partial problem. The environmental safety of the entire ship is that mainly interesting. It seems that just the so defined models have a chance to be applied in the worldwide shipping. Hence the presented models exemplify only a methodical approach to solving the problem of environmental safety assessment.

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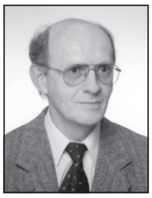
**ACRONIMS**

| | |
|---|---|
| ALARP | - As Low As Reasonably Practicable |
| ASEP-HRAP | - Accident Sequence Evaluation Program - Human Reliability Analysis |
| CFC | - (chlorofluorocarbons) |
| DNV | - Det Norske Veritas |
| DTR | - technical-operational documentation |
| FSA | - Formal Safety Assessment |
| HCFC | - (hydrochlorofluorocarbons) |
| IMO | - International Maritime Organisation |
| MARPOL | - International Convention for the Prevention of Pollution from Ships |
| MEPC | - Marine Environment Protection Committee of IMO |
| MSC | - Marine Safety Committee of IMO |
| PN | - Polish Standard |
| TESEO | - Tecnica Empirica Stima Errori Operatori |
| THERP | - Technique for Human Error Rate Prediction |

# A concept of the predicting of technical state of devices

**Janusz Lemski**, D.Sc.,Eng.

ABSTRACT



*The paper presents possibilities and limitations associated with technical state predicting procedure of devices, as well as it indicates the necessity of improving the present scheduling methods of preventive maintenance operations of ship power plant devices. It has been also showed that by making transformation of the wear rate distributions of device elements into the distributions of time of their correct- to-limit-state operation it is possible to predict their technical state and to suitably schedule their maintenance operations.*

**Keywords** : technical state, prediction, maintenance, mathematical transformations, probability distributions

## INTRODUCTION

In the process of predicting technical state of a device, in a result of which an optimum prediction containing a date and range of the next testing of the state is formulated, the device's user generates information which makes it possible to control keeping the device in the state of serviceability. An essential element of such control is to know prediction methods of device's state, containing procedures for scheduling a date and range of the next testing and the possible scope of overhaul operations resulting from it.

Due to a complexity of ship devices, their profitability and operational safety, the preventive maintenance strategy realized on the basis of their wear value is the most important. In order to elaborate such strategy it is necessary to choose an appropriate strategy for predicting working time resources of a given object with taking into account its service conditions [9,11].

When analyzing prediction methods it is not possible to unambiguously show a greater suitability of any of them against other ones because it depends on which kind of objects is subjected to predicting investigations [1,6].

In the present phase of knowledge and practical experience dealing with predicting problems in diagnostics, it is today hard to select an algorithmic procedure which would contain clear-cut recommendations and would exactly precise which methods have to be used for elaborating an optimum prediction in given conditions. Experience shows that in very many situations there is no general method which could be applied as a rule in realizing the process of choosing right diagnostic predictions.

If the ageing of the object is considered as a set of results of actions of processes limiting its service effectiveness, a question arises if it is possible to predict amount of wear of a repairable object and to assess cost-effectiveness of its further using without its replacement or carrying-out its major repair.

For this reason the attempt to present possibilities and limitations which concern the predicting of technical state of devices, as well as to indicate a necessity of improving the current methods of planning preventive maintenance operations of ship power plant devices, has been undertaken.

One of the possibilities may be to elaborate models of failure predicting and scheduling maintenance operations, in which wear phenomena of device elements would be accounted for. A way of transforming the wear rate distributions of device elements into the distributions of time of their correct-to-limit-state operation, was demonstrated. It makes it possible to estimate service life indices for devices, e.g. their working time resources, and to schedule their maintenance operations appropriately.

## POSSIBILITES AND LIMITATIONS OF THE PREDICTING OF STATE OF DEVICES

Physical ageing of devices usually occurs due to destructive processes taking place in them, associated with friction, corrosion, errosion, cavitation and fatigue of structural materials of their elements. Along with time of their operation it leads to often and often occurring failures and in consequence to lowering their reliability.

Service reliability investigations are aimed at determination of reliability function of an object, changeable with time of its operation, on the basis of data obtained in service.

Any decrease of reliability level cannot exceed a permissible value, hence it should be assumed that if such event happens the object in question will be taken out of service.

Below has been undertaken an attempt to present some essential possibilities and limitations associated with the predicting of reliability of devices , in order to show that searching for new predicting methods is necessary.

The irreversible wear processes occurring in devices in service cause a monotonically changeable trend of values of diagnostic control parameters to occur.

A basic condition to succeed in predicting the state of devices is to assume a uniform wear rate of their elements, i.e. that a trend of measured quantities is known (failure rate - in case of reliability considerations).

Changes of values of diagnostic parameters obtained from particular state investigations may differ seriously, and their probability density functions are usually not known.

Therfore it can be assumed that :

- the scheduling of the successive diagnostic test of a device is possible as a result of the technical state prediction which consists in determination of future changes of diagnostic parameters and in comparison their instantaneous values with those limiting [12,13,15]
- assessment of a trend depends on all observations of a variable, but a weight attributed to relevant realizations decreases in function of the time passing from the instant of observation up to now. It means that the longer time ago an event has occurred the smaller the weight, and that its importance decreases with the time passing from the instant of occurence of the event [5].

The main tasks for realization of the so described prediction process are the following :

- selection of optimum diagnostic parameters describing a current state and its change during operation of a device
- determination of a predicted value of a diagnostic parameter behind the time horizon of the diagnosis, by using an optimum diagnostic method
- scheduling the time of the sucessive diagnostic test.

When considering the real operation processes one has to do with both deterministic and random processes. The random processes cause that [8] :

- to formally describe phenomena is not possible or very complicated
- the considered phenomena are non-measurable, by convention
- the considered objects and phenomena are subject to continuous changes.

For this reason the following components appear in description of the real processes which occur during operation of a device :

- deterministic ones – possible to be calculated by using inductive methods
- stochastic ones – possible to be determined by using deductive methods on the basis of statistical data,
- or not being a basis for any predictions.

The reliability theory contains only a few theoretical models whose application makes it possible to obtain an exact description of influence of different factors on reliability characteristics of technical objects; therefore further elaborating and testing the mathematical models for reliability investigations of such objects, is justified.

However in any case an empirical mathematical model is simplified and it belongs to an a priori determined class. Usability and properties of some mathematical models important for reliability investigations of ship equipment are presented in [6]. In the case of ship equipment traditional methods for verification of statistical hypotheses by means of tests may be not sufficient to select the most suitable model. Knowledge of a kind of device failure (e.g. sudden or progressive) can facilitate the selecting of an appropriate mathematical model [6,17].

The systems for reliability prediction of objects, described in the subject-matter literature and used in practice, are based on investigations of influence of many factors which determine failure rate analytically described in an empirical model. Accuracy of such models and credibility of reliability predictions resulting from them, depends mainly on accuracy of empirical data obtained from laboratory tests or service investigations.

Wear phenomena in object elements are very complex and many factors influence their course therefore to take all of them into account is impossible; hence to make choice of an optimum set of diagnostic parameters is difficult.

Predicting the technical state of machines on the basis of values of diagnostic parameters changeable in service and related to a longer time period, is associated with a risk that the predicting diagnosis would be based on an out-of-date model, i.e. that whose elements (an optimum set of diagnostic parameters and an optimum prediction method) do not reflect real relationships between the technical state of the system and the predicting diagnosis, any longer . Hence the optimum predicting diagnosis should be stable within the entire period of state predicting. If it appears unstable then it may be applied to a dynamical operation system of machines. Otherwise should be made a decision to modify assumptions and limitations, e.g. by deliberate not-accounting for factors causing unstability and decreasing versatility of the obtained solution.

If in a performed predicting action the statistical information required for the used model is unavailable then it becomes necessary to apply a testing procedure which makes it possible to test in conditions of incomplete data. It is usually associated with the necessity to limit the range of predicting analysis to a short-range horizon.

Diagnostic prediction not always provides results expected from the point of view of its rightness and accuracy. The prediction rightness can be determined by probability of fulfilment of the prediction. The more fully and objectively are parametrized realization conditions of a process the greater the prediction rightness.

In the context of the presented possibilities and limitations it is necessary to search for ways to improve current methods of planning. One of them can be to elaborate such models in which wear phenomena of the object's elements would be taken into account. To this end may serve the transformation of the wear rate distributions of device elements into the distributions of their correct-to-limit-state operation, which would make predicting their technical state and appropriate scheduling their maintenance operations, possible.

## TRANSFORMATION OF THE WEAR RATE DISTRIBUTION MODELS INTO THE MODELS OF TIME OF CORRECT--TO-LIMIT-STATE OPERATION OF DEVICE ELEMENTS

To assess service life of devices at the initial stage of their operation is only possible by analyzing their wear processes, to this end failure predicting models accounting for wear phenomena of device elements, are the most suitable [14].

The reliability model of element's wear can be presented in the following form [14] :

$$\int_0^t f_1(t_s)dt_s = \int_{z_g}^{\infty} f_2(z)dz \qquad t \geq 0 \qquad (1)$$

where :

$f_1(t_s)$ – density function of the serviceability time T, for : $0 \leq t_s \leq t$

$f_2(z)$ – density function of the wear *z* for the fixed value $t \geq 0$

$t_s$ – current duration time of the process of the wear *z*.

The formula (1) expresses equality of the probability of the event consisting in that in the instant *t* or earlier a failure of the object occurred, and the probability that in the instant *t* the object attained the wear value $z_g$ or greater. The relationship (1) is illustrated in Fig.1 [14].

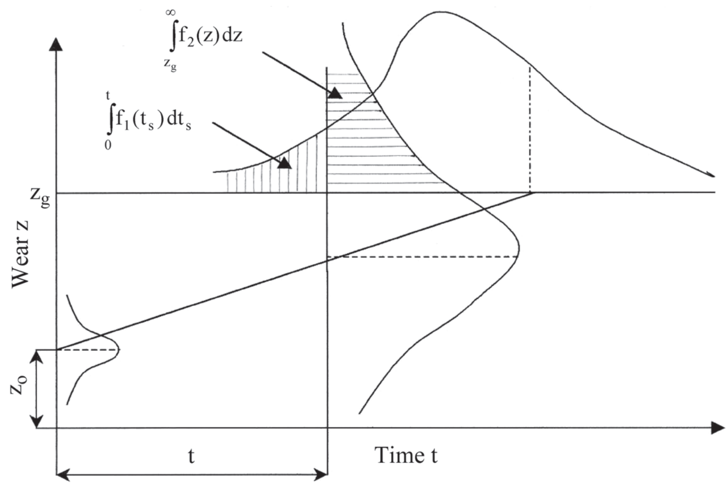



***Fig.1.*** *Schematic diagram of service life assessment on the basis of wear value taken as a random variable*

Making use of the equality (1) one can perform service life assessment by determining the density function $f_1(t)$ on the basis of distribution characteristics of the wear described by the function $f_2(z)$.

If the function $f_2(z)$ for the instant *t* is determined on the basis of observations of changes of wear distribution parameters, performed in the instants earlier than *t*, then by applying the equality (1) one can predict object's service life. To perform such prediction it is necessary to know the allowable wear value $z_g$ and stochastic model of wear process (see [14]).

A wear rate analysis of elements of ship devices shows that their distributions can be approximated by some theoretical distributions : normal, Rayleigh's and Weibull's one. The two-parameter Weibull distribution is the most versatile. In some special cases the exponential distribution can be also applied [2,4,6,7,15].

The wear rate distribution empirically or theoretically determined can be transformed into the distribution of time of correct-to-limit-state operation. By assuming that the wear rate distribution g(u) was obtained for the sufficiently large set of $n_o$ monomial elements, the number of elements n(T) which reached the limit state within the time t = T can be determined by means of the following equation :

$$n(T) = n_o \cdot F(T) \tag{2}$$

If $G(u) = P(U_g < u)$ is the cumulative distribution function of wear rate, and g(u) its density, then the unreliability function F(t) can be expressed as follows [2,3] :

$$F(t) = P(T < t) = P\left(\frac{z_g}{U_g} < t\right) = \\ = P\left(U_g > \frac{z_g}{t}\right) = \int_{z_g/t}^{\infty} g(u)du \tag{3}$$

Hence :

$$R(t) = 1 - F(t) = \int_{-\infty}^{z_g/t} g(u)du \tag{4}$$

Tab.1 presents the solutions of the equation (3) for the density functions of wear rate g(u) according to the normal, Rayleigh, Weibull and exponential distributions.

***Tab.1.*** *Transformation of wear rate distributions into the distributions of time to correct- to-limit-state operation*

| Wear rate distributions of elements | | | Distributions of time to correct- to-limit-state operation | | |
|---|---|---|---|---|---|
| Distribution | Parameters | Distribution density function g(u) | Probability of reaching the limit state, F(t) | Failure rate $\lambda(t) = \frac{f(t)}{1-F(t)}$ | Probability density function $f(t) = \frac{dF(t)}{dt}$ |
| Normal | $\bar{u}$ $\sigma$ | $\frac{1}{\sigma\sqrt{2\pi}}\exp\left[-\frac{(u-\bar{u})^2}{2\sigma^2}\right]$ | $1-F\left(\frac{z_g-\bar{u}t}{\sigma t}\right)$ | $\dfrac{\frac{z_g}{\sigma t^2\sqrt{2\pi}}\exp\left\{-\left[\frac{1}{2}\left(\frac{z_g-\bar{u}t}{\sigma t}\right)^2\right]\right\}}{F\left(\frac{z_g-\bar{u}t}{\sigma t}\right)}$ | $\frac{z_g}{\sigma t^2\sqrt{2\pi}}\exp\left[-\frac{1}{2}\left(\frac{z_g-\bar{u}t}{\sigma t}\right)^2\right]$ |
| Rayleigh's | c | $\frac{u}{c^2}\exp\left[-\frac{u^2}{2c^2}\right]$ | $\exp\left[-\frac{1}{2}\left(\frac{z_g}{ct}\right)^2\right]$ | $\left(\frac{z_g}{c}\right)^2 t^{-3}\left[\exp\left(\frac{1}{2}\cdot\frac{z_g}{ct}\right)^2-1\right]^{-1}$ | $\left(\frac{z_g}{c}\right)^2 t^{-3}\exp\left[-\frac{1}{2}\left(\frac{z_g}{ct}\right)^2\right]$ |
| Weibull's | a k | $\frac{k}{a}\left(\frac{u}{a}\right)^{k-1}\exp\left[-\left(\frac{u}{a}\right)^k\right]$ | $\exp\left[-\left(\frac{z_g}{at}\right)^k\right]$ | $\dfrac{\left(\frac{z_g}{a}\right)^k kt^{-(k+1)}}{\left[\exp\left(\frac{z_g}{at}\right)^k\right]-1}$ | $\left(\frac{z_g}{a}\right)^k kt^{-(k+1)}\exp\left[-\left(\frac{z_g}{at}\right)^k\right]$ |
| Exponential | $\lambda$ = const | $\lambda\exp[-\lambda u]$ | $\exp\left[\frac{-z_g\lambda}{t}\right]$ | $z_g\lambda t^{-2}\left[\exp\left(\frac{z_g\lambda}{t}\right)-1\right]^{-1}$ | $z_g\lambda t^{-2}\exp\left(-\frac{z_g\lambda}{t}\right)$ |

For instance the way of transforming the density function of wear rate g(u) according to Weibull distribution is the following [2,3] :

$$F(t) = \int_{z_g/t}^{\infty} g(u)du = \int_{z_g/t}^{\infty} \frac{k}{a}\left(\frac{u}{a}\right)^{k-1} \exp\left[-\left(\frac{u}{a}\right)^{k}\right] du = \tag{5}$$

$$= -\exp\left[-\frac{u}{a}\right]^{k} \Big|_{z_g/t}^{\infty} = \exp\left[-\left(\frac{z_g}{at}\right)^{k}\right]$$

Making use of the relationships :

$$f(t) = \frac{dF(t)}{dt} \tag{6}$$

and :

$$\lambda(t) = \frac{f(t)}{1-F(t)} \tag{7}$$

one obtained the expressions for the distribution density function of time to correct-to-limit-state operation, f(t), and the failure rate function λ(t), presented in Tab.1. This way the wear rate distributions were transformed into the distributions of time to correct-to-limit-state operation of elements. The parameters of the wear rate distributions and the allowable limit wear value of element, $z_g$ , become the parameters of the new distributions.

If the parameter W :

$$W = \frac{z_g}{\bar{u}} \tag{8}$$

is introduced and the expressions for the expected wear rate values $\bar{u}$ for particular distributions are assumed, the following forms of the function f(t) are obtained :

***for normal distribution of wear rate***,

where : $z_g = W \cdot \bar{u}$ (9)

$$f(t) = \frac{W \cdot \bar{u}}{\sqrt{2\pi}\,\sigma t^2} \exp\left[-\frac{\bar{u}}{2}\left(\frac{W-t}{\sigma t}\right)^2\right] \tag{10}$$

***for exponential distribution***,

where : $z_g = W \cdot \bar{u} \qquad \bar{u} = \lambda^{-1}$ (11)

$$f(t) = \frac{W}{t^2} \exp\left(-\frac{W}{t}\right) \tag{12}$$

***for Rayleigh's distribution***,

where : $\bar{u} = \sqrt{\frac{\pi c^2}{2}}$ (13)

$$f(t) = \frac{\pi}{2}\frac{W^2}{t^3} \exp\left[-\frac{\pi}{4}\left(\frac{W}{t}\right)^2\right] \tag{14}$$

***for Weibull's distribution***,

where : $\bar{u} = a \cdot \Gamma\left(\frac{1}{k}+1\right)$ (15)

$$f(t) = \frac{kW^k \Gamma^k\left(\frac{1}{k}+1\right)}{t^{k+1}} \exp\left\{-\left[\frac{W}{t}\Gamma\left(\frac{1}{k}+1\right)\right]^k\right\} \tag{16}$$

Because the Weibull's distribution is the most versatile, the equation (16) may be taken as the general form of f(t) which – for selected values of the parameter *k* – yields particular solutions.

Hence :

- for the parameter value k = 1 the equation (16) is transformed into the equation (12)
- for k=2 the equation (14) is obtained.

The equation (16) is the probability density function f(t) for the two-parameter (W, k) distribution of time of correct-to--limit-state operation , shortly called the W distribution [2].

In this case the reliability function R(t) which expresses the probability that a device will not reach its limit state, can be represented as follows :

$$R(t) = 1 - \int_0^t f(t)dt = 1 - \exp\left\{-\left[\frac{W\Gamma\left(\frac{1}{k}+1\right)}{t}\right]^k\right\} \tag{17}$$

The function R(t) may be used to determine, a.o., γ-percent resources of working time of devices, hence for scheduling their maintenance operations. A good conformity of the course of the function R(t) and results of statistical investigations on diesel engine cylinder liners is shown in [3]. Therefore, depending on information at one's disposal, service data or theoretical relationships can be used for assessing the process of running down the working time resources of device elements.

## ASSESSING THE PROCESS OF RUNNING DOWN THE WORKING TIME RESOURCES OF DEVICE ELEMENTS

Working time of a device, after which its working time resource is exhausted, can be determined on the basis of the working time resource specified by the device's producer or that determined with the use of statistical data obtained from service.

A course of running down the working time resource of any device element is exemplified in Fig.2, at accounting for the following assumptions [11] :

- the worn-out element (that of exhausted time resource) is replaced with a new one
- at the beginning of the service process or just after replacement of an element the ratio of the working time resource $t_j$ and the time resource $t_n$ (specified by the device's producer) equals 1, and it equals 0 when the working time resource is run out, i.e the limit state is attained
- the running-down process of working time resource of an element in service, is linear.

The diagram shown in Fig.2 presents the way of determining the time after which an element is to be replaced with a new one or subjected to major repair.

In the course of operation one does not usually let the limit state of device elements to occur, therefore their replacement is made in advance. In Fig.2 such procedure is represented by dotted lines. Technical state of an entirely used up (worn out) element of the device whose values of control parameters are contained within the interval between allowable and limit ones [2], is so low that further use of the device in which such element is installed is associated with greater and greater expenditures. Hence the in-advance replacement of an element with a new one as a matter of fact shortens its working time, however it also makes it possible to restore the demanded technical state of the device and to reduce its operational cost.

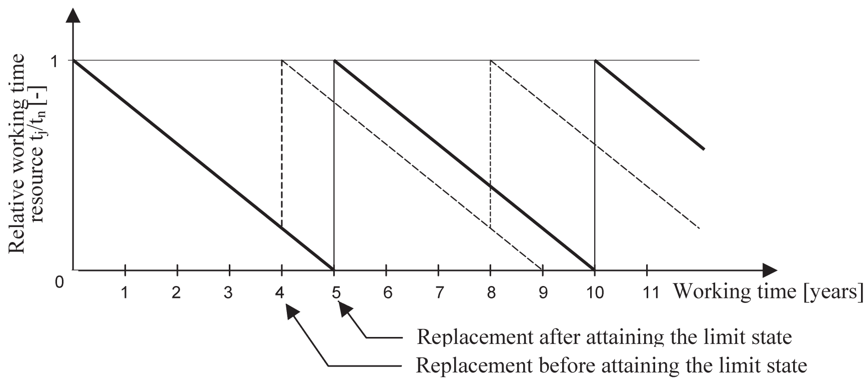



*Fig.2. Running-down process of working time resource of an object*

In the case of replacement of the entire device with a new one the system (e.g. power plant) in which it has been installed, becomes restored , however if only one of its elements is replaced then the device (or power plant) becomes restored partly. Therefore it can be assumed that in the first case the device's technical state and working time resource become restored to their maximum values, whereas if the object is subjected to overhaul its technical state and working time resource can be maintained at a given level or restored , however they will no longer reach their initial values (those at the beginning of the device's use) and the device in question will be no longer considered as „ a new one". It means that in the case of repaired objects the line of the maximum relative working time resource shown in Fig. 2 will monotonically descend. The entire problem in question is presented a.o. in [12].

By applying the way of the determining of working time resource, presented in Fig. 2, great advantages are obtained by means of analysis of running-down process of working time resources of elements of complex technical objects, e.g. diesel engine, in order to answer the question : **which the engine's elements require to be serviced, and after which time of operation (see Fig.3)**. Similar analysis can be performed for e.g. ship power plant. As a result of it specification of the objects which require maintenance operations in particular years of ship service would be obtained.

Various possible schedules of servicing can be assumed, e.g. after every voyage, each year etc; in each case the scope, time and cost of servicing will be different. It will also depend on service conditions of particular objects, which greatly influence working time resources of the objects.

For instance, Fig.4 presents the way of determining the resources of working time to preventive maintenance, routine repair and major repair, counted from a given service year $t_A$ , at different service conditions.

In the example shown in Fig.4 the object's serviceability time was divided into equal intervals in such a way that the

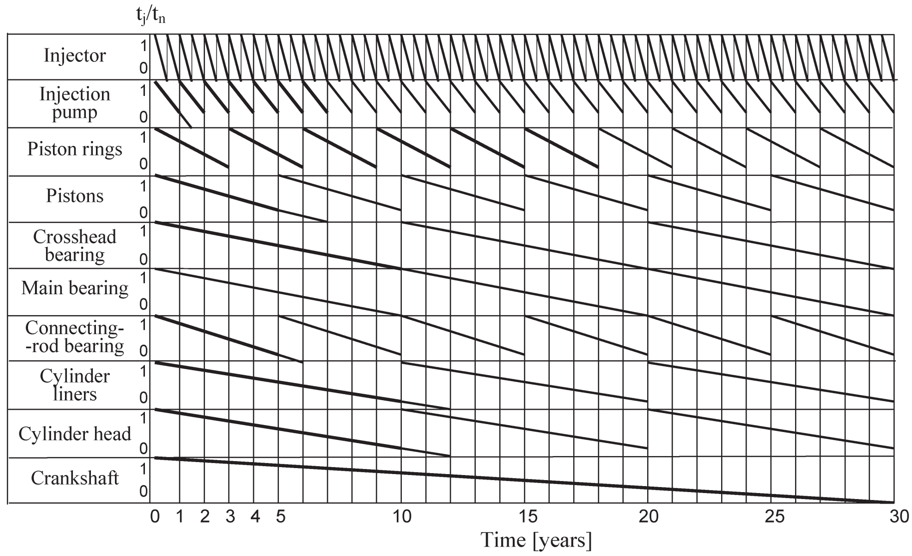
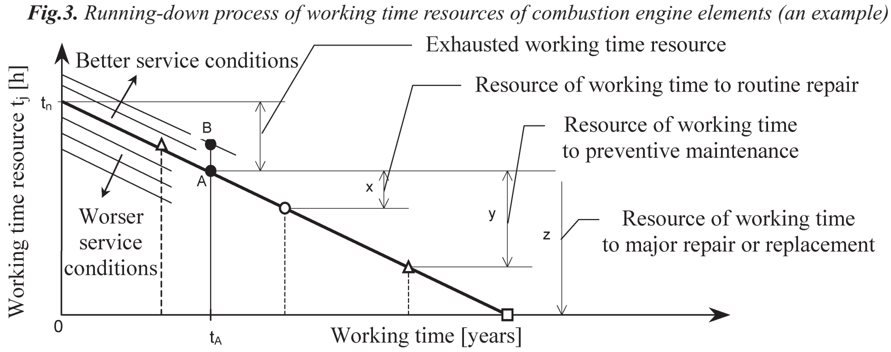



*Fig.3. Running-down process of working time resources of combustion engine elements (an example)*




Symbols : △ – preventive maintenance e.g. adjustment, cleaning, lubricating, etc.
○ – routine repair □ – major repair (in shipyard) or replacement

*Fig.4. Influence of service conditions on working time resource of an object and on scheduling its servicing operations*

time intervals to preventive maintenance, routine repair and major repair were equal (usually given by device producers or shipowner). Any change of service conditions would result in changing the working time resource of the device in question , manifested by shifting-up the line of running-down the working time resource of the device, shown in Fig.4.

For instance, an improvment of service conditions stands for the passing from the point A to the point B and for an appropriate increase of the distances x, y and z determined relative to a new line of running-down the working time resource. As a result it would be necessary to modify the initial schedule of servicing the device.

## FINAL REMARKS

- In the subject-matter literature there are no established procedures for the determining of working time resources of technical objects in changeable service conditions.
- The presented concept of the predicting of working time resources may facilitate the scheduling of preventive maintenance of objects and the choosing of optimum servicing plan.
- Usefulness of a given prediction method increases if a comprehensive physical analysis of development of wear processes and their features are performed in advance.

For all prediction realizations it is important to know history of development of the to-be-predicted process. Moreover, an analysis of the conditions which determined past course of the process, assessment of significance of structural components influencing its course, as well as estimation of possible influences in the future, is also important.

- Wear rate distributions of device elements can be used for determining distributions of their working time to limit state and in consequence to determine service life indices characterizing the devices e.g. their working time resources.
- From theoretical and practical point of view the presented distribution W can be useful in describing progressive failures.

### NOMENCLATURE

a – scale parameter of Weibull distribution
f(t) – probability density function of time to limit state (time to failure)
F(t) – failure probability function (unreliability function), cummulative distribution function of serviceability time
g(u) – wear rate probability density function
h(z) – wear-out probability density function
k – shape parameter of Weibull distribution
$n_o$ – number of objects in question
n(t) – number of objects failed to the instant *t*
R(t) – reliability function
t – time
$t_\gamma$ – time of correct operation of object at γ-percent probability (γ-percent time resource)
T – time to limit state (random variable)
u – wear rate (realization); $u = z/t$
$u_g$ – limit wear rate; $u_g = z_g/t$
$\overline{u}$ – expected wear rate value
U – wear rate (random variable); $U = Z/t$
$U_g$ – limit wear rate (random variable); $U_g = z_g/T$
W – expected service life value; $W = z_g/\overline{u}$
z – wear (realization); increment of control parameter value
$z_d$ – initial (lower) wear value
$z_g$ – limit (upper) wear value
Z – wear (random value)
λ – failure rate, exponential distribution parameter
λ(t) – failure rate function
σ – normal distribution parameter; standard deviation.

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# An analysis of characteristics of ship gas turbine propulsion system (in the light of the requirements for ship operation in the Baltic Sea)

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ABSTRACT

*The paper concerns a propulsion system of merchant ships intended for sailing in the Baltic Sea zone. Such system is to satisfy the ecological requirements determined by relevant international conventions for special zones to which the Baltic Sea also belongs. The paper draws attention to gas turbine used as a prime mover for such ships, because it satisfies the ecological requirements and has also other advantages. Application of gas turbine for ship powering does not require exhaust gas to be purified, however it requires fuel oils of a low sulphur content to be used. If the ecological rules impose the using of the fuel oils of similar quality for diesel engines then gas turbine propulsion system will be comparable – also economically (regarding specific fuel oil consumption cost) – with that of diesel engine. It would be even more favourable in a combine gas turbine /steam turbine system, especially at compound production of electric and heat energy (i.e. COGES systems). In the Baltic Sea zone gas turbines will find application to powering a. o. such ships as : fast car-passenger ferries, fast cargo ships, special vehicles (hydrofoils, hovercraft, motor yachts).*

**Keywords** : ship propulsion system, gas turbines, environment protection

## INTRODUCTION

In the world's merchant fleet diesel engines still have prevailed as the prime movers of ships. And, about 70% of ship main diesel engines are low-speed ones. The general reason is their high efficiency even at partial load, and additionally – – a low cost of fuel oil used for low-speed engines (so-called heavy oil).

Gas turbines do not substantially contribute either in total number or total output of ship main engines. For recent years neither number of gas turbines nor their total output has increased significantly [6]. It's worth adding, for instance, that the total output of the low-speed diesel engines adjusted for using heavy oil, ordered in 2002, dropped by 48% relative to the records of the precending year. This is an important circumstance from the point of view of development of gas turbine application as ship prime mover because mainly the above mentioned advantages of diesel engine over gas turbine are brought up in economical comparisons.

Low speed diesel engines working on heavy oil are first of all used to drive large ships. Such engines are disadvantageous for ecological reasons (large $SO_X$ pollution of the environment). However it were not those reasons which made the number of orders for them in 2002, lower. Rather general safety reasons and economical situation in different regions of the world belonged to them [6].

This paper is devoted to a propulsion system of merchant ship intended for sailing in the Baltic sea zone. Hence such system is to satisfy the ecological requirements contained in the relevant international conventions for special zones [1, 12, 14] to which the Baltic Sea belongs too. It draws attention to gas turbine as a main engine for such ship, because it satisfies the ecological requirements, moreover it has another advantages (discussed in detail below).

## ANALYSIS OF ECOLOGICAL REQUIREMENTS FOR PROPULSION SYSTEMS OF SHIPS OPERATING IN THE BALTIC SEA ZONE

In accordance with the provision [14] of the International Convention for the Prevention of Pollution from Ships (adopted in 1973, modified in 1978 and amended in 1997) [12] suphur content in marine fuel oils is not to exceed 4,5%. And, in the zones subject to control of sulphur dioxide emission (e.g. the Baltic Sea, ports, costal waters up to 200 NM from land, and other zones specified by IMO) one of the two following conditions is to be fulfilled :

- sulphur content in fuel oil is not to exceed 1,5%
- sulphur dioxide emission is not to exceed 6,0 g/kWh.

A reason for such control and limitation of the environmenal pollution by sulphur oxides is the protection of human health and the environment. Sulphur dioxide penetrates the environment as a product of combustion of natural fuels used for energy production purposes. 80% of natural fuels is combusted on land, and 20% on ships. Most of sulphur dioxide released from ships pollutes the marine environment. Sea pollution due to sulphur dioxide emitted from ships does not participate much in its total content in the marine environment if to compare the total amount of the sulphur contained in sea water (over $10^{15}$t) with that contained in natural fuel resources ($10^{11}$t). However despite that, in some zones of the marine environment, including the Baltic Sea, the amount of sulphur oxides emitted from ship propulsion systems is a threat to the environment.

Among other, the Baltic Sea environment is polluted to an extent greater than average. The reason of the susceptibility of the Baltic Sea is its rather small water depth and salinity.

Some reduction of the environmental pollution by sulphur oxides emitted from ships may be obtained in the following ways [4] :

- by lowering sulphur content in fuel
- by increasing propulsion efficiency
- by separating sulphur oxides from exhaust gases
- by applying new energy sources, other than those have been using so far.

However until now, application of an energy source other than fuel oil for ship propulsion has appeared either irrational (e.g. nuclear energy) for economical reasons, or impractical (e.g. wind).

A source of sulphur dioxide in exhaust gas is the sulphur contained in fuel oil. Substitution of a fuel oil of high sulphur content for that of lower one or natural gas would result in a lower sulphur dioxide content in exhaust gas.

However in this case some limitations associated with availability and cost of a fuel oil of low sulphur content, could appear. In present, the sulphur weight content in marine fuel oil amounts to 2.8%÷3.0%. The situtation has not changed for the last ten years. The reducing of the sulphur dioxide content in exhaust gas by 50% , postulated by IMO, would require to lower half as much the present sulphur content in fuel oil. Cost of such fuel oil would reach even 50 USD/t.

For the zones of a more limited sulphur dioxide content in exhaust gas, such as the Baltic Sea, the limitation of the sulphur content in fuel oil down to 1% is considered. Cost of such fuel oil would increase by 58÷85 USD/t.

Due to efforts resulting from competition between ship engine producers diesel engine efficiency of both two-stroke and four-stroke engines has reached a high value (i.e. a low value of specific fuel oil consumption). To attain any higher efficiency would be very difficult. Hence the increasing of engine efficiency has a small influence on the reducing of sulphur dioxide content in exhaust gas.

In marine conditions the washing purification method of exhaust gas from sulphur dioxide by making use of absorbing features of sea water (respective to sulphur oxides) is considered. However this is a sophisticated and expensive system. Moreover, in such case fuel oil consumption increases and – in consequence – also the environmental pollution due to carbon dioxide. Moreover, its effectiveness depends on salt content in sea water, which is low in the case of the Baltic Sea.

In association with the above given comments it seems that some lowering of sulphur content in marine fuel oil may be expected. Trends to lower sulphur dioxide content in exhaust gas emitted from ships would probably lead to elimination of heavy oil application for ship propulsion systems. This seems unavoidable especially in such zones as the Baltic Sea. This way a higher share of diesel oil in total fuel oil consumption for ship propulsion would cause that the gas turbine would become competitive against the high-pressure combustion piston engine, also from the point of view of specific fuel oil consumption cost, especially if applied in a compound gas-steam system (see Tab.1).

The presented (Tab.1) specific fuel consumption of a gas turbine working in a compound gas-steam system is a little higher than that of a diesel engine. However as distinct from the gas turbine, the diesel engine consumes a rather large amount of lubricating oil, which results in that the specific power cost of both kinds of propulsion becomes comparable.

***Tab.1.*** *Sulphur content in marine fuel oil*

| Kind of ship propulsion system | Fuel | Sulphur content [%] | Specific fuel consumption [kg/MWh] | Sulphur emission [g/MWh] |
|---|---|---|---|---|
| Low-speed diesel engine | liquid : | | | |
| | heavy oil | 4 | 170 | 68 |
| | diesel oil | 1.5 | 170 | 26 |
| Gas turbine | liquid : | | | |
| | diesel oil | 1.5 | 240 | 36 |
| | natural gas | 0 | 240 | 0 |
| Compound gas-steam system | liquid : | | | |
| | diesel oil | 1.5 | 180 | 27 |
| | natural gas | 0 | 180 | 0 |

## COMPARISON OF TWO SHIP PROPULSION SYSTEMS : BASED ON DIESEL ENGINE AND GAS TURBINE, RESPECTIVE TO TECHNICAL, ECONOMICAL AND ECOLOGICAL ASPECTS

Gas turbine used as a ship main engine has many advantageous features, namely :

- low specific initial cost
- small dimensions
- low weight
- modular structure
- low level of vibrations and noise
- long time between successive overhauls.

Despite that its share as a ship prime mover in total output and number of applied engines on ships is still low, because gas turbine has some significant drawbacks in comparison with high-pressure combustion engine; these are :

- low efficiency when working in a simple propulsion system
- fast dropping efficiency at decreasing load
- it requires light fuel oil to be applied, which is more expensive than heavy oil used for low-speed diesel engines (covering more than 70% of total power used for ship propulsion).

For this reason the intensive research ordered by US Navy is carried out , aimed at development of a new type of marine gas turbine working on low-cost fuel oil [26, 27, 28].

The gas turbine applied on ships are mainly of an aircraft type (about 95%). Their output is in the order of 3000÷27000 kW, initial gas temperature : 850÷1100°C, compression ratio: 10÷26, efficiency: 27÷40%, specific fuel oil consumption: 220÷260 g/kWh [13]. Most of them is delivered by General Electric, among which LM 2500 turbine is most popular.

The investigations in question are aimed first of all at increasing initial temperature of combusted gas in order to make combusting a lower quality fuel oil possible. Moreover, waste heat recovery from exhaust gas is also planned to lower fuel oil consumption down to 170÷180 g/kWh. The application of exhaust gas recuperation at outlet from turbine, and of interstage compressor cooling makes it possible to decrease specific fuel oil consumption by about 30%. The investigations are also aimed at obtaining a lower fuel oil consumption at partial load levels of gas turbine.

TheRolls-Royce, a British firm in cooperation with the Westinghouse, a US firm has built WR-21 ship gas turbine of a new type in which the interstage cooling and recuperation system (ICR) have been applied. It is a triple-shaft turbine of 25000 kW output and a modular structure.

General Electric has modernized its known LM2500 turbine (now marked : LM2500R) by adding an interstage cooling and recuperation system [26,28].

At outlet either from a turbine or a diesel engine a steam generator supplying a steam turbine can be installed to utilize exhaust gas heat. In the case of gas turbine, exhaust gas transfers to environment more heat than in the case of diesel engine in which a significant part of the heat is absorbed by the water cooling the engine . Therefore in a combined gas turbine/steam turbine system, the steam turbine's output is much greater than that in a combined diesel engine/steam turbine system.

For ship propulsion two kinds of combined gas/steam systems are used :

- in the COGAS system (COmbined Gas And Steam turbine) ship's screw propeller is driven through a mechanical transmission gear
- in the COGES system (COmbined Gas and Steam turbine Electric) the so-called electrical transmission is used. The turbines drive electric generators which supply a common electrical power system (electrical network), and ship screw propeller(s) is (are) driven by electrical motors.

The Deltamarin, a Finnish advising firm has compared the COGES propulsion systems and diesel-electric propulsion systems (fitted with an electric transmission) installed on different merchant ships. On this basis it has specified the following advantages of the COGES propulsion system [17] :

- low initial cost
- high thermodynamical efficency: specific fuel oil consumption equal to or lower than that of a diesel engine of a comparable output (working through an electrical transmission)
- low operational cost (servicing, maintenance, lubricating oil)
- low sulphur content in exhaust gas (exhaust gas purification devices are not necessary)
- low level of vibrations and noise (e.g. passenger cabins can be located in the vicinity of engine room)
- high availability (high reliability, and possible fast operation of repair or replacement of engine)
- much smaller number of auxiliary devices, which makes initial and operational costs lower
- small engine room dimensions resulting in a greater space for cargo.

## *Technical characteristics*

Out of the technical factors influencing choice of a kind of ship propulsion system the following are considered :

- **operational reliability**
  The gas turbines show low failure frequency and high availability greater than that of the diesel engines and steam turbines. Reliability of gas turbines used as ship prime movers amounts to about 98% [9]; the result is based on multi-year service experience on aircraft, ship and industrial turbine engines.
- **overhaul labour consumption**
  Due to a modular structure of gas turbine, access to its particular units and replacement of failed parts is very fast. It is possible to overhaul the engine without taking the ship out of operation. Gas turbine engine may be replaced within less than 24 h. Its small weight and gabarites make it possible to handle the whole engine at once, if necessary.
- **power transmission**
  In the case of fully mechanical powering, the gas turbine propulsion system requires a mechanical transmission gear to be applied. It is also required in the case of using high- and medium-speed diesel engines as ship prime movers, however low-speed diesel engines do not require it.
  As applications of the electrical transmission are still growing the way of power transmission from both gas turbine and diesel engine becomes analogous.
- **manoeuvrability**
  Diesel engines are reversible; gas turbines are irreversible as a rule. The stage of controllable guide vanes which makes reversible action of gas turbine possible, is rarely applied. The disadvantage of gas turbine can be overcome by using reversible transmission gears or controllable pitch propellers. Small gabarites of gas turbine rotor result in its very small inertia. Due to this, manoeuvrability of gas turbine ships is very high. Starting operation of gas turbine is very fast; starting from cold state, it is capable of developing full output within several minutes.
- **automation of operation**
  In comparison with diesel engines, gas turbines can be automatically controlled in a more easy way. It leads to possible reduction of a number of operators.
- **vibrations, noise**
  Vibrations generated by gas turbine in operation are much smaller than those due to diesel engine. They are easier damped due to their higher frequencies than those generated by diesel engines. Noise intensity due to gas turbine operation is much lower than that generated by diesel engine. Moreover, gas turbines are usually covered by an acoustic insulation.

## *Economical characteristics*

Economical characteristics of the considered kinds of ship propusion system were compared by using a selected ship as an example.

It was the fast Ro2000 ship of 10.000 DWT load-carrying capacity, built by German Flender shipyard [2,3]. Its two controllable pitch propellers driven by two medium-speed, four-stroke diesel engines of 50 MW total output , propell the ship up to 28 kn speed. The ship's propulsion system was compared with a system of the same power but driven by a gas turbine, as well as with the COGES, a combined gas/steam system.

As far as the diesel engine propulsion system is concerned the following information was assumed valid :

- Two MAN-B&W 12V 48/60 medium-speed diesel engines, each of 12600 kW output at 600 rpm, were assumed to be the ship's prime mover
- Electrical power was produced by four generating sets of 1000 kW output each, driven by four HFC5 632/14K-T260L-EX Yanmar diesel engines
- It was assumed that four waste heat boilers delivered each of 1000 t/h of steam
- The amounts of consumed electric and heat energy were estimated on the basis of available data dealing with a ro-ro ship of a similar load-carrying capacity.

As far as the gas turbine propulsion system is concerned it was assumed that :

- the system consisted of two WR-21Northrop Grunman / Rollce-Royce gas turbines of 25000 kW output each, driving two ship screw propellers
- the turbines are equipped with an interstage cooling and recuperation systems due to which the turbine's efficiency amounts to 40%
- Exhaust gas from the recuperator supplies a waste heat boiler.

✦ Specific fuel oil consumption of the turbines not much varies within 50÷100% interval of the rated output.

As far as the COGES system is concerned it was assumed that two LM2500 General Electric gas turbines of 22400 kW output each, as well as a steam turbine of 10000 kW output, was applied.

In order to compare the above specified propulsion systems the following measures were used :

*propulsion efficiency :*

$$\eta = \frac{P_e \cdot \eta_g \cdot 3600}{B_h \cdot W_d}$$

*energy efficiency :*

$$\eta' = \frac{(P_e \cdot \eta_g + P_{el}) \cdot 3600 + Q}{B_h \cdot W_d}$$

*specific fuel oil consumption :*

$$b = \frac{B_h}{P_e \cdot \eta_g}$$

where :

$B_h$ - total fuel oil consumption per hour (by main engine, auxiliary engines and boilers)
$P_e$ - main engine output
$P_{el}$ - generated electric power
$Q$ - hourly flow rate of produced heat
$W_d$ - net calorific value of fuel oil
$\eta_g$ - transmission gear efficiency.

Appropriate values of the economical characteristics of the ship propulsion systems in question are compared in Tab.2.

***Tab. 2.** Comparison of the economical characteristics of the considered ship propulsion systems*

| Characteristic elements | MAN-B&W 12V 48/60 diesel engine | WR-21 gas turbine | COGES system |
|---|---|---|---|
| propulsion efficency, [%] | 43.6 | 40 | 46.8 |
| specific diesel fuel oil consumption, [g/kWh] | 194 | 214 | 183 |
| energy efficency, [%] | 49.7 | 45.6 | 49.1 |
| specific lubricating oil consumption, [g/kWh] | 1.0 | ~ 0 | ~ 0 |
| price of marine diesel oil, [USD/t] | ~ 140 | ~ 140 | ~ 140 |
| price of lubricating oil, [USD/t] | ~ 2000 | ~ 2000 | ~ 2000 |
| specific cost of marine diesel oil, [USCent/kWh] | 2.72 | 3.00 | 2.56 |
| specific cost of lubricating oil, [USCent/kWh] | 0.2 | ~ 0 | ~ 0 |
| price of heavy oil, [USD/t] | ~ 91 | – | – |
| specific cost of heavy oil, [USCent/kWh] | 1.64 | – | – |

It was assumed that both the diesel engine and gas turbine is supplied with the same kind of fuel oil (marine diesel oil). In such conditions the specific fuel oil consumption of the COGES system appears the smallest, somewhat smaller than that of the medium-speed diesel engine in question. However, the diesel engine also consumes lubricating oil (much more than the combined system COGES), which makes that the specific power cost of the COGES system amounts to about 0.88 of that of the diesel engine propulsion system (see Tab.2).

The specific power cost of the gas turbine propulsion system (of a simple configuration) is only a little greater and it amounts to about 1.03 of that of the diesel engine propulsion system.

The specific cost of heavy oil is given in the last row of Tab.2. On its basis it can be stated that the specific power cost of the COGES system amounts to about 1.4 of that of the low--speed diesel engine propulsion system.

As a matter of fact at partial load diesel engine's efficiency decreases much slower than that of gas turbine. At the half--load, diesel engine's efficiency drops e.g. by 5% relative to its value at the rated load, whereas the efficiency of LM2500 gas turbine drops by 10%. However, in the COGES system 50% of the rated load can be obtained by keeping in operation at its rated load only one of the two gas turbines.

## *Ecological characteristics*

Exhaust gas from diesel engines contains more not only sulphur oxides but also nitrogen oxides due to a greater effectiveness of fuel combustion processes, in consequence – a greater temperature. The exhaust gas also contains more carbon oxide and hydrocarbons. Tab.3 enables to compare in this respect the considered ship propulsion systems.

***Tab. 3.** Comparison of the ecological characteristics of the considered ship propulsion systems*

| Kind of propulsion system | Kind of fuel | Sulphur content (weight) [%] | Sulphur emission [g/MWh] | $NO_x$ emission [g/kWh] | CO emission [ppm] | HC emission [ppm] |
|---|---|---|---|---|---|---|
| diesel engine | heavy oil<br>fuel oil | 4<br>1.5 | 68<br>26 | 11÷16 (medium -speed engine) | 60 | 180 |
| gas turbine – simple configuration | fuel oil<br>natural gas | 1.5<br>0 | 36<br>0 | 3÷5 | 25 | 5 |
| gas turbine – COGES system | fuel oil<br>natural gas | 1.5<br>0 | 27<br>0 | ~ 0 | 25 | 5 |

The necessity of complying with the environmental protection rules entails undertakings dealing with construction of the diesel engine itself, as well as with the exhaust gas purification devices [31]. Such undertakings are costly. However the gas turbine propulsion system can satisfy the environmental protection rules without any additional devices.

## COMPARISON OF SHIP POWER PLANT CHARACTERISTICS RESPECTIVE TO A KIND OF SHIP PROPULSION SYSTEM

A kind of ship propulsion system (a kind of prime mover) influences such characteristics of the ship power plant as : its weight, volume of occupied space , number of operators. They play an important role from the economical point of view. By decreasing the volume of its space it is possible to increase amount of cargo or number of passangers to be shipped. The lower the weight of the power plant, the smaller the power demanded for ship propulsion (the smaller fuel consumption). A smaller number of operators leads to a lower personnel cost.

Comparison of the ship power plant respective to its weight and volume with taking into account a kind of its propulsion system, can be performed by using two indices : the so-called specific mass and power density (concentration). The specific mass is defined as the ratio of the total power plant mass (of main engine, auxiliary devices, and empty piping systems) and the engine's rated output. The power density is defined as the ratio of the engine's rated output and the engine room volume. Values of the indices, presented in Tab.4, make it possible to compare the example power plant with the respect to the three considered kinds of propulsion system.

***Tab. 4.*** *Specific mass and power density of the example ship power plant in three considered versions of propulsion system*

| Index | Considered ship propulsion system based on : | | |
|---|---|---|---|
| | 12V 48/60 diesel engine | WR-21 gas turbine | COGES system |
| Specific mass [kg/kW] | 14.52 | 2.53 | 4.56 |
| Power density (concentration) [kW/m$^3$] | 48 | 247 | 780 |

Out of the considered kinds of ship propulsion system, the diesel engine power plant is of the greatest specific mass, about six times greater than the gas turbine one, and over three times greater than that of COGES type. The power density (concentration) of the gas turbine power plant is over five times greater than that of the diesel engine power plant; the greatest power density has the COGES version of the power plant - over sixteen times greater than that of the diesel engine power plant, and over three times greater than that of the gas turbine power plant (see Tab.4.).

As it has been already mentioned , the gas turbine power plants (including their COGES versions) are suitable for automation and control. This way, decreasing the number of power plant operators and simultaneous decreasing volume and mass of power plant as well as lowering its operational cost, can be possible.

Growing interest for application of electrical drive to ship propellers (by using the electrical transmission) also directs greater and greater attention to gas turbines. In this respect it is worth adding that in such version of propulsion system gas turbines can be located on the deck, which eliminates a troublesome arrangement of compressor's air supply pipes delivering air from the deck to the power plant.

## SPECIFIC ASPECTS OF GAS TURBINE APPLICATION ON SELECTED SHIPS

Choice of a main engine location depends on a type of ship and applied engine, and a power transmission way from the engine to propeller. For instance application of an electrical drive to ship propeller provides the designer with a great freedom in locating the main engine relative to a mechanical transmission of power. Any propelling engine due to its mass, thrust, torque, transient state loads and sea state must have a suitable foundation firmly connected with ship's hull structure. Such propelling engine as gas turbine is a source of high temperature and noise, hence protective means should be applied to lower noise and high temperature. The features have an influence on location of main engine within engine room as well as on construction of its foundation. In ship power plants based on gas turbines the main engine is usually installed in an individual casing seperated from ship's hull structure. The gas turbine module is connected to air intake and exhaust gas systems. Such solution of the engine's casing (container-like) enhances also fire-safety of the whole power plant.

Location of engine's seating depends on many aspects. One of them is troublesome centering the engine and propeller shafts. To avoid such operation a non-mechanical way of power transmission from the engine to propeller, usually an electrical transmission system, is applied. In this case different variants of the engine's location within ship's engine room are possible. The main engine can be located either aft or fore; if just aft then it is possible to make the shaft between the engine and propeller as short as possible , which reduces engine room's volume. The electrical transmission makes it possible to locate the main engine fore, without applying any long shaft. In most cases ship's power plant is usually located within ship hull's space.

Along with application of gas turbines new possibilities to locate the main engine even on the ship's deck, have appeared. Small mass of gas turbine provides the designer with a greater freedom in locating the power plant. For instance it is allowed to locate ship power plant above the main deck both on large and small ships, also in the case of application of heavy industrial gas turbines.

In Fig.1 is presented a variant of location of a power plant equipped with a heavy gas turbine installed on the main deck of a tanker. In the power plant a gas turbine with heat recuperation system, as well as electrical drive of propeller, are applied. Mass of propeller, shaft, electrical motor and gas turbine determines mass centre location below the main deck level (engine - generator - electrical motor - turbine shaft).

Dimensions of air and gas ducts of gas turbine influence location of the gas turbine itself. If the gas turbine is located high then lengths of inlet and outlet pipes are shorter, hence mass and dimensions of air and gas ducts are also changed.

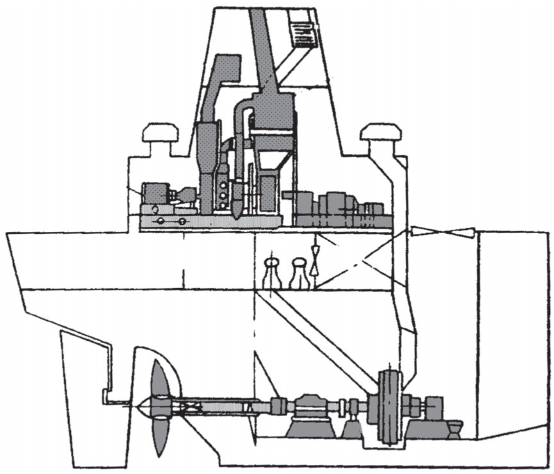

***Fig.1.*** *An example arrangement of tanker's power plant fitted with gas turbine and electrical drive of propeller*

In Fig.2 a gas turbine power plant installed on a container--ship is presented. Location of the gas turbine with compressor aft makes it possible to shorten inlet piping as much as possible.In such location of the gas turbine the reduction gear is placed before the engine, and the propeller shaft is led aft below the engine. This way the length of the power plant appears shorter in comparison with those of other types of heat engines.

Application of gas turbines to ship propulsion makes free access to all devices possible, both during service and overhauls. Moreover, lengths of air and gas ducts can be minimized. The number of devices associated with auxiliary mechanisms is reduced, and the engines themselves do not require to be additionally cooled with sea water (hence sea water pumps, their piping systems etc are not necessary).

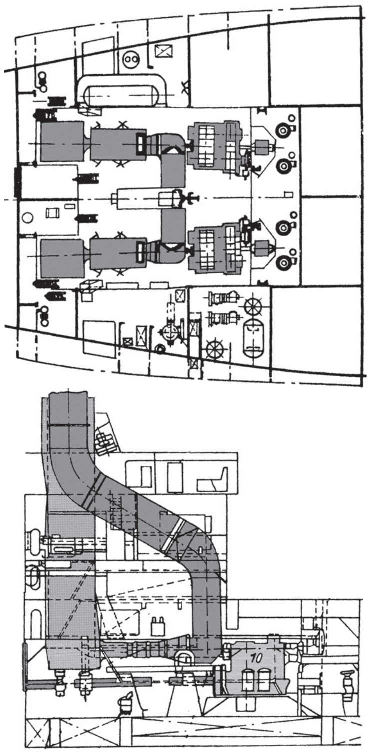

*Fig. 2.* An example solution of gas turbine power plant of a containership

## SUMMARY

- Application of gas turbines on merchant ships is still limited [6] due to their high operational cost (specific fuel oil consumption cost) especially in comparison with that of low-speed diesel engines working on heavy oil. However the ecological requirements established by the International Maritime Organization (IMO), limiting content of sulphur oxides in exhaust gas from ship propulsion engines [1,12,14], can make it possible – or even force – a broader application of gas turbines as ship prime movers. For special zones to which the Baltic Sea belongs too, the requirements are more stringent.
- Fulfilment of the requirements by diesel engines could be achieved by :
  - using a fuel oil of an appropriately low sulphur content (it would mean to stop using heavy oil)
  - installing exhaust gas purifying devices.

  Both the ways of reducing content of sulphur oxides in exhaust gas are expensive.
- Application of gas turbine to ship propulsion does not require exhaust gas to be purified, however it requires a fuel of low sulphur content to be used. In the case if the ecological requirements force application of fuel oil of a similar quality for diesel engines , then the gas turbine propulsion system will be comparable with the diesel engine one also respective to economical aspects (specific fuel oil consumption cost). In this respect it would be even more favourable if used in a combined gas turbine /steam turbine system, especially at common production of electrical and heat energy (COGES system).
- Gas turbine propulsion system has many technical advantages which make gas turbine superior , in this respect, over diesel engine. Diesel engines, especially low-speed ones working on heavy oil, are superior over gas turbines only in the respect of specific fuel oil consumption cost. This is why their share in the total propulsion power installed in merchant ships amounts to about 70%.
- In the Baltic Sea zone the gas turbines are expected to be more and more used for propulsion systems of such ships as :
  - fast car-passenger ferries
  - fast cargo ships
  - special vehicles (hydrofoils, hovercraft, motor yachts).

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# A proposal of ship gas-turbine driven waterjet propulsion – preliminary considerations

**Wiesław Próchnicki**, Assoc.Prof.,D.Sc.

ABSTRACT

*In the paper are presented preliminary considerations concerning the efficiency of waterjet ship propulsion system, as well as the calculation of main dimensions of waterjet channel.The friction and momentum losses of the flow channel have been roughly estimated by using Fliegner's equations. An important conclusion is confirmed that the summary losses are inversely proportional to square of ship velocity (~$1/u^2$). On the other hand the ship propulsion power is directly proportional to third power of ship velocity (~$u^3$). Therefore to minimize ship's hull resistance, hulls of waterjet-driven ships ought to be of a great slenderness – e.g. L/B = 15, stabilized by sponsons, or of semi-swath hydrofoil-supported construction.*

**Keywords** : ship propulsion, hydromechanics, waterjet

## INTRODUCTION

Attempts to apply waterjet propulsion to ships have already appeared in the ancient ages[a)]. However the Hamilton's low--power waterjets (1954) intended for propulsion of fast river boats can be considered as the first developed construction[b)]. Nowadays many firms build waterjets of a wide range output : from a few hundred kW to a few dozen MW.

Rolls-Royce, a leader of modern waterjet constructions, offers complete propulsion systems for very fast ferry ships, cargo vessels, warships, cruise ships and yachts.

Up to now about 1400 assemblies of the output from 220 to 70 800 kW have been built. And, a ship propulsion system of 250 MW output is under development

An example of Rolls-Royce waterjet propulsion system for a passenger ferryship is shown in Fig.1.

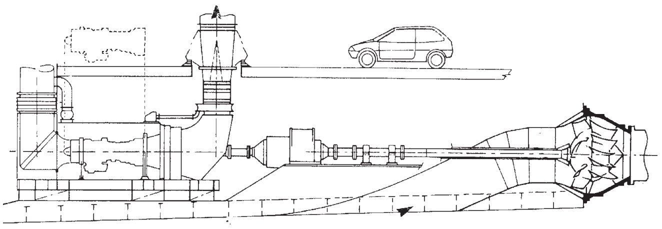

***Fig. 1.*** *ROLLS-ROYCE waterjet propulsion system for a passenger ferryship*

## CONSTRUCTIONAL ELEMENTS OF WATERJET PROPULSOR

In Fig.2 the main functional elements of waterjet propulsor are diagramatically presented.

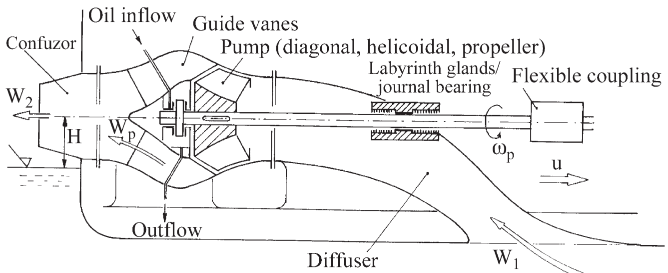


***Fig. 2.*** *Main elements of waterjet propulsor.*
***Notation*** *: $w_p$ – angular velocity of pump; $w_1$, $w_2$, $w_p$ – inflow, outflow, and pump flow velocity, respectively; u – ship speed; H – water elevation*

The flow channel consists of a diffuser, rotordynamic pump[c)], guide vanes (stationary) and confusor. The channel surface should be of high smoothness (low coefficient of relative roughness of surface).

The ship manoeuvring devices are shown diagramatically in Fig.3.

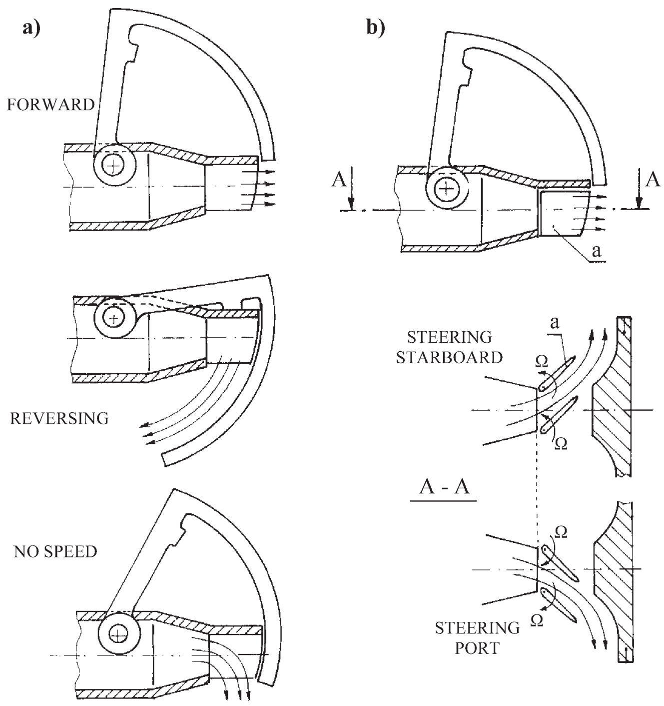


***Fig. 3.*** *Schematic diagram of ship manoeuvring elements:*
***a)*** *for ship reversing;* ***b)*** *for ship turning*
***Notation*** *: a - steering blades, Ω - blade turn angle*

The effectiveness of ship steering is very high because the waterjet force $F_u$ for $u \to 0$ is $F_u(o) \to F_{u,\max}$, and the manoeuvring operations are usually executed at low ship's speed : $u \to o$.

## ANALYSIS OF WATERJET PROPULSION EFFICIENCY

By taking into account that the relation :

$$w_1 \cong u \tag{1}$$

is satisfied the absolute values of water velocities at channel inflow and outflow, are :

$$\left.\begin{aligned} c_1 &= w_1 - u \cong 0 \\ c_2 &= w_2 - u \end{aligned}\right\} \tag{2}$$

The force of water stream reaction, is :

$$F_u = m(c_2 - c_1) \cong m(w_2 - u) \tag{3}$$

where : m [kg/s] – rate of water outflow.

For u = o

$$F_u(o) = F_{u,max} \cong m \cdot w_2 \tag{4}$$

Therefore the ship propulsion power is :

$$N_u = F_u \cdot u = m(w_2 - u) \cdot u \tag{5}$$

and carry – over loss output :

$$N_c = m\frac{c_2^2}{2} = m\frac{(w_2 - u)^2}{2} \tag{6}$$

Applying the notation :

$N_p$ – pump propulsion power
$\eta_p$ – overall pump efficiency
$N_L$ – power of summary flow channel losses

one obtains :

$$N_p \cdot \eta_p = N_u + N_c + N_L \tag{7}$$

The propulsion efficiency $\eta_{N_u}$ is defined as follows :

$$\eta_{N_u} \stackrel{df}{=} \frac{N_u}{N_p} \tag{8}$$

hence :

$$\eta_{N_u} = \frac{N_u \cdot \eta_p}{N_u + N_c + N_L} \tag{9}$$

If it is assumed that the efficiency $\eta_p = 1$, then :

$$\eta_{N_u,\eta_p=1} = \eta^o_{N_u} = \frac{N_u}{N_u + N_c + N_L} \tag{10}$$

Inserting (5) and (6) in (10) one obtains

$$\eta^o_{N_u} = \frac{2(1-\bar{u})\cdot\bar{u}}{1-\bar{u}^2 + \frac{2N_L}{m\cdot w_2^2}} = \frac{2(1-\bar{u})\cdot\bar{u}}{1-\bar{u}^2\left(1-2\frac{N_L}{m\cdot u^2}\right)} \tag{11}$$

where : $\bar{u} = \frac{u}{w_2}$ – velocity index.

The expression of relative (nondimensional) losses power :

$$2\frac{N_L}{m\cdot u^2}$$

which may be represented as a sum of *n* particular losses of the waterjet flow in channel, yields the following [5] :

$$2\frac{N_L}{m\cdot u^2} = \sum_{i=1}^{n}\zeta_i \tag{12}$$

Fig.4 shows the diagram of efficiency $\eta^o_{N_u}$ in function of the velocity index :

$$\bar{u} = \frac{u}{w_2}$$

and of the summary losses $\sum\zeta_i$.

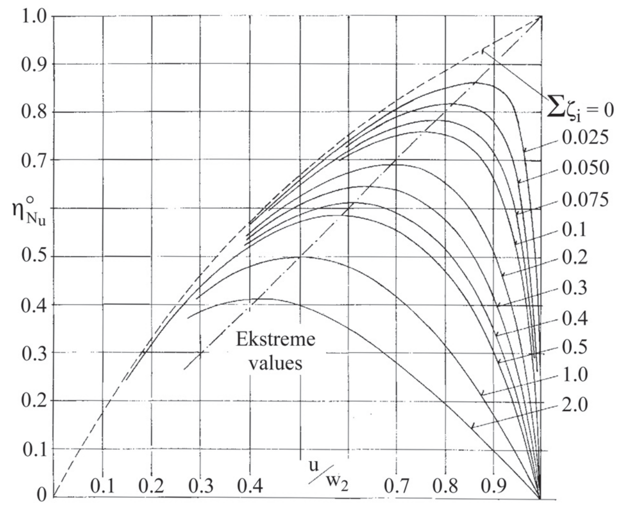


***Fig. 4.*** *Propulsion efficiency $\eta^o_{N_u}$ (for $h_p = 1$) as a function of $\bar{u} = \frac{u}{w_2}$ and $\Sigma\zeta_i$, see [3]*

The maximum (optimum) efficiency is the value :

$$\eta^o_{N_u,max} = \eta^o_{N_u}(\bar{u}_{opt})$$

where :

$$\bar{u}_{opt} = \left(1 + \sqrt{\sum_{i=1}^{n}\zeta_i}\right)^{-1} \equiv \eta^o_{N_u,max} \tag{13}$$

In Fig.5 the theoretically determined summary losses (of friction and momentum) are shown :

$$2\frac{N_L}{m\cdot u^2} = \sum_{i=1}^{n}\zeta_i$$

as a function of :

$$\sum_{i=1}^{n}\zeta_i = f(u, H, \bar{u}, \lambda_f) \quad [5] \tag{14}$$

where :

$\lambda_f(Re)$ – the friction number (dependent on the Reynolds number Re, and surface roughness coefficient of the flow channel), are shown for $\bar{u} = 0.8$ and $Re=10^7$.

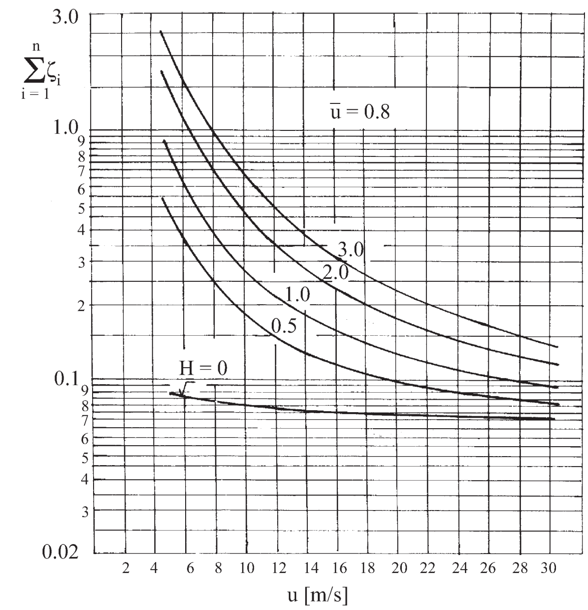


***Fig. 5.*** *Coefficient of summary losses $\left(\sum_{i=1}^{n}\zeta_i\right)$ for $\bar{u} = 0.8$ and $Re=10^7$*

In Fig.6 the efficiency $\eta^{o}_{N_u,max}$ is presented in function of the summary losses $\left(\sum_{i=1}^{n}\zeta_i\right)$

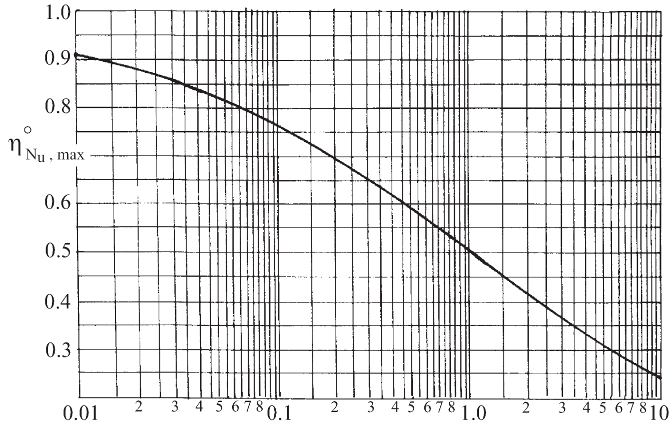


*Fig. 6. The optimum efficiency $\eta^{o}_{N_u,max}$ in function of $\left(\sum_{i=1}^{n}\zeta_i\right)$*

## MAIN DIMENSIONS OF WATERJET PROPULSION SYSTEM

The power of ship propulsion is :

$$N_u = m\cdot(w_2 - u)\cdot u = 5402.16\cdot\alpha\cdot u^3 \qquad (15)$$

where :

$$\alpha = \frac{\sqrt[3]{D_w^2}}{C_o}\left[\frac{HP}{kt.^3}\right] \text{ - power coefficient}$$

$C_o$ - Admiralty coefficient
$D_W$ [t] = $\rho_W\cdot V_Z$ - ship mass
$\rho_W$ = 1.02÷1.03 [t/m$^3$] - water density
$V_Z$ [m$^3$] - ship displacement
u [m/s] - ship speed.

Inserting the equation of waterjet flow rate :

$$m = 2\cdot A_2\cdot w_2\cdot\rho_w{}^{d)} \qquad (16)$$

into (15) one obtains :

$$5.3\frac{\alpha}{2A_2}\left(\frac{u}{w_2}\right)^2 + \frac{u}{w_2} - 1 = 0$$

hence :

$$A_2 = 2.65\alpha\frac{\bar{u}^2}{1-\bar{u}} \qquad (17)$$

The graph of the waterjet outflow area of one of both nozzles $A_2 = A_2(\bar{u},\alpha)$ is shown in Fig.7.

Dimensional proportions of the outflow nozzles are shown in Fig.8 [6].

## AN EXAMPLE ARRANGEMENT OF THE SHIP WATERJET PROPULSION SYSTEM WITH GAS-TURBINE DRIVE

The considered waterjet system is intended for a passenger ship of the following parameters :

ship length L = 80 m, L/B = 6.3 , ship mass $D_W$ = 4000 t, speed u = 6 m/s (11.7 kt.), Admiralty coefficient $C_o$ = 300, power output $N_u \cong$ 980 kW, having main engine : one-cylinder gas turbine with heat exchange regeneration (air preheater), of effective efficiency :

$$\eta_e^{GT} = 0.33 \div 0.34$$

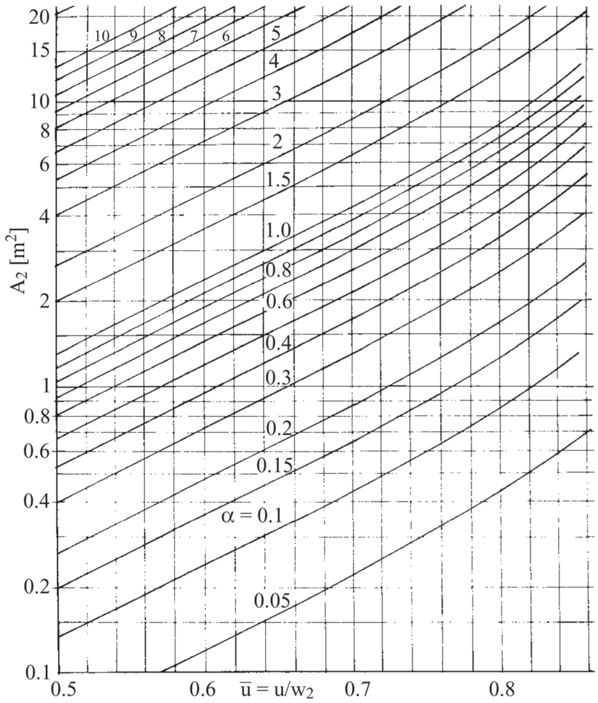


*Fig.7. Waterjet outflow area of one of both nozzles $A_2$ in function of the power coefficient $\alpha$ and velocity index $\bar{u} = \frac{u}{w_2}$*

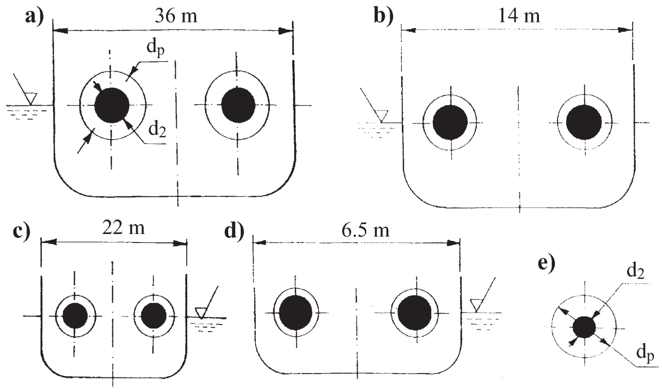


*Fig.8. The stern area of various types of ships (maintaining the dimensional proportions): **a)** – passenger liner of L = 280 m, u = 12 m/s, $\alpha$ = 5.01 HP/kt.$^3$; **b)** – passenger vessel of L = 88 m, u = 6 m/s, $\alpha$ = 0.907 HP/kt.$^3$; **c)** – passenger ship of L = 150 m, u = 9 m/s, $\alpha$ = 2.23 HP/kt.$^3$; **d)** – tug of L = 40 m, u = 5.6 m/s, $\alpha$ = 0.207 HP/kt.$^3$; **e)** – racing boat of u = 25 m/s*

Assuming the velocity index $\bar{u} = \frac{u}{w_2} = 0.75$ and taking into account the power coefficient

$$\alpha = \frac{1}{C_o}\sqrt[3]{D_w^2} \cong 0.84$$

one gets from the diagram in Fig.7 : $A_2 \cong 5$ m$^2$ hence, the outflow diameter is: $d_2 \cong 1.8$ m.

If H = 0 , $\eta_p$ = 0.9 and $\left(\sum_{i=1}^{n}\zeta_i\right) = 0.085$ is assumed (Fig.5), then :

$$\eta^{o}_{N_u,max} = \frac{1}{1+\sqrt{\sum_{i=1}^{n}\zeta_i}} = 0.774$$

hence :

$$\eta_{N_u} = \eta^{o}_{N_u}\cdot\eta_p = 0.696$$

The effective power of the gas turbine is :

$$N_e^{GT} = \frac{N_u}{\eta_{N_u}} \cong 1400 \text{ kW}$$

The waterjet propulsion system with gas turbine, located in the stern part of the ship, is shown in Fig.9.

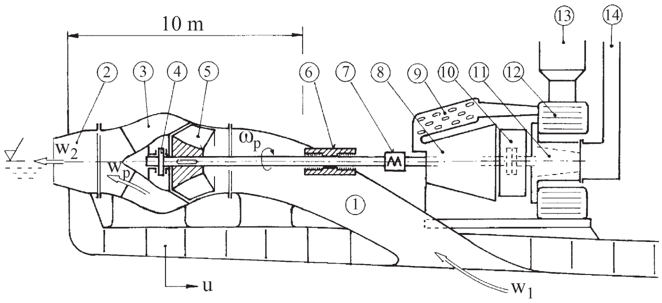


*Fig.9. The waterjet propulsion system located in the ship's stern part: 1 - diffuser; 2 - confusor (outflow nozzle); 3 - guide vanes; 4 - thrust & journal bearing; 5 – diagonal-helical pump; 6 - labyrinth gland/journal bearing; 7 - highly flexible coupling; 8 - gas turbine; 9 - combustion chamber; 10 - gear box; 11 - axial–radial compressor; 12 - recovery heat exchanger (air preheater); 13 - exhaust gas duct; 14 - air inflow duct*

The driving gas turbine with regeneration heat exchanger, is shown diagramatically in Fig.10.

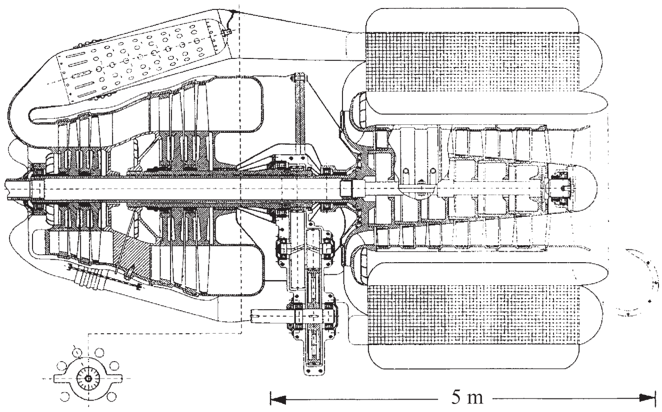


*Fig.10. Longitudinal cross section of a gas turbine with regeneration heat exchanger (suitably shaped and located)*

If the ship's hull is very slender, e.g. of L/B = 15, and stabilized by two pairs of the slim sponsons (s) – see Fig.11, the ship will develop the speed up to 8 m/s (15.55 kt.), at the same output of main engine.

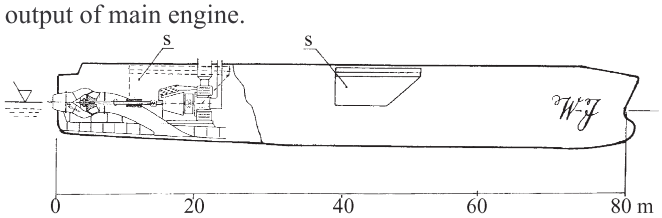


*Fig.11. Profile of a very slender ship (L/B = 15) propelled by a gas turbine waterjet system*

## FINAL REMARKS

- The waterjet propulsion system has some essential advantages in comparison with that conventional based on screw propellers :
  - absence of any rotational elements behind the hull
  - smaller losses due to wake current (of about 13% ÷ ÷ 19% [2])
  - favourable efficiency index at partial loads
  - torsional vibration diminution
  - effective ship turning and reversing
  - possible elimination of rudder blade, resulting in a lower hull resistance.
- Summary losses are inversely proportional to $u^2$, and monotonically decreasing with the ship speed $u$ increasing. They are very weakly dependent on the velocity index $\bar{u}$.
- The great dimensions of flow channel are a serious problem of waterjet arrangement. The outflow area $A_2$ is functionally dependent as follows :

$$A_2 \propto \alpha\left(\frac{\bar{u}^2}{1-\bar{u}}\right)$$

where : α, acc. to eq.(15), is a coefficient dependent on dynamic ship hull properties, whereas the expression $\left(\frac{\bar{u}^2}{1-\bar{u}}\right)$ is directly proportional to the propulsion efficiency, which means that the greater the propulsion efficiency, the greater the nozzle dimensions (!).

- The following problems are important for improving construction of waterjets :
  - smoothness maintenance
  - the pump and flow channel protection against accidentally flowing-in hard pieces
  - development of automatic control elements for ship turning and reversing
  - development of construction of journal & thrust bearing situated inside pump, e.g. a ceramic bearing with water-film lubrication.

**NOMENCLATURE**

| | | |
|---|---|---|
| B | - | ship breadth |
| $c_1$ , $c_2$ | - | relative water speed at inflow and outflow, respectively |
| $d_p$ | - | diameter of pump outflow |
| $d_2$ | - | diameter of confusor outflow |
| $F_u$ | - | propulsion force |
| H | - | waterjet height in relation to sea level |
| HP | - | horsepower |
| kt | - | knots |
| L | - | ship length |
| m | - | flow rate |
| $N_c$ | - | carry-over loss output |
| $N_u$ | - | ship propulsion power |
| u | - | ship speed |
| $w_1$ , $w_2$ | - | inflow and outflow water velocity, respectively |
| $\xi_i$ | - | nondimensional loss of i-th component |

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a) Archimedes axial water pump (287-212 AChN), Leonardo da Vinci pump (1452-1519), Toogwood's & Hayese's patent (1661), Beniamin Franklin's pulsators (1706-1790)
b) So-called „First Hamiltonians"
c) The pump may be of various type: diagonal, helicoidal or propeller pump
d) Water flow rate of two waterjet nozzles

# Automatic control systems for ships fitted with podded propulsion drive (POD)

**Czesław Dymarski**, Assoc.Prof., D.Sc.
**Karol Dąbrowski**, M.Sc., Eng.

ABSTRACT

*The paper shows purposefulness and possiblity of automation of ship propulsion systems , especially those of POD type, intended to be used on four Baltic navigation ships : a containership, ro-ro ship, product tanker, as well as river-sea-going ship, designed within the frame of the Eureka „Baltecologicalship" project. Requirements of ship classification societies, and general ones for modern ship control systems were presented, as well as a review and analysis of currently applied power plant automation systems for diesel- electric ships fitted with podded propulsion drive (POD), were performed. Moreover real ways and possibilities of controlling ship's course and speed were indicated. Information contained in this paper may facilitate selecting appropriate design assumptions for a propulsion control system to be applied on the above mantioned ships.*

**Keywords** : ship propulsion, azimuthal propulsion, ship control systems

## INTRODUCTION

The outfitting of ships with automatic control systems has already been a standard for many years. Such systems can be found practically in any shipboard device. However development of computer and data transmission technology has opened new possibilities for control systems. They are associated with joining control systems into large ones, which is a sucessive phase of development of automatic control for ships.

In a further prospect the development should lead to building ship power plants practically unmanned regarding their automation level. Unfortunately the goal is still very distant at the present development level, mainly due to too low reliability of mechanical devices.

The podded propulsion drive (POD) is a step forward in this respect as instead of a single high-output main engine a set of relatively low-output engines driving electric generating sets can be applied. A failure of one of the engines does not cause the blackout of the entire propulsion system to occur.

The other advantage of the POD system is a large number of electrical elements which can be easily automated.The devices are often fitted with its own microprocessor system and communication interface. This way vast possibilities have been opened for designers of automatic control systems.

The presented paper concerns the automatic control systems for four types of ships : a containership, ro-ro ship, product carrier, driven by azimuth propellers, as well as a river-sea-going ship driven by screw-rudder propellers. It can be assumed that a ship type only to a small degree influences overall structure of automation systems, therefore in this work no separate analysis for each of the ship types was performed.

## REQUIREMENTS OF SHIP CLASSIFICATION SOCIETIES

The ship control systems are to provide high level of safety. The assumption has to be realized by satisfying a number of requirements established in the rules of ship classification societies. Below, the following requirements for ship control systems are formulated on the basis of the rules and publications of Polish Register of Shipping [11,12] :

- ➲ ensuring control of functioning and technical state of all important devices, i.e. electric motors installed in PODs, frequency converters, diesel engines etc, as well as generating appropriate signals and alarms to inform operators on occurrence of a malfunction
- ➲ making it possible to control ship's motion by influencing its propulsion system both from the bridge and other places on the ship, at which such need could appear
- ➲ ensuring an automatic response of the system in the case of hazardous situations.

## ADDITIONAL REQUIREMENTS FOR CONTROL SYSTEMS OF POD-FITTED SHIPS

The PODs differ substantially from the classical ship propulsion system of a fixed or controllable pitch propeller and passive rudder system. Their essence consists in joining together propeller and rudder systems into the whole. In this case the propeller simultaneously fulfils the role of active rudder. Hence the automatic control system has to integrate both the function. Even if it is composed of two subsystems : steering and propelling, they will be expected to be functionally integrated into an entity. First of all they should be felt by the user as a single system. For this reason it is very important to appropriately design and locate control heads.

However the POD does not substitute all ship manoeuvring devices. For manoeuvring in ports bow thrusters are used. Therfore they are also provided on the designed ships in question. Because of high availability of electric energy on the ships it is recommended the thrusters to be electrically driven as it makes it possible to connect their control systems with the POD control system. This way ship's crew can have at its disposal a complete ship control system both during sea voyage and port manoeuvring, integrated in a single panel which makes it possible to control all propulsion and manoeuvring devices.

Such solution makes it necessary to design a superior system responsible for control of information flow within the entire system. It may be a separate computer which, on the one hand, could control communication between control systems of particular devices : i.e. frequency converters, main engines, switching devices etc., on the other hand it could serve as a module making it possible to hand-over control from one to another control heads. It is assumed that direct control over a given device can be executed from one control head only, hence a system giving assent for such control is necessary in this case.

The remaining requirements for a control system one can describe by dividing it into the following constructional elements :

- **System's wiring** – should ensure fast and safe data transmission. Simultaneously, number of communication cables should be limited. The first postulate could be best fulfilled by lightguide connections having rotatable optical connectors installed in the places where the cables pass from PODs to the deck.The best way to limit number of cables is to apply data buses (e.g. profi-BUS, an up-to-date and still modernized solution). This solution has been used for many years and it has proved to be correct.
- **Drive control and supervison system** – should prevent against exceedance of limiting values in frequency converters. In the case when the control system covers a greater number of devices (e.g. thrusters) several drive systems are involved.
- **Drive system** – contains logic and drive supervision system. It analyzes operation of supply network, motor, transformers and remaining drive control elements. Its work is aimed at :
  - automatic switching-off in the case of exceedance of appropriate values in power supply circuit
  - safeguarding against overloads and shortings
  - checking the system before starting, generating availability signals and stopping the drive in case of incorrect values of operational parameters
  - control of starting and stopping the drive
  - rotational speed control of driving motor
  - communication with the remaining automation elements
  - switching on/off of activity of control panels on ship.

  Practically this is a management system which plays a decisive role in drive operating.
- **Data transmission control** – ensures distribution and recording of the data obtained from transducers and measuring instruments, moreover it makes communication between control heads and the drive control & supervision system, possible.
- **Operator panels, control heads** – serve as basic communication means between ship's crew and automation system. It should be remembered that to apply a POD system the ship is be fitted with an electric power plant , associated switching devices and a transforming station. They are also equipped with an automatic control system of electric power production and distribution. All the systems should be coupled together, and they should be made accessible from control heads grouped in one place. It is also crucial to design interfaces of operator panels in such a way as to ensure easy operation of the system, without any additional cost of ship's crew training.

A separate problem is to ensure cooperation with an autopilot. It can be realized by making use of a data transmission control system. The autopilot would be considered by such system as an additional control panel, without any possibility to control drive of thrusters. However the problem of a way of ship course-keeping remains to be solved. In the case of applying the POD system it can be attained in two ways :

1st – in the same way as it is realized in the case of the conventional propulsion system ( screw propeller and passive rudder), i.e. by controlling direction of propeller thrust vector. The solution requires the entire POD to be rotated, but rotational speed of propellers can be maintained constant.

2nd – the other way is to change rotational speed, hence by changing the thrust delivered from the propeller of one POD only. Such action would produce a non-uniform propulsion generating a ship course changing moment. Choice of one of the possibilities should be based on an effectiveness analysis of both solutions and their energy efficiency. From the point of view of automatic autopilot system both the versions are rather easily realizable.

## ANALYSIS OF POSSIBLE APPLICATION OF DIFFERENT CONTROL SYSTEMS

A system approach is necessary for designing a modern ship propulsion system fitted with electric power plant and switching station . An analysis of a ship as the whole , and not as a sum of its particular equipment elements, may help in lowering risk of malfunctioning, incorrect communication between devices etc. Also, users of such system may be easier and faster trained in operating it. In the case in question continuous development of user's interfaces (operator panels) is also of a great importance. It makes faster and more effective reacting by ship crew during ship operation, possible, as well as that special qualifications for operating automation devices by ship crew, are not required.

The ship electric power plant supplies all driving devices and delivers electric energy to the ship electric network. As all supply systems are mutually connected the intergrating of automation systems into a global system whose elements are capable of adapting their work to an energy situation of the entire ship, becomes rational.

### *Propeller thrust control*

Two kinds of ship propulsion control are possible: maintaining a set value of ship speed, and maintaining a set value of power. In the first case a speed setting will be translated into an appropriate number of propeller rotations. It will cause demand for engine output changeable. In the case of the engine output control a reference value is the main shaft power, and the shaft rotational speed should be so changed as to maintain the power value constant.

Setting an appropriate speed value on the bridge should initiate a power changing program which determines slope values of power rising and trailing edges in function of the set value and current load. It has to prevent the propulsion system from an overload at too sudden speed change. An example diagram showing increase of main shaft power, speed and torque is presented in Fig.1.

| Speed change versus load | | Power change versus load | |
|---|---|---|---|
| Range | Slope of characteristics | Range | Slope of characteristics |
| 0-50% | 10%/s | 0-5% | 5%/s |
| 50-80% | 1%/s | 5-50% | 1.25%/s |
| 80-100% | 0.7%/s | 50-100% | 0.5%/s |

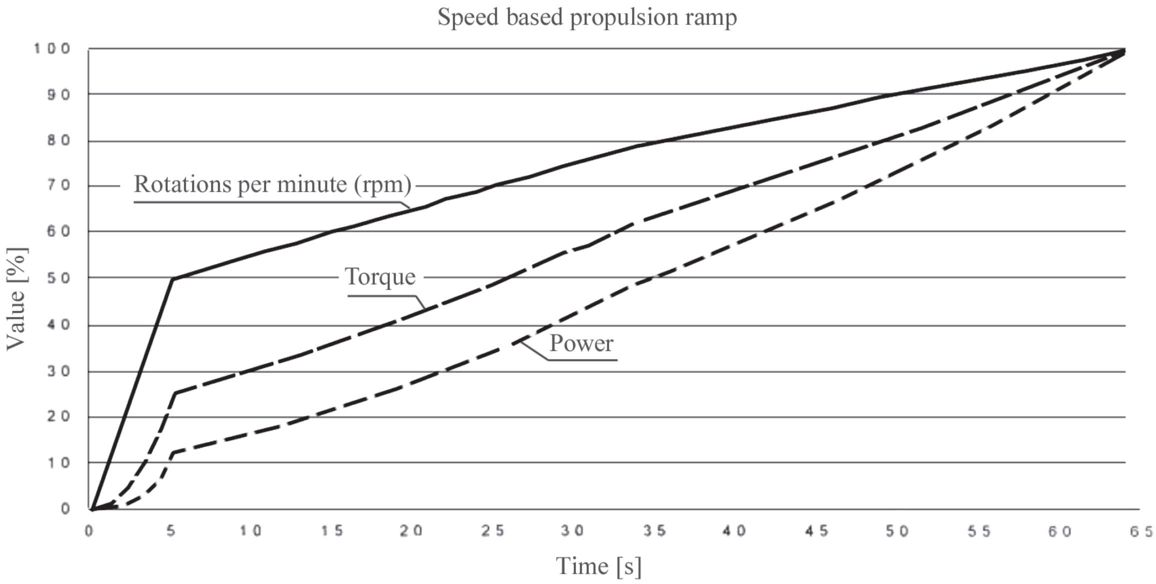


***Fig. 1.*** *An example diagram showing increasing course of main shaft power, speed and torque in function of time*

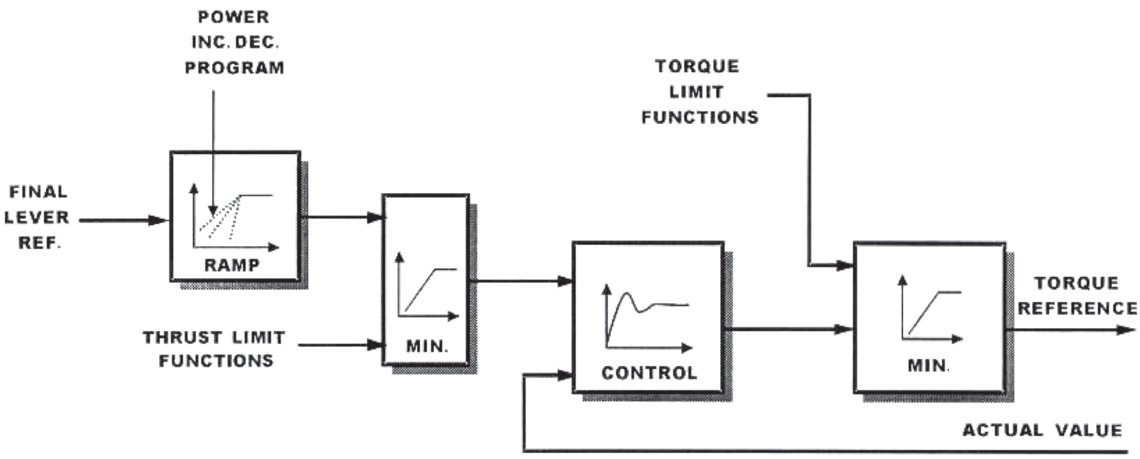


***Fig. 2.*** *Schematic diagram of the ship speed control procedure*

As soon as a power level becomes steady a limiting value of propeller thrust is checked. The maximum set value of ship speed will be hence dependent on limitation of propeller thrust force. Next the system checks a value of engine shaft torque and adjusts its rotations in such a way as not to exceed the maximum shaft torque. After performing the procedure an appropriate value is sent to the engine.

A sudden drop of rotational speed of the engine should cause the engine passing onto generating work mode and transferring a part of its output to an inverter and network, and a power surplus to a resistor.

## *Propeller thrust direction control*

Rotation of propeller's POD makes it possible to produce large forces transversally directed to ship's longitudinal axis. It may be hazardous as a large heeling moment appears in such situation. Apart from a large heel of ship, which itself is hazardous (possible shipping of water on deck) it generates large accelerations which may cause a shift of cargo and – in consequence – trigger other hazards including loss of stability and ship's capsize. Hence the control system is required not only to control a propeller thrust value depending on a value of POD's rotation angle, but also to limit speed of its rotation. The speed limitation should not be dependent on an actual thrust value. At its low values the POD's rotation can be executed rather fast. Greater thrust values should cause POD's rotation speed lowering.

## *Limitation of power level in the system*

A number of generators running in the system depends on an actual power input. Appropriate limits for power input after exceedance of which the number of running generators is changed by one, are established. This way the energy production becomes adjusted to the power demand from the side of ship's network.

## *Limitation of engine output*

In the case of sudden manoeuvres it may occur that engin's output or torque reaches its respective maximum value earlier than set speed value. In the situation the system should be capable of stopping the rotational speed increasing.

Besides, in the case of automatic switching - off the generators the control system reduces engine output to such an extent as to adjust its value to the power available in the network.

## *Requirements for safeguarding system*

It is necessary to control the following :

- pump operation parameters of cooling system
- temperature of stator's winding of electric motor
- temperature of lubricating oil or bearing
- temperature of transformer's winding
- overloads of frequency converter
- temperature of frequency converter's cooling medium.

If any of the above enumerated parameters is not contained within its permissible range the system has to start procedure of propulsion power lowering.

The control system should be capable of auto-diagnosing. Due to the function it is possible to monitor : all programs executed by the system, values of voltage on auxiliary devices, supervision over communication system , I/O panel of frequency converter, as well as analogue input signals. Besides it is important to ensure possiblity of recording course of the system's operation in order to perform later its analysis in the case if ship's crew is not able to cope with a failure.

## *Starting and stopping main drive*

Starting the engine is usually executed from Central Control Station (CCS). All auxiliary devices such as :

- cooling water pump of engine cooling system
- cooling water pump of inverter cooling system
- fan of engine cooling system
- auxiliary voltage source of inverter
- pumps and fans of cooling system

are to be started separately before starting the main engine.

The following sequence of the main drive starting operation should be realized :

- switching-on an appropriate number of generating sets
- starting auxiliary devices
- releasing the engine's breaking system
- starting the main engine.

Stopping the engine should consist in lowering its rotational speed up to zero and next in the following :

- switching-on the breaking system
- switching-off auxiliary devices
- successive switching-off the generating sets.

## *Synchronization of drives*

In the case of applying a greater number of drives on a ship it is necessary to synchronize their work to avoid beating which happens if their rotational speeds are incompatible. To synchronize their work the following conditions are to be complied with :

- synchronization function is to be activated by means of a graphical operator panel
- difference between the reference rotational speed and actual speeds of drives cannot be greater than the value established by their producer
- value of the reference rotational speed of each drive cannot differ by more than the number of rotations per minute (rpm) established by its producer
- actual values of rotational speed of drives cannot differ by more than the number of rpm established by their producer
- communication between drives is to be realized through a data bus.

However if synchronization of drives were necessary then it would be rather impossible to keep the ship on a set course by changing the rotational speed of one of the drives only.

## POSSIBLE CHOICE OF A CONTROL SYSTEM

During the analysis of control and automation systems for a POD-fitted ship the following products of two leading producers were considered :

- PodCon and Marintronics™ systems of ABB firm
- Alstom - Mermaid™ system of Rolls-Royce Group.

All the products totally cover the automation system of ship propulsion and manoeuvring, and they are intended for POD drives. Additionally the products can be shaped into many configurations and matched up to specific features of a ship. More information on the systems can be found in [14] and the producer technical pamphlets referred to in the below attached Bibliography.

From the information contained in the sources it results that both firms : ABB and Rolls-Royce-Alstom intend to introduce the global supervising systems. It makes it possible to greatly improve efficiency of a power system, and to reduce number of ship's personnel in the future. Even a not large decrease of shipping costs may improve competitiveness of a given shipowner.

It is very important to correctly manage the power in changeable load conditions (frequent speed changes, switching on/off the thrusters etc), as then profits associated with lowering fuel oil consumption will be manifested the most. Correct adjusting a number of generating sets which supply the network, depending on instantaneous power demand is the area of the greatest possiblities in improving the power system effectiveness.

In such waters as the Baltic Sea, time of sea voyage of a ship is rather short, hence the share of duration time of port manoeuvers is substantial. Therefore application of an integrated ship management system may bring in measurable profits in the form of lowering operational costs to such an extent as to cover initial cost of the system.

Choice of a definite system is not easy; however the problem goes beyond the scope of this paper.

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FOUNDATION FOR SAFETY OF NAVIGATION
AND ENVIRONMENT PROTECTION

# SHIP HANDLING RESEARCH AND TRAINING CENTRE AT IŁAWA

The safe handling of ships depends on many factors such as ship's manoeuvring characteristics, human factor (operator experience and skill, his behaviour in stress situations etc.), as well as environmental conditions, and possible water area restrictions. The interface between ship operator and the available control devices is also a very important factor.

Results of analysis of CRG (collisions, rammings and groundings) casualties show that human errors are involved in one third of them, and the same amount of CRG casualties is attributed to the poor controllability of ships. Training on ship handling is largely recommended by IMO as one of the most effective methods for improving the safety at sea. The goal of the above training is to gain theoretical and practical knowledge on ship handling in a wide number of different situations met in practice at sea.

The main advantage of the method of training on ship handling based on manned models is realistic representation of all complex hydrodynamic phenomena affecting ship manoeuvrability. The models which satisfy similitude criteria have hydrodynamic characteristics very similar to those of real ships. This is very important e.g. for manoeuvres in restricted waters and ship-to-ship interaction, where bow cushion and suction areas considerably influence ship behaviour. Also manoeuvres where hull-propeller-rudder interaction plays a significant role can be reproduced on models in a realistic way.

Training on manned models makes it possible to account for psychological aspects of training in comparison with that on an electronic simulator, by better feeling of effects of groundings, rammings and collisions, as well as environmental effects such as wind and current.

Since 1980 more than 2000 ship masters and pilots from 27 countries have been trained at Ilawa Ship Handling Research and Training Centre in weekly courses. The Foundation for Safety of Navigation and Environment Protection, being a non-profit organisation, is re-investing all spare funds in new facilities and each year new models and new training areas have been added to the existing ones. Also, the existing training models are modernised each year.

Currently six models representing a wide spectrum of ship types are available at the Centre for training purposes. The ship models are presented in the table and figure. The models are equipped with all necessary devices simulating ship's systems, and the basic navigation aids (gyro, log, GPS, navigational lights, wind velocity and direction sensor, etc.).

Last year a new system for simulation of tug–ship co-operation entered into service. The system was installed on VLCC, LCC and Large Container models, and similar system will be soon installed onboard other ship models.

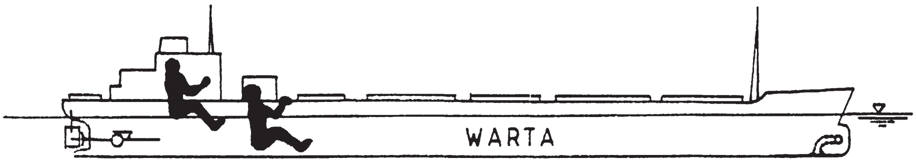
In summer 2000 the Training Centre at Iława was equipped with a high precision tracking system based on GPS technology. At present the GPS receivers working in RTK mode and allowing to measure the model's position with accuracy of 0.01m are installed on two models, and in the next years similar receivers will be also fitted on other models. The new installation improved very much quality of training.

The following training areas are currently available :

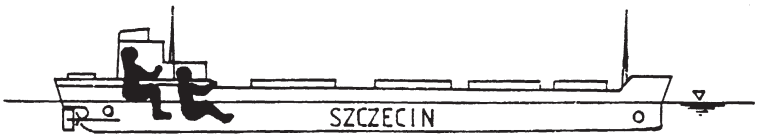
- Mock-ups of harbours with different configuration of entrances, turning areas, piers and quays, ferry terminals, docks and locks. The main exercises realised by using these training areas are : approaching harbours and terminals, berthing and unberthing under various conditions, with wind and current present, entering the lock .
- Training areas including straight line and curvilinear waterways and sharp turns. All passages are marked by buoys and leading marks for daylight exercises and a set of navigational lights for night-time exercises. Main current generator is also installed in this area.
- Open sea training area, which includes curvilinear deep water canal, sea berths, and a SPM (single point mooring buoy). This area is also used for realisation of standard manoeuvres, rescue and search manoeuvres with taking into account MERSAR recommendations. Ship – to – ship manoeuvre can also be trained there.

***Main particulars of the models***

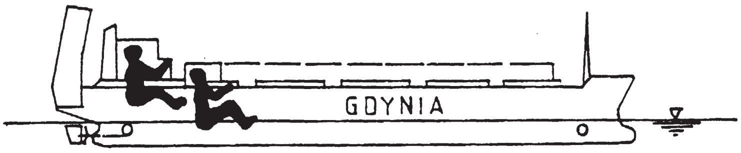
| Model's type | LCC tanker | Panamax bulk carrier | Ro-ro ship | Passenger car ferry | VLCC tanker | Large container |
|---|---|---|---|---|---|---|
| Length/Breadth $L_{BP}/B$ | 5.79 | 6.72 | 5.87 | 5.41 | 5.68 | 7.08 |
| Length/Draft $L_{BP}/d$ | 18.13 | 16.96 | 19.16 | 23.54 | 15.73 | 20.98 |
| Breadth/Draft B/d | 3.13 | 2.52 | 3.26 | 4.35 | 2.77 | 2.97 |
| Block coefficient $C_B$ | 0.884 | 0.774 | 0.634 | 0.687 | 0.830 | 0.646 |
| Number and type of propellers | 1 FP | 1 FP | 1 FP | 2 CP | 1 FP | 1 FP |
| $D_{Prop}/d$ | 0.46 | 0.48 | 0.68 | 0.78 | 0.44 | 0.63 |
| Rudder area ratio $A_R/(Lxd)$ [%] | 1.76 | 1.39 | 1.82 | 2x1.33 | 1.75 | 1.85 |
| Scale | 1:24 | 1:24 | 1:24 | 1:16 | 1:24 | 1:24 |



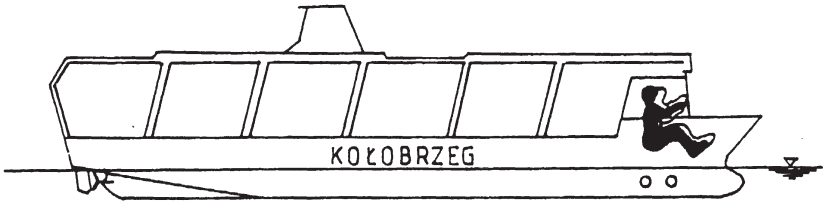

*Model of a LCC tanker*



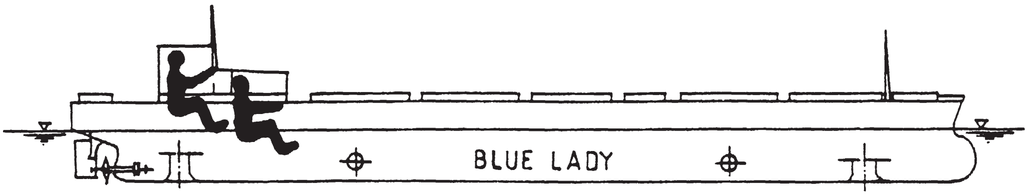

*Model of a Panamax bulk carrier*



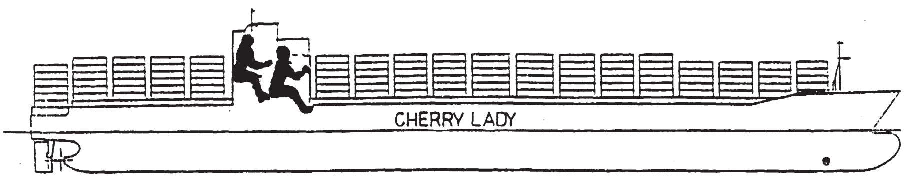

*Model of a ro-ro ship*




*Model of a twin screw passenger car ferry*




*Model of a VLCC tanker*




*Model of a large container ship*

***Ship training models used at Ilawa Centre***

- Shallow water curvilinear canal. It consists of restricted cross--section canal with a bend and straight sections, 140 metres in length. The canal is mainly used for passing and overtaking manoeuvres.
- River training area consisting of curvilinear stream restricted from both sides where current is created. Piers for mooring ship models in current are installed there.
- Restricted water areas for demonstration of bank and shallow water effects.

On the shore site there are two hangars for storing models, as well as slipways and the building which contains lecture, recreation and computer rooms, staff accommodations, stores, small workshop, and other facilities.

The scope of lectures and programme of practical exercises is flexible. Each year new items are including into the course programme, so that the current programme differs much from the training realised some years ago. The new training areas together with the new models of modern ships (the seventh model representing a LNG carrier is under construction now) and the modernized and new equipment of training models make now possible the realisation of training on specific types of ships. It is also possible to arrange special courses focused on chosen problems of manoeuvrability as well as to realize training on specific types of ships.

**Jacek Nowicki**