UDC 629.5.01:629.543
629.5.01:629.546.2

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Design Procedure and Mathematical Models in the Concept Design of Tankers and Bulk Carriers

Original scientific paper

Paper presents design procedures and mathematical models applicable in initial design of merchant ships with high block coefficient. Special attention has been paid to two dominant ship`s groups: tankers and bulk carriers. Presented design procedure is common for both groups and it can be applied using various application techniques: from the simplest handy methods to the most sophisticated optimization methods and techniques. Presented mathematical model includes optimization of main ship characteristics as well as optimization of commertial effects of newbuildings. Mathematical models are based on designer´s long-time work experience. Large number of data has been derived from more than 150 executed designs and more than 40 ships built in *Shipyard Brodosplit*. Recommendations for execution of design are shown in number of pictures and diagrams. Presented design procedure and mathematical models have been applied in the multiattribute decision support optimization programme developed in *Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb*.

Keywords: *bulk-carrier, full hull forms, mathematical modelling, multiattribute approach, ship design, tanker*

Projektne procedure i matematički modeli u projektiranju brodova za tekuće i rasute terete

Izvorni znanstveni rad

U radu su razvijene projektne procedure i matematički modeli za osnivanje trgovačkih brodova pune forme. Posebna je pozornost posvećena dvjema dominantnim skupinama ovakvih brodova: brodovima za prijevoz rasutih tereta i brodovima za prijevoz tekućih tereta (tankerima). Izložena projektna procedura je zajednička za obje skupine i može se primijeniti u postupku osnivanja broda različitim metodama: od najjednostavnijih metoda priručnim alatima do suvremenih složenih optimizacijskih metoda i postupaka. Prezentirani matematički model osnivanja broda se zasniva na dugogodišnjem projektantovom iskustvu. Iz više od 150 izvedenih projekata i više od 40 izgrađenih novogradnji u *Brodogradilištu Brodosplit* je selektiran veliki broj podataka o brodovima. Zasnovano na tim podacima su dane preporuke i za projektiranje koje su prikazane slikama i dijagramima. Izložena projektna procedura i matemački modeli su primijenjeni u višeatributnom programu za sintezu projekta razvijenom na *Fakultetu strojarstva i brodogradnje* u Zagrebu.

Ključne riječi: *brodovi za prijevoz tekućih tereta, brodovi za prijevoz rasutih tereta, modeliranje, projektiranje broda, pune forme, višeatributni pristup*

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Received (Primljeno): 2008-02-19
Accepted (Prihvaćeno): 2008-06-03
Open for discussion (Otvoreno za raspravu): 2009-12-31

1 Introduction

Over years, the development of merchant ships has been directed to obtaining increasingly higher deadweights without increasing main dimensions of the ship or decreasing the ship speed. This trend, very often contradicting the designer's beliefs, is caused by commercial effects of the ship operation. To put it in simple words, full hull form of merchant ships with bigger deadweight brings higher profit to the shipping company. In view of that, there is a real competition going on in the design and building of ships with deadweights quite unimaginable until very recently. In order to achieve the targeted deadweight, the designer has at his disposal only two possibilities: to reduce the ship's light weight or to choose the full hull form with a high block coefficient.

This trend in the development of full hull form merchant ships, of bulk carriers and tankers in the first place, started in Japanese shipyards some thirty years ago. A few years later, Korean shipyards joined the Japanese ones, and then all other shipyards, which had been trying to be competitive in building these ships, joined them. The magnitude and power of the Far East shipyards have caused the development of own projects. While shipyards are building "mass-produced" newbuildings with minimal modification possibilities during the building process, a new generation of a "standard" design is being developed simultaneously. When completely developed, it will replace the one of the previous generation. These designs have reached the very frontiers of current technical knowledge; therefore they are made very difficult to compete with.

The ship design development in less powerful shipyards, including Croatian ones, is completely different from the Far East model. In order to accommodate the design to specific requirements of potential customers, it is defined on the level of conceptual and partially preliminary design before the shipbuilding contract is signed. The completion and detailed development of the design is postponed for the post-contract phase, so that they overlap with preparatory activities for the shipbuilding process, and very often with the building process itself.

In such a situation, designers have a very short time at their disposal. Basic design assumptions cannot be confirmed in the pre-contract phase; therefore, designers are forced to take some risk while developing their design. In order to minimize the risk, it is of vital importance to base the design in its conceptual and preliminary phases on quality design procedures and adequate mathematical models.

Therefore, the development of design methods and the application of modern optimization techniques in all phases of ship design have a major importance. Without a continuous development it is not possible to retain the position of one of leading countries in the modern ship design development, which is an indispensable precondition of further strengthening the position of Croatian shipyards at the world shipbuilding market growing ever so more competitive.

This goal can be achieved only by a continuous development and by sharing experience and ideas between all shipbuilding centres: shipyards, scientific and shipbuilding institutions. The purpose of this work is to give a modest contribution to the improvement of basic design and to the application of optimization procedures in the design of full hull form ships.

The basic aim of this paper is to give a systematic and comprehensive overview of the conceptual design of ships which dominate the world shipping fleet. The paper represents the design procedure of ships with a high block coefficient, primarily of modern bulk carriers and tankers. The presented design model can be applied to a wide variety of design tasks and with different working techniques.

The design procedure and mathematical models used for the design of full hull form merchant ships presented in this paper are based on a number of successful designs and nebuildings of the *Brodosplit* shipyard in the past fifteen years. The applied design procedure is built on and extends the so-far publicized design models [7, 8, 9, 10, 11, 12, 13, 14, 15].

In this paper at first are shortly described common basic features of full hull form merchant ships, i.e. of basic elements which have a dominant influence on the design procedure. It also gives a summary of reasons for choosing full hull form ships, of interrelations and cause-effect relations between particular influential factors and ship elements, as well as of solutions to basic problems.

The next section gives a classification of merchant ships with a high block coefficient, a list of particulars for two dominant groups of vessels, i.e. of bulk carriers and tankers.

The fourth section gives a detailed description and a more precise definition of specific problems encountered in the design of vessels belonging to these two groups, based upon published papers [14, 25, 26].

Bulk carriers are divided into two major groups: ore carriers and ships for the transport of light bulk cargo. Their typical cross-sections are given in the relevant figures and their main particulars are described. A short description of transported cargoes and the related problems is also given, together with basic factors determining the design of these ship types.

Tankers are divided into vessels for carriage of special liquid cargoes, vessels for carriage of liquid cargoes that need to be cooled down or heated to high temperatures, chemical carriers, crude oil carriers and oil product carriers. A short description of all groups is given. The figures represent typical cross-sections of dominant groups: crude oil tankers and product tankers for carriage of petroleum products and less hazardous chemical substances. A description of basic characteristics affecting the design of these vessels is given at the end of the section.

The fifth section deals with international legislation and requirements of classification societies, which refer to the relevant ship types [1, 2, 3, 4, 5, 6]. The SOLAS rules defining requirements regarding bulkheads and stability are given in a short overview, as well as the basic MARPOL rules referring to the tanker cargo space configuration and stability requirements, the ICLL rules used for the calculation of the minimum freeboard and the basic classification society requirements affecting the basic ship structure. In addition, rules and constraints of the three most important canals, i.e. the Suez Canal, the Panama Canal and the St. Lawrence Canal, are briefly outlined.

The next section represents in detail the mathematical models for the design of full hull form merchant ships. Basic input data and their classification are defined. In addition, criteria which can greatly affect the choice of optimum design are listed and explained. The author represents his subjective designer's suggestions and constraints through graphical representations. He also represents his data bases for particular ship types and sizes, gathered from his own experience, to be used as an auxiliary means in the calculation of particular groups significantly affecting the total weight of the ship.

The final section gives conclusive considerations of this work. The applied procedure is commented on and compared with traditional design methods. Possible advantages of the applied procedure in daily shipbuilding practice are described and, finally, suggestions for further development and improvement of the presented methodology are given.

2 Main Characteristics of Full Hull Form Merchant Ships

Main characteristics of full hull form merchant ships feature the following: a high block coefficient (C_B), generally ranging from more than 0.80 to the highest value of 0.89; moderate speeds characterized by the Froude numbers from 0.15 to 0.20; heavy wake fields in the plane of propeller operation; a higher degree of risk due to flow separation around the propeller; a high ratio of cargo holds volume to the total volume of the ship; moderate power of main engines; short engine rooms with adverse effects on the design of propulsion system; and finally, high efficiency in service. The latter of the listed characteristics is a dominant feature which is the cause of continuous efforts focused on improving and perfecting technical solutions of all other features and related problems.

Full hull forms are characterized by a heavy wake field in the plane of propeller operation, i.e. a high wake. This problem is being alleviated by the development of new generations of hull forms which are intended to improve the wake field and to maintain the value of the full hull form block coefficient at the same time [9, 10, 11, 12].

The present hull forms have a pronounced aft bulb, i.e. U-shaped stern lines (gondola). This bulb form results in a slight deceleration of the mean wake, resulting in a more uniform wakefield, and, consequently, easier and more efficient performance of the ship's propeller. Naturally, there are some undesired side effects, such as the lack of space in the engine room, poor seakeeping in following waves, and more complex hull structures for the stern.

At present, full hull form merchant ships are predominantly bulk carriers and liquid cargo carriers (tankers). Both ship types have similar hull forms, but tankers, as freighters of a higher quality (more expensive) cargo, can reach a bit higher speeds, i.e. higher Froud numbers. Although these two ship types are completely different with respect to the type of the cargo, general ship configuration and relevant regulations, they do have some common characteristics [14, 25, 26].

Both ship types need cargo holds/tanks with high cubic capacity. Also, in most cases, the main dimensions of the ship are limited by their particular route, e.g. the St Lawrence Seaway, the Panama Canal, the Suez Canal or some ports. Despite distinct differences in their structure, the longitudinal strength and the structure of both ship types depend on the same loading conditions.

The characteristics and design problems related to the full hull form ships discussed above are just a consequence of their high commercial value in service. A comparison of previous generations of standard size ships with the present projects shows clearly a trend towards the development of increasingly fuller hull forms. A question remains where the ultimate limits are and how they can be reached.

3 Classification of Full Hull Form Merchant Ships

There are two major groups among full hull form merchant ships: bulk carriers and tankers.

High block coefficient ship forms are applied to some specific designs of merchant ships intended for other purposes (in cases when main dimensions are strictly limited, and the speed requirement is not of major importance) and to specialized vessels (e.g. draggers). As these ships have a small share in the world fleet, and have very few common characteristics, only two prevailing groups will be considered:

- bulk carriers, and
- tankers.

Bulk carriers have the following main characteristics:

- high block coefficient,
- moderate speed,
- one (main) deck,
- high cubic capacity of cargo holds (with the exception of ore carriers),
- short engine rooms and peaks,
- accommodation and engine room positioned aft,
- minimum/reduced freeboard,
- vertically corrugated transverse bulkheads (only in rare cases double-plated bulkheads),
- large hatches (the width of hatches is equal to or greater than the half beam),
- specific cross-section with double-bottom, bilge and wing tanks (requirements for double side are expected to be regulated).

The main characteristics of tankers are as follows:

- high block coefficient,
- slightly higher speed,
- one (main) deck,
- high cubic capacity of cargo tanks,
- short engine rooms and peaks,
- accommodation and engine room positioned aft,
- freeboard exceeding minimum requirements,
- plane or corrugated bulkheads in cargo holds (depending on the ship size and the "quality" of the cargo),
- deck structures below or above the deck (depending on the ship size and the "quality" of the cargo),
- cross-section with double bottom and double sides.

4 Specific Design Characteristics of Particular Ship Types

The presented classification of bulk carriers and tankers and specific design characteristics of these ship types are based on the author's design experience and relevant literature [14,25,26].

4.1 Design of bulk carriers

Modern bulk carriers can be generally divided into two main groups:

- ore carriers for the transport of ore and other heavy dry bulk cargo;
- ships for the transport of light bulk cargo (grains, light ores).

The former group of ships is characterized by high density of the intended cargo, hence by a narrow specialization. The required cargo holds capacity is relatively small in relation to the cargo mass. Therefore, satisfying the requirement of the minimum volume of buoyancy entails ballast tanks of a large volume. Generally, it is sufficient to satisfy the requirement of reduced minimum freeboard. High specific cargo mass is the cause of a low centre of gravity in loaded condition, i.e. "over stable" ship with a stiff ship behaviour on waves. Accelerations occurring in such conditions are inadequate for a long-term quality accommodation of the ship's crew and for neat operation and good maintenance of particular ship's equipment. This problem is dealt with by lifting the cargo position.

Considering the problems stated above, there are two possible solutions: increasing the height of double bottom above the required minimum (either by regulations of classification societies, by conditions for the ballast tanks minimum volume or by results of ship structure optimization) and/or adaptation of the cargo holds geometry with sloped longitudinal bulkheads. Since these vessels have a very narrow specialization, there are not many of them and they usually have high deadweight (capsize). A typical cross-section of an ore carrier is given in Figure 1. Considering the problems stated above, there are two possible solutions: increasing the height of double bottom above the required minimum (either by regulations of classification societies, by conditions for the ballast tanks minimum volume or by results of ship structure optimization) and/or adaptation of the cargo holds geometry, i.e. by sloping longitudinal bulkheads. Since these vessels are highly specialized, they represent a smaller number of bulk carriers and are of large sizes (Capsize ore carriers). A typical cross-section of an ore carrier is given in Figure 1.

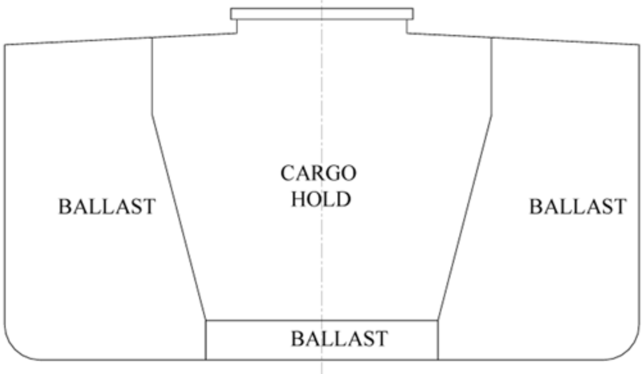


Figure 1 **Typical cross-section of an ore carrier**
Slika 1 **Tipični poprečni presjek broda za prijevoz rudače**

The ships usually called "bulkers" or bulk carriers belong to the latter group in the previously represented classification. They are greater in number than ore carriers, more universal and their exploitation for the transportation of various bulk cargoes, or even general cargo, is economically feasible.

Although the large capacity of their cargo holds enables them to transport relatively light cargoes at the scantling draught (cargo density of approximately 0.8 kg/m^3), modern "bulkers" can also transport very heavy cargoes. In such cases, the ship is alternatively loaded into particular, specially strengthened holds. Alternative loading is carried out into odd holds, i.e. holds number 1, 3 and 5 for "handy" size, 1, 3, 5 and 7 for "laker" and "panamax" size, and 1, 3, 5, 7 and 9 for "cape" size.

When loading very light cargos, especially timber, the cargo holds capacity is not sufficient for loading the ship up to its maximum draught. In that case, the cargo is also loaded onto the open deck. The cargo on the deck is secured by special deck equipment. This loading condition is specially considered in the calculation of minimum freeboard.

These ships can also transport packed cargo. Until recently, the most common requirement was the transport of containers on the open deck or in cargo holds. In those cases, ships were additionally equipped by fixed and portable equipment for cargo fastening. This loading condition does not greatly affect the ship design as it represents only a possibility of carriage of an additional cargo type. The only thing to be dealt with is to adapt the geometry of cargo hatches to the standard container dimensions, and possibly to maximize the number of containers by adapting the beam and depth of the ship, as well as by the general arrangement of the main deck.

Recently, there have been requirements for the transport of semi finished steel products (steel coils, mostly). Such cargos do not greatly affect the ship design in the initial phase, but they affect the later phase of ship steel structure dimensioning (inner bottom plating) and loading condition calculation (packed cargo with a great number of possible position variations).

Bulk carriers are characterized by minimum capacity of ballast tanks. While sailing in a light ballast condition, it is important to achieve the aft draught which enables minimum immersion of the ship propeller and its cavitation-free operation, and the fore draught to avoid slamming in most cases. Safe sailing on heavy seas and the minimum draught for passing through the Panama Canal are obtained by ballasting one or more cargo holds. To enable that, it is necessary to design and construct a cargo hold and a hatch cover for that particular loading condition, and to equip it with devices for the ballast loading/unloading.

The design of bulk carriers is commonly characterized by the following:

a) standard size:
   - lake freighters or lakers –ships that can sail the Great Lakes;
   - Handy and Handymax vessels of 35,000-40,000 dwt or over 50,000 dwt, respectively, with a limited beam to be able to pass through the Panama Canal, and, with the maximum draught of up to 40 feet (12.2 m);
   - Panamax vessels that can pass through the Panama Canal, and, in most cases, with the length over all limited to only 225 m;
   - Capesize vessels – the biggest vessels for carriage of bulk cargo, deadweight of approximately 170,000 dwt;
b) large volume of cargo holds;
c) general configuration with 5 to 9 cargo holds (depending on the size of the vessel);
d) reduced freeboard (B-60);
e) moderate speed (generally 14.5 to 15 knots in the trial sailing conditions and at the design draught);
f) the use of high tensile steel;
g) typical cross-section represented in the following figure.

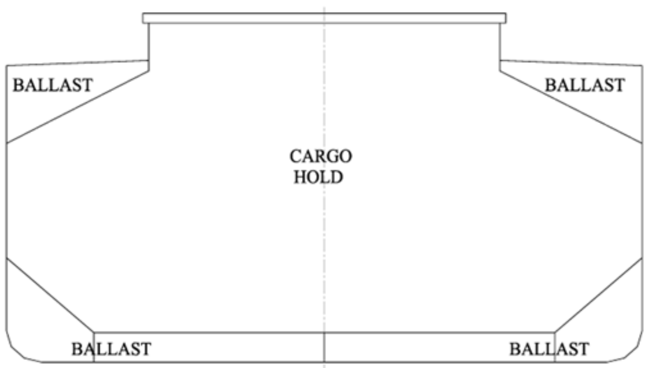


Figure 2 **Typical cross-section of a bulk carrier**
Slika 2 **Tipični poprečni presjek broda za prijevoz rasutih tereta**

The cross-section is characterized by a low double bottom – of the minimum height required by regulations of classification societies or slightly higher (in case it is required by technological causes or as a consequence of optimization of the double bottom structure). The inclination of bilge tanks is usually set at an angle of 40^o, which enables efficient cargo unloading, as well as the structural design of aft cargo hold end (stern frames in this area of the ship are rather "sloped"). In some designs, the inclination is at smaller angles, which makes the design of the stern structure easier, but increases the frame span.

The geometry of topside tanks is determined by the deck hatch width and the angle of rise of the tank bottom. This angle is set at approximately 30°, which satisfies the condition of normal loading of most cargoes (angle of repose of bulk cargoes).

In the design of modern "bulkers", the requirement for a great width of cargo hatches is very important, in the first place because of easier cargo manipulation and handling. Hatch widths range from the values slightly lower than the half beam of the ship with side rolling hatch covers to 55-60% of the ship beam with end folding hatch covers. This situation entails a more complex solution of the deck framing.

4.2 Design of Tankers

Tankers may be generally divided into the following groups:

- ships for carriage of special liquid cargoes (water carriers, tankers for carriage of natural juices and oils, ships for carriage of urea, etc.);
- ships for carriage of liquid cargoes that need to be cooled down or heated to high temperatures;
- chemical carriers;
- crude oil carriers and oil product carriers.

The first group comprises highly specialized ships with their basic particulars and designs are strictly determined by the properties of the cargo they carry. In the total number of tankers in the world fleet, they represent only a very small group. Due to their special features, they can be considered as special purpose ships; therefore this group will not be dealt with in this paper.

Tankers for carriage of liquid cargoes that need to be cooled down or heated to high temperatures have a common property that their tanks are subjected to high thermal delatations due to a big difference in the temperature of the cargo and of the environment. This group incorporates liquefied natural gas vessels (LNG tankers), liquefied petroleum gas vessels (LPG tankers) and vessels for carriage of liquid cargoes heated to very high temperatures (e.g. asphalt carriers).

Cargo tanks can be structural and non-structural. In the case of structural cargo tanks, the ship structure is separated by multi-layer insulation from the cargo. In the other case, non-structural cargo tanks are connected with the ship structure by special foundations which allow thermal dilatations of cargo tanks and insulate the ship structure from the tanks. This group is a very important group of tankers which require special design. As they are not characterized by high block coefficients and moderate speeds, they will not to be considered in this paper.

Chemical tankers are characterized by a large number of cargo holds and cargo segregations, high double bottoms and wider double sides, and in some cases, by the use of stainless steel for the construction of cargo tanks. As these tankers pose great danger to the environment due to the nature of their cargo, there are numerous rules and regulations pertaining to their design and building. They have some common characteristics with oil product carriers, so design models of such tankers can be applied to chemical tankers, provided some necessary changes are made.

The fourth and dominant group are crude oil tankers and oil product tankers having the common feature of high capacity cargo tanks. Crude oil tankers have slightly smaller capacity of their cargo tanks (the density of the cargo at the maximum draught is about 0.9 t/m^3). Product tankers are designed to have larger relative volume of cargo tanks (the usual density of the cargo at the maximum draught is approximately 0.8 t/m^3).

Crude oil tankers are vessels of larger sizes (from the „panamax" size upwards), usually with three cargo segregations and cargo pumps driven by steam turbines. Cargo tank bulkheads are usually of plane type.

Product tankers are vessels of smaller dimensions (usually up to the panamax or postpanamax dimensions), with a larger number of segregations and the cargo piping system with pumps driven by steam turbines or with deep-well pumps (driven by either hydraulic or electric motors). Corrugated bulkheads are often used in cargo tanks, and in some cases the deck framing is constructed above the deck. Thus, extreme cleanliness of cargo tanks is obtained, but also the right solution for the ship structure is made more difficult to find.

The usual configuration of tankers comprises a double bottom, double skin and a centreline longitudinal bulkhead. The largest tankers, i.e. VLCCs, have two centreline longitudinal bulkheads. The minimum double bottom height and the double skin thickness are determined by international regulations. By satisfying these regulations, sufficient capacity of ballast tanks is obtained and thus the MARPOL requirement of minimum draught is met in almost all cases. Only the largest tankers of suezmax and VLCC sizes have double bottoms and double skins with dimensions exceeding the required minimum. Typical cross-sections of an oil tanker and an oil product tanker are given in the following figures.

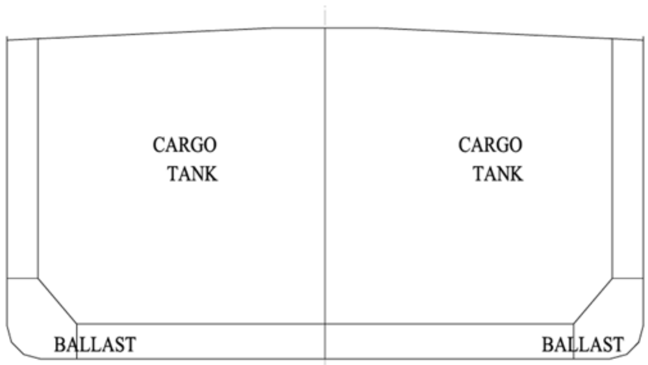
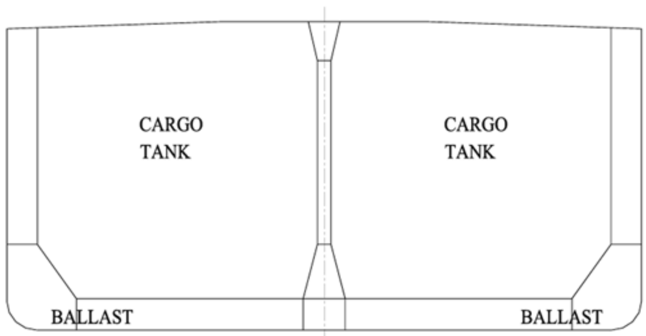


Figure 3 **Typical cross-section of a crude oil tanker**
Slika 3 **Tipični poprečni presjek tankera za prijevoz sirove nafte**

Figure 4 **Typical cross-section of a product tanker**
Slika 4 **Tipični poprečni presjek tankera za prijevoz naftnih derivata**



Basic conditions influencing the design process of tankers are as follows:

a) standard size with dominant groups:
    - Handy size group – tankers of 45,000 – 50,000 dwt, generally with the L_{oa} of up to 600 ft (182.88 m), the beam limited by the ability to pass through the Panama Canal and the maximum draught of up to 40 ft (12.2 m);
    - Panamax size group – tankers with the ability to pass through the Panama Canal, and in most cases with the L_{oa} limited to 750 ft (228.6 m);
    - Aframax size group - tankers of approximately 110,000 dwt at the maximum draught, and with the design draught, in most cases, of 40 ft (12.2 m);
    - Suezmax size group – tankers of 150,000 – 170,000 dwt, (named after the Suez Canal limitations which were in force by mid-2001);
    - VLCC size group – very large crude oil tankers (of approximately 300,000 dwt);
b) high capacity of cargo tanks;
c) general configuration with a double bottom, one or two longitudinal bulkheads, five or more pairs of cargo tanks, a pair of slop tanks and a pump room (for the cargo and ballast, or only for ballast);
d) the speed in most cases from 15.5 to16 knots in the trial sailing conditions and at the design draught;
e) the use of high tensile steel.

5 International Regulations and Requirements of Classification Societies

A great number of regulations cover the area of ship design, construction and exploitation. This section will deal with the most important rules and regulations which affect the design of tankers and bulk carriers to a great degree. These rules and regulations may be classified as follows:

- rules and regulations imposed by the International Maritime Organization (IMO): the International Convention for the Safety of Life at Sea (SOLAS), the International Convention for the Prevention of Pollution from Ships (MARPOL) and the International Convention on Load Lines (ICLL);
- rules of classification societies for the building of ships (the new harmonized IACS Common Structural Rules for tankers and bulk carriers);
- rules for sailing through canals.

5.1 Rules and Regulations Formulated by the International Maritime Organization

The SOLAS Convention [1] specifies minimum standards for the construction, equipment and operation of ships, compatible with their safety. The part of the Convention dealing with rules for subdivision and stability is the most interesting part for the initial design phase (Chapter II-1 Construction - Structure, subdivision and stability, machinery and electrical installations). It refers, in the first place, to the probabilistic calculation of the ship stability in damaged condition (Part B-1 - Subdivision and damage stability of cargo ships).

As the probabilistic calculation is very complex, it will not be dealt with in detail in this paper. Basically, it sets a great number of calculations related to the damage stability of the ship in various conditions of flooding. The effect of each condition of flooding on the overall quality of the damaged ship stability is weighted by the degree of probability that such damage should occur. The basic requirements and definitions are presented in Appendix A1.

The MARPOL Convention [2] comprises a set of rules which deal with the prevention of operational pollution. Requirements and rules dealing with the parameters to be taken into consideration in the design of a tanker are grouped in two chapters dealing with the tanker geometry and stability: Chapter II - Requirements for control of operational pollution and Chapter III - Requirements for minimizing oil pollution from oil tankers due to side and bottom damages. Because of the fact that rules are written and set in order by their time of adoptance, their usage during the design procedure can be uncomfortable. That is the reason to expound the most important rules in the Appendix A2 in order of their appeareance in the design procedure.

The ICLL Convention (1966) with its amendments [3] gives a definition of the minimum freeboard calculation for all ship types except for warships, yachts, ships of the length less than 24 m, existing ships of less than 150 GT and fishing vessels.

As the effect of all influential factors (block coefficient, depth, freeboard and trunk deck, camber, sheer, dimensions of forecastle and poop, etc.) are considered in the calculation, it is not possible to describe the calculation in detail here. Attention will be focused only on the definition of ship types with respect to their assigned freeboard (Chapter III, Regulation 27). Ships are generally divided into two ship types:

- type "A" – ships designed to carry only liquid cargoes in bulk (tankers), having cargo tanks with only small access openings closed by watertight gasketed covers;
- type "B" – all other ships.

Due to their design characteristics, the survival of tankers after flooding is of better quality than it is the case with other vessels. This is the reason why the minimum required freeboard is lower in type "A" vessels than that in type "B". Type "B" vessels can be assigned a lower freeboard than the calculated one – the type "B" reduced freeboard (usually, there is a difference of up to 60% between type "B" and type "A" freeboards if all conditions of the ship survival in the conditions of flooding defined in the convention are met).

5.2 Rules of Classification Societies for Ship Construction

A large number of classification societies are authorized to work in maritime countries with a tradition in shipbuilding all over the world. Their primary functions are to lay down requirements for the ship construction, survey of ships during the processes of building and exploitation, as well as to improve the level of ship quality and safety by developing the engineering, technological and scientific knowledge which can be applied to shipbuilding and shipping industry.

The most prominent classification societies are members of the International Association of Classification Societies (IACS). The purpose of such an association is to share experience and data, to develop better rules for ship construction and to adjust and unify rules of all the members. IACS developed new, uniform rules for the construction of particular ship types, e.g. uniform rules for the construction of bulk carriers.

For the same purpose, the biggest classification societies in the world (*Lloyd's Register of Shipping, American Bureau of Shipping* and *Det Norske Veritas*) have coordinated their joint efforts in issuing new, common rules for the construction of tankers, and Bureau Veritas and some other classification societies have done the same for the construction of bulk carriers (*Croatian Register of Shipping* developed new set of rules and programme CREST). New rules came into force in mid-2006.

In the ship design phase, the choice of a classification society is not of vital importance for the design model. Experience can lead to a conclusion on the influence of a classification society on the own mass of a particular ship type and size, but this influence can almost be neglected. Rules of classification societies have a more considerable influence on the ship design through their requirements regarding the general configuration of the ship. Special attention should be paid to the requirements presented in Appendix A3.

5.3 Regulations for Sailing Through Canals

There are a great number of canals and sea and river passageways where only vessels of limited dimensions can sail. Only three most important canals and their restrictions regarding sailing will be briefly dealt with here: St. Lawrence Seaway, the Panama Canal and the Suez Canal.

5.3.1 St. Lawrence Seaway

Rules for sailing are published in [4]. In ship design, the following rules and restrictions have to be taken into consideration:
- maximum length overall - 222.5 m;
- extreme breadth - 23.8 m;
- maximum draught - 7.92 m;
- maximum air draught - 35.5 m.

5.3.2 Panama Canal

Rules for sailing are defined in [5]. Restrictions and requirement to be met by tankers and bulk carriers are as follows:
- maximum length overall - 289.6 m;
- extreme breadth - 32.31 m;
- maximum draught - 12.04 m, provided that the minimum bilge radius is 1.79 m (in tropical fresh water with a density of 0.9954 kg/m^3);
- maximum air draught - 57.91 m;
- minimum draughts in sea water are defined as follows:

Table 1 **Panama Canal minimum draughts requirements**
Tablica 1 **Ograničenja izmjera broda za prolaz Panamskim kanalom**

for the ship's length exceeding (m)	draught forward (m)	draught aft (m)
129.54	2.44	4.30
144.80	5.50	6.10
160.02	6.10	6.71
176.80	6.71	7.32
190.50	7.32	7.93

The minimum draught requirement for passing through the Panama Canal is important because it is stricter than the previously stated MARPOL requirement, thus making it a major parameter in determining the minimum capacity of water ballast tanks. In the case of bulk carriers, the problem is solved by loading the ballast into a cargo tank intended for that purpose.

5.3.3 Suez Canal

Rules for sailing through the Suez Canal are published in [6]. Vessels with the breadth of up to 49.98 m (164 ft) may sail through the canal at the draught of up to 18.89 m (62 ft). Vessels with the breadth exceeding 49.98 m have the maximum draught defined in the table where the ratios between the ship's breadth and draught are given. The following table is taken from the rules.

Table 2 **Ship dimensions for passing through the Suez Canal (excerpt)**
Tablica 2 **Ograničenja izmjera broda za prolaz Sueskim kanalom (izvaci)**

Breadth (m)	Draught (m)	Breadth (m)	Draught (m)	Breadth (m)	Draught (m)
49.98	18.89	56.33	16.76	64.46	14.65
50.80	18.59	57.37	16.46	65.83	14.32
51.66	18.28	58.47	16.15	67.38	14.02
52.52	17.98	59.58	15.85	68.88	13.72
53.44	17.68	60.75	15.54	70.43	13.41
54.38	17.37	61.97	15.24	75.59	12.50
54.34	17.07	63.24	14.93	77.49	12.19

The product of breadths given in the table above and the appropriate draughts gives a constant value of approximately 944.5 m^2, which shows that the limiting value for the passing through the canal is the area of the cross-section of the ship.

One can conclude from Table 2 that all ships of all sizes, except VLCCs, can freely pass through the Suez Canal. Modern VLCC tankers usually have the deadweight of 300,000 tons, the breadth of approximately 60 metres, and the maximum draught is in the range of 20-22 metres. Their permissible draught for passing through the canal is approximately 15.7-15.8 metres, which means that they can pass through the canal with slightly more than 200,000 dwt.

6 Mathematical Models of Full Hull Form of Merchant Ship Design

Mathematical definition of the previously described design procedure is dealt with in [7, 8, 13, 15]. The mathematical model follows the steps of the procedure and, in the course of the process, defines the values required for obtaining final results.

Following the logic of the general design procedure, the mathematical model can be presented in the following way:

6.1 Definition of the Design Task

6.1.1 Design Variables and Parameters

Design variables and parameters are as follows:

a) Main dimensions:
- length between perpendiculars L_{pp} (m),
- breadth B (m),
- scantling draught d_s (m),
- block coefficient C_B (-);

b) Main engine identifier I_{ME},
c) Design tasks to be fulfilled within defined margins are:
  - deadweight DW (t),
  - capacity of cargo holds (tanks) V_{car} (m^3),
  - required trial speed v_{tr} (kn) (in most cases, defined for the trial sailing conditions at the design draught).
d) Specific voluminosity of the ship $\kappa = V_{car} / (L_{pp} B D)$ - depends primarily on the ship type and size. It provides the ratio of the "net used ship's volume", i.e. of the cargo space volume and the "maximum volume" determined by the product of three main dimensions. Ships with smaller engine rooms, ballast tanks and other under deck spaces have a higher specific voluminosity (that is why bulk carriers usually have higher voluminosity than tankers). The size of the ship also affects the value of this parameter (as a rule, a larger vessel has higher specific voluminosity). In addition, the value of this parameter is affected by the value of block coefficient.
e) The factor defining the influence of the high tensile steel use on the reduction of the steel structure mass is given as a percentage of the estimated reduction with respect to the ship structure completely built of mild steel. The maximum value of mass savings (when high strength steel is used to a high degree) is up to 15%.
f) Maximum power of particular main engines MCR_i that can be selected as the main engine. While selecting the main engine, special attention must be paid not only to maximum power which can be obtained, but also to the associated nominal revolutions and to the general configuration of the engine.
g) Data required for the calculation of costs of material comprise:
  - costs of feasible main engines C_{MEi},
  - average unit costs of steel c_{st},
  - other costs, comprising costs of other materials and equipment, C_{oc}.
h) Data required for the calculation of costs of labour:
  - shipyard productivity P_{cGT},
  - unit hourly wage V_L,
  - other costs C_{oc}.

6.1.2 Design constraints

Design constraints may be defined by minimum and maximum values of basic design variables or by maximum values of ratios between basic design variables.

a) Min-max values of basic design variables (main dimensions of the ship) are as follows:
  - min-max length between perpendiculars: $L_{pp\,min}$, $L_{pp\,max}$;
  - min-max breadth: B_{min}, B_{max};
  - min-max scantling draught: $d_{s\,min}$, $d_{s\,max}$;
  - min-max block coefficient: $C_{B\,min}$, $C_{B\,max}$.

Maximum values of main dimensions are most often limited by constraints of shipyard technological capabilities of building a ship, by rules and regulations of international legislation or by shipowner's requirements.

Minimum values of main dimensions are generally given empirically as the area bounds below which an acceptable design solution cannot be expected.

Minimum and maximum values of block coefficient are also, in most cases, empirical data. The minimum value of block coefficient is given as an empirical data below which an acceptable design solution cannot be expected, and it has no major importance in defining design constraints. The main problem is to determine the maximum value of block coefficient at a level which will not deteriorate the quality of optimum design solution, and which will enable a quality design of hull form.

Defining maximum values of block coefficient is a complex task which depends on several parameters: length/breadth ratio, breadth/draught ratio, fore body shape and fore bulb size, bilge radius, aft body shape, etc. All these ratios cannot be considered at the initial design stage, and only two dominant ratios, i.e. L_{pp}/B and B/d_s, are in the focus of the designer's attention.

The length/breadth ratio affects the maximum value of block coefficient in the way that higher values of this ratio enable higher values of block coefficient. This can be easily explained by the example of increase in the length of parallel middle body on the existing hull form: both L_{pp}/B and C_B increase.

The breadth/scantling draught ratio affects the block coefficient in the opposite way, i.e. the higher B/d_s, the lower is the achievable value of block coefficient. It can also be easily explained by the fact that C_B increases with deeper immersion of the ship (due to an increase in the waterplane coefficient); due to an increase in draught, the B/d_s ratio decreases.

Recommended maximum values of block coefficient presented in Figure 5 are based on the author's experience and on the latest generation of hull forms developed in *Brodosplit* [9,10,11,12]. It is also important to note that design solutions at the very maximum value of block coefficient should be avoided unless it is an imperative.

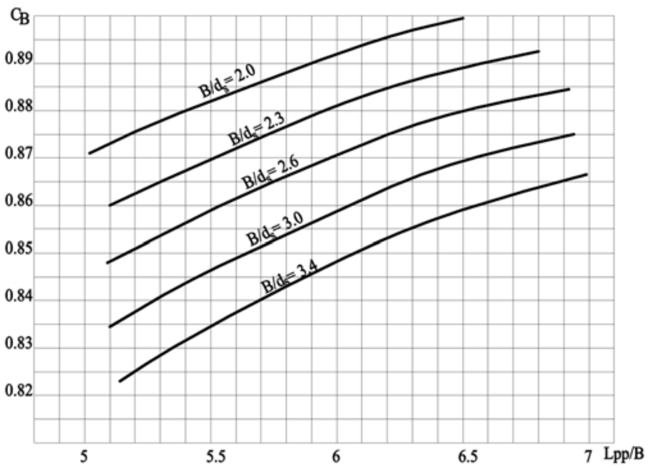


Figure 5 **Recommended maximum values of block coefficient**
Slika 5 **Preporučene maksimalne vrijednosti koeficijenta punoće**

b) Extreme values of ratios between basic design variables incorporate the following empirical or design constraints:
  - min-max length/breadth ratios: $(L_{pp}/B)_{min}$, $(L_{pp}/B)_{max}$;
  - min-max length/scantling draught ratios: $(L_{pp}/d_s)_{min}$, $(L_{pp}/d_s)_{max}$;
  - min-max breadth/scantling draught ratios: $(B/d_s)_{min}$, $(B/d_s)_{max}$;
  - min-max length/depth ratios: $(L_{pp}/D)_{min}$, $(L_{pp}/D)_{max}$.

Design constraints are based on the design experience. Recommended values of constraints vary depending on the ship size and type. They should usually be in the following ranges:

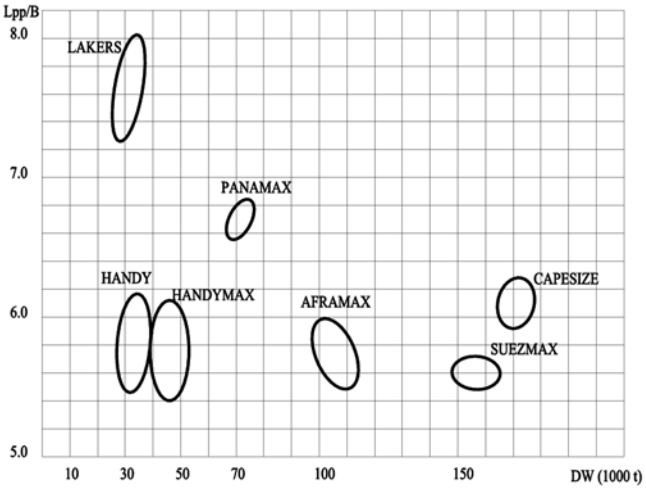


Figure 6 **Recommended constraints on the L_{pp}/B ratio**
Slika 6 **Preporučena ograničenja odnosa L_{pp}/B**

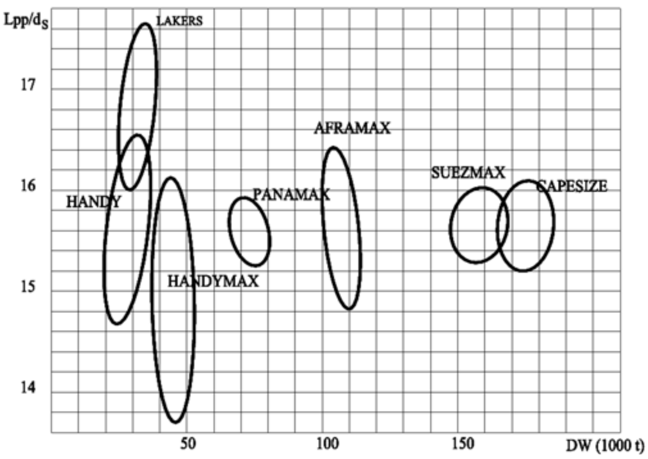
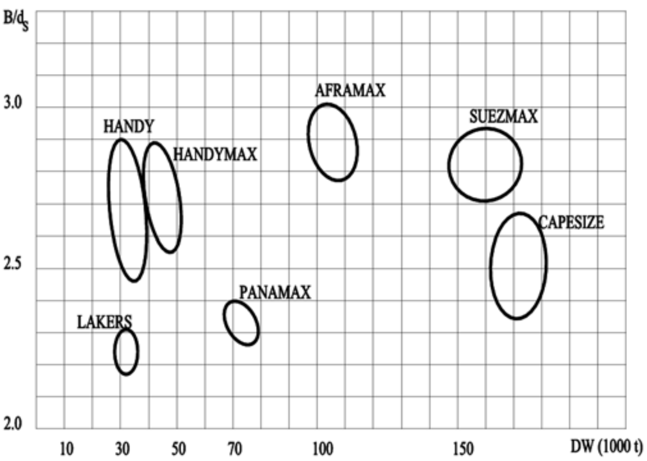


Figure 7 **Recommended constraints on the L_{pp}/d_s ratio**
Slika 7 **Preporučena ograničenja odnosa L_{pp}/d_s**

Figure 8 **Recommended constraints on the B/d_s ratio**
Slika 8 **Preporučena ograničenja odnosa B/d_s**



$$
\begin{aligned}
5.0 &\le (L_{pp}/B) \le 8.0 \\
14.0 &\le (L_{pp}/d_s) \le 18.0 \\
2.2 &\le (B/d_s) \le 3.3 \\
9.5 &\le (L_{pp}/D) \le 13.0
\end{aligned}
\tag{6.1}
$$

Recommendations for defining design constraints are given in Figures 6, 7, 8 and 9. These recommendations are based on some sixty designs made in the several past years in *Brodosplit* and should be used only as guidelines.

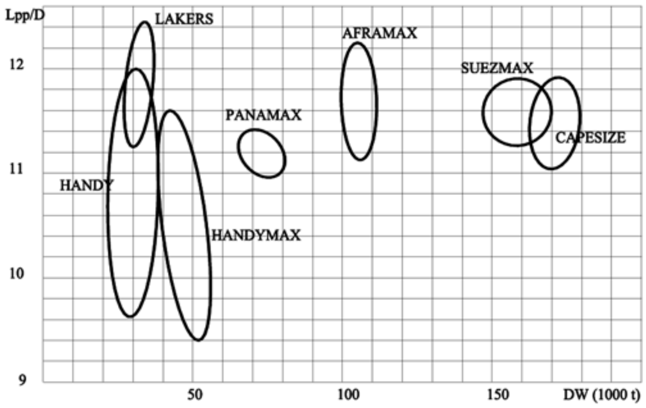


Figure 9 **Recommended constraints on the L_{pp}/D ratio**
Slika 9 **Preporučena ograničenja odnosa L_{pp}/D**

6.1.3 Dependent design properties (attributes)

Dependent design properties (attributes) described in the following sections are the properties whose values depend on input values (design variables and parameters).

a) Weight of the steel structure W_{st} (t) depends on the main dimensions, type and size of the ship. The steel structure weight is also affected by specific features of a particular design (size of the superstructure, ice class, forecastle, poop, etc.).
b) Cost of material (US $) depend on the total costs of steel, costs of the selected main engine, and on other costs related to materials.
c) Cost of labour (process) (US $) is calculated from the total volume of the ship, complexity of the ship, unit hourly wage and the shipyard productivity.
d) Cost of a ship (US $) is a sum of costs of material, costs of labour and other costs.
e) Obtained deadweight DW (t) depends on the ship's main dimensions and its light weight.
f) Obtained cargo space volume V_{car} (m^3) depends on main dimensions and a given "specific voluminosity" of the ship.
g) Obtained trial speed v_{tr} (kn) depends on the ship's main dimensions and propeller revolutions.

6.1.4 Design objectives

In the design of tankers and bulk carriers, possible design objectives can be defined:

a) Minimizing the weight of steel structure

The design objective of minimum weight of steel structure is particularly interesting in the light of a tendency to minimize the weight of the steel used (the criterion of minimum weight of light ship is very similar to that since the weight of steel structure in the total weight of the ship is a dominant element). Depending on the type and size of the ship, the share of steel may reach up to 30% of the total costs.

b) Minimizing the main engine power

The main engine is the most expensive item in the ship's equipment and its share in the total costs of a ship can be up to 15%. Hence, minimizing the main engine power is of great importance. Also, attention should be paid to the fact that the maximal power (and costs) of potential main engines rises steeply with each increase in the number of cylinders; the same applies to the type of the selected main engine. Therefore, this design objective is of major importance, and the targeted main engine should be used to its maximal power.

c) Minimizing the cost of material built into a ship

When minimal costs of material required to build a ship are concerned, there are two dominant values – costs of main engine and costs of steel. The costs of other material and ship's equipment embody a large number of small items which cannot be correlated with the basic characteristics of the ship at this design stage; therefore, the amount of these costs can be considered as a constant.

d) Minimizing the cost of labour (process)

In some cases it is of importance to minimize the costs of labour (process). This refers primarily to the situations when there is a shortage of skilled workforce at the market so that a possibility of optimizing the design towards this design objective has to be considered.

e) Minimizing the cost of newbuilding

For the shipyard, this is a dominant design objective. Although it is very important to meet all design requirements, minimizing the costs of newbuilding is of major importance for the shipbuilder. This results in the most favourable commercial effects for the contracted design and the total costs of a ship.

f) Minimizing the own mass of the ship

The design objective of minimum own mass of the ship is particularly interesting in the situation when the main dimensions of the ship are strongly limited. In these cases is possible to reach requested deadweight only in the way of minimizing the own mass of the ship.

g) Maximizing the stability

This objective is very important when ship is carrying significant amount of deck cargo.

h) Maximizing the speed

In some cases maximazing the ship's speed can be of the great interest for Shipyard and/or Shipowner. Maximazing the speed can also appear in the form of minimazing the ship's resistance (when the main engine is hardly reaching needed power).

6.2 Varying the Design Variables and Checking the Design Constraints

Main dimensions (length between perpendiculars L_{pp}, breadth B, scantling draught d_s, and block coefficient C_B) are varied between their minimum and maximum values in appropriate steps:

- step of length between perpendiculars $L_{pp\ step}$,
- step of breadth B_{step},
- step of scantling draught $d_{s\ step}$,
- step of block coefficient $C_{B\ step}$.

In determining the values of respective steps, due attention should be paid to the fact that their values can be technologically feasible in the shipyard, or on the other hand, that they are not too small.

6.3 Calculation of Depth and Minimum Freeboard

Calculation of the ship's depth for every combination of design variables, i.e. L_{pp}, B and V_{car}, and a given κ parameter is performed as follows:

$$D = V_{car} / (L_{pp}\ B\ \kappa)\ (\mathrm{m}) \tag{6.2}$$

Calculation of minimum freeboard is performed by a simplified calculation of minimum freeboard based on the actual combination of design variables (L_{pp}, B, d_s, C_B) and on predetermined values of other influential factors (forecastle, camber, sheer, etc.).

In this phase it is not possible to make an absolutely accurate calculation, but it is not necessary. During the phases of design development, it is always possible to correct the calculation to a certain degree.

After having checked the ship's depth in relation to the minimum required freeboard, the calculation with the actual combination of design variables is either continued or the combination is discarded.

6.4 Calculation of the Main Engine Minimum Power

A precise method for the approximation of continuous service rating (CSR) is used in [7, 8, 13, 15]. It will be briefly described in the following sections of the paper.

Approximation of power by the function of a given shape [16] is carried out on the basis of data for the main engine brake power and the ship's speed within the range of design constraints of main dimensions (length between perpendiculars L_{pp}, breadth B, scantling draught d_s and block coefficient C_B). Data base may contain results of serial model testing, results of a large number of trial sailings or results of available programs for the calculation of the form drag and the speed of ship.

The SEAKING program based on the ITTC recommendations and the SSPA correction factors has been used in [7,8,13,15]. The required power of main engine is calculated for a selected area of basic design variables, L_{pp}, B, d_s, C_B, and for the speed area around the required speed as well as for the predicted propeller revolutions. By regression analysis [16], independent parameters in the approximation function ($a_1 – a_n$) are determined and the mean deviation from the data base results is minimized. Different general forms of approximation function are possible.

The form used in [7, 8, 13, 15] will be used in this paper. Thus, the CSR is defined by the following approximation:

$$\mathrm{CSR} = a_1\ L_{pp}^{\ a2}\ B^{a3}\ d_s^{\ a4}\ C_B^{\ a5}\ v_{tr}^{\ a6}\ (1 + a_7\ L_{pp}/d_s)\ (\mathrm{kW}) \tag{6.3}$$

In the case when there is only one choice of the main engine type, the calculated power in relation to the maximum continuous service rating that a selected main engine can deliver is verified, and the design solution is either accepted as satisfactory, or is discarded.

If there is a choice between two or more main engines, the correction of the calculated power for predicted revolutions of every particular alternative main engine has to be carried out.

6.5 Calculation of the Ship's Displacement, Light Ship and Deadweight

Displacement Δ is defined as:

$$\Delta = L_{pp}\, B\, d_s\, \gamma_{tot}\ \text{(t)} \tag{6.4}$$

where γ_{tot} is defined as sea water density including the influence of ship's outside plating and appendages (t/m^3)

Deadweight is defined as a difference between displacement and light ship:

$$DW = \Delta - LS\ \text{(t)} \tag{6.5}$$

The light ship LS is defined as a sum of the steel structure weight W_{st}, the weight of machinery W_m and the weight of other equipment W_o, that is:

$$LS = W_{st} + W_m + W_o\ \text{(t)} \tag{6.6}$$

For the calculation of particular weights, there is a wide range of empirical data and formulae available in literature, e.g. [7,8,13,14,15]. Here, the following general forms of empirical formulae will be given:

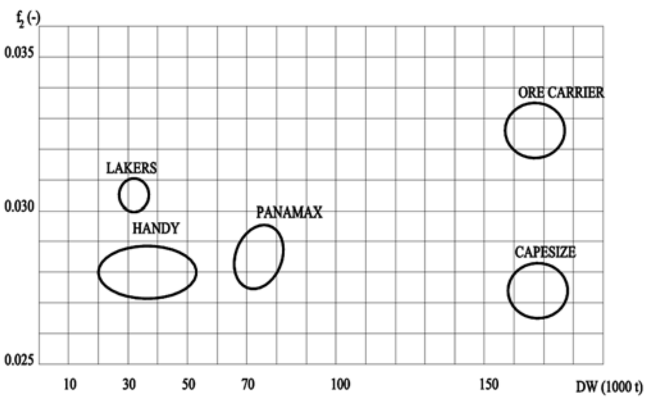
a) Steel structure weight

$$\begin{aligned} W_{st} = {} & (1 - f_1/100)\,(f_2\,[L_{pp}\,(B + 0.85\,D + 0.15\,d_s)]^{1.36} \\ & \{1 + 0.5\,[(C_B - 0.7) + \\ & + (1 - C_B)\,(0.8\,D - d_s)\,/\,3\,d_s]\} + f_3)\ \text{(t)} \end{aligned} \tag{6.7}$$

where:

- f_1 – factor of influence of high tensile steel on the reduction of steel structure weight
- f_2 – empirical factor presented in Figures 10 and 11
- f_3 – addition of the accomodation steel structure mass and specific features of a particular design (forecastle, ice class, etc.) (t)

Figure 10 **Factor f_2 (bulk carriers)**
Slika 10 **Faktor f_2 (brodovi za prijevoz rasutih tereta)**



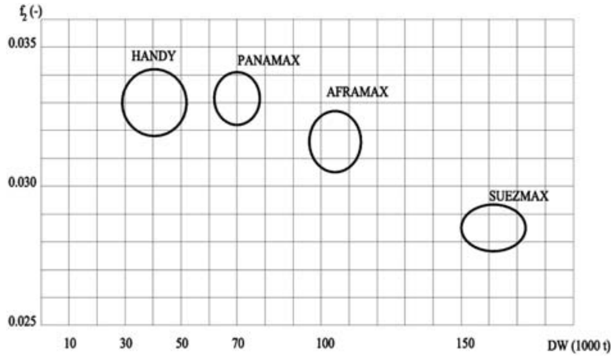


Figure 11 **Factor f_2 (tankers)**
Slika 11 **Faktor f_2 (tankeri)**

b) Weight of machinery

$$W_m = SMCR\ (f_4 - 0.0034\ SMCR)\ /\ 7350\ \text{(t)} \tag{6.8}$$

where:

- $SMCR = CSR / f_5$ - maximum selected power of main engine (kW)
- CSR - continuous service rating (kW)
- f_4 – empirical factor presented in Figures 12 and 13
- f_5 – CSR/ SMCR ratio, ranging from 0.85 to 0.9, depending on the optimization point of main engine

Figure 12 **Factor f_4 (bulk carriers)**
Slika 12 **Faktor f_4 (brodovi za prijevoz rasutih tereta)**

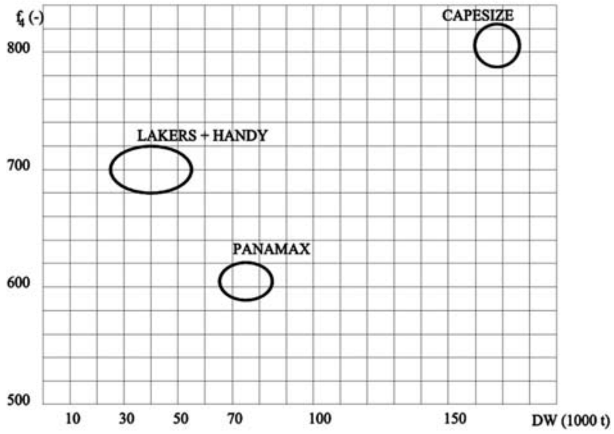
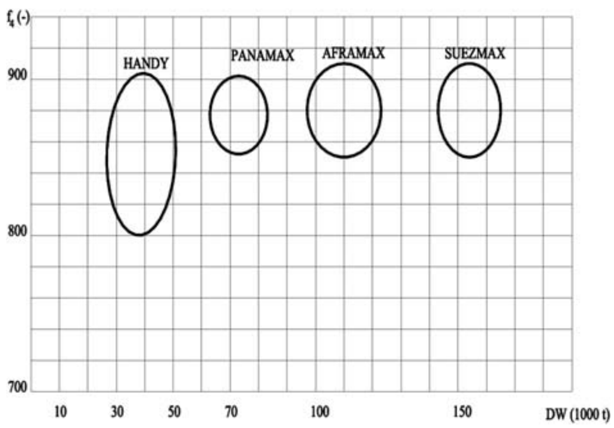


Figure 13 **Factor f_4 (tankers)**
Slika 13 **Faktor f_4 (tankeri)**



c) Weight of equipment

$$W_e = (f_6 - L_{pp} / 1620)\, L_{pp}\, B + f_7 \text{ (t)} \qquad (6.9)$$

where:

f_6 – empirical factor presented in Figures 14 and 15
f_7 – addition of the weight of ship equipment which is specific for a particular design (deck cranes, helicopter platform, etc.) (t)

Figure 14 **Factor f_6 (bulk carriers)**
Slika 14 **Faktor f_6 (brodovi za prijevoz rasutih tereta)**

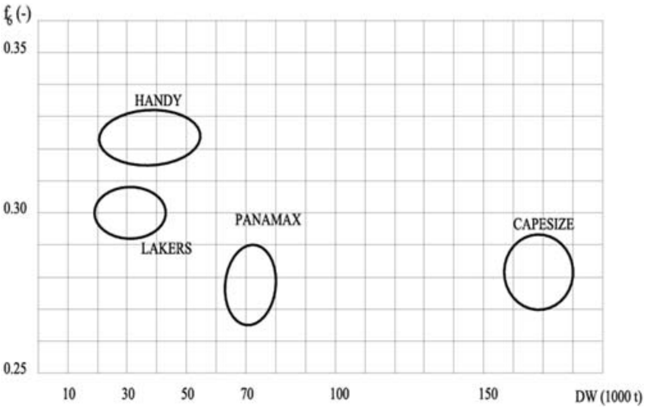
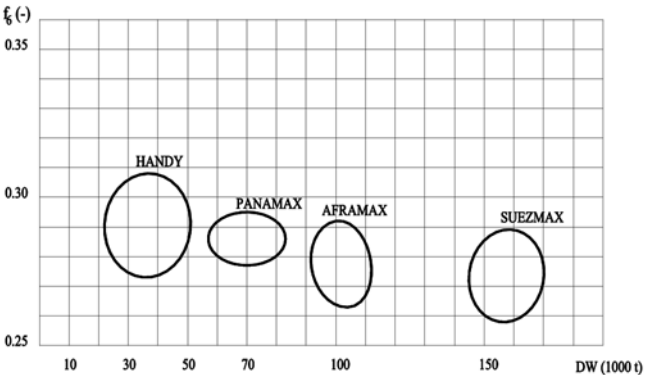


Figure 15 **Factor f_6 (tankers)**
Slika 15 **Faktor f_6 (tankeri)**



6.6 Calculation of costs of newbuilding

Costs of newbuilding C_{NB} comprise the costs of material C_M, costs of labour (process) C_L and other costs C_{oc}, i.e.:

$$C_{NB} = C_M + C_L + C_{oc} \text{ (US \$)} \qquad (6.10)$$

6.6.1 Calculation of costs of material

Costs of material C_M can be defined in the following way:

$$C_M = C_{ME} + C_{st} + C_{fix} \text{ (US \$)} \qquad (6.11)$$

where

C_{ME} – costs of main engine (US $)

$$C_{st} = W_{gst}\, c_{st} \text{ (US \$)} \qquad (6.12)$$

W_{gst} – gross weight of steel (required quantity of steel increased by 10-15% in relation to the weight of steel structure W_{st} because of scraps produced in material processing) (t)
c_{st} – average unit price of steel (US $/t)
C_{fix} – costs of other material and equipment (US $)

6.6.2 Costs of labour (process)

Costs of labour C_L can be calculated as follows:

$$C_L = cGT\, P_{cGT}\, V_L \text{ (US \$)} \qquad (6.13)$$

where:

P_{cGT} – productivity (working hours/cGT)
V_L – unit hourly wage (US $/working hour)
cGT – compensated gross tonnage, according to the OECD and defined as:

$$cGT = A * GT^B \qquad (6.14)$$

where:

GT – gross tonnage, defined as [17]:

$$GT = K_1\, V \qquad (6.15)$$

$$K_1 = 0.2 + 0.02 \log V \qquad (6.16)$$

V – total ship volume (m^3)

Factors A' and B' are defined by the following table 3.

Table 3 **Factors A' and B' (excerpt)**
Tablica 3 **Faktori A' i B' (izvaci)**

Ship type	A'	B'
Oil tankers (double hull)	48	0.57
Chemical tankers	84	0.55
Bulk carriers	29	0.61
Combined carriers	33	0.62

6.6.3 Other costs

These costs (costs of financing, docking, hiring tugs, model testing, external services, etc.) can be considered as fixed at this design stage and are given as a design parameter.

7 Conclusions

Design procedure and mathematical models published in this paper are basis for development of modern design tools based on multiattribute optimisation methods. Standard design procedure traditionally represented with so called "design spiral" is replaced with presented design procedure which enables application of modern optimisation methods and algorithms.

The published procedure can be universally applied to the design of bulk carriers, tankers and other ship types with similar basic characteristics. The advantage of the presented procedure over standard procedures (e.g. design using a design spiral) is that it can be applied and adaped to different methods used for carrying out the design procedure.

A further development of the design procedure can take place in two parallel directions: extending data bases of mathematical models for the design of particular ship types and sizes and extending data bases to include the exploitation life of a ship. The former direction leads to the preparation of Croatian shipyards to move on to building more complex ships. The latter direction leads to the research of the field which has not been adequately researched in the world shipbuilding and marine practice, i.e. to the design optimization not only from the point of view of the shipyard and the prospective customer, but also to the design optimization with respect to the ship's life – from contracting and building, to exploitation and final sale or laying up.

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Appendices

Appendix A1: Basic elements of damage stability probabilistic calculation (e.g. environmental pollution problems)

Required subdivision index (for ships longer than 80 m)

$$R = (0.002 + 0.0009\, L_s)^{1/3} \qquad \text{(A1.1)}$$

where L_s (subdivision length of the ship) is defined as the greatest projected moulded length of that part of a ship at or below deck, or as decks limiting the vertical extent of flooding with the ship at the deepest subdivision load line.

The attained subdivision index is

$$A = \Sigma\, p_i\, s_i \qquad \text{(A1.2)}$$

where

i – represents each compartment or group of compartments under consideration,

p_i – accounts for the probability that only the compartment or a group of compartments under consideration may be flooded, disregarding any horizontal subdivision,

$s_i = C\,[0.5\,(GZ_{max})\,(\text{range of stability})]^{1/2}$ - accounts for the probability of survival probability after flooding the compartment or a group of compartments under consideration, including the effects of any horizontal subdivision.

$C = 1$ if $\theta_e \leq 25^o$

$C = 0$ if $\theta_e > 30^o$

$C= [(30 - \theta_e) / 5]^{1/2}$ otherwise

GZ_{max} – maximum positive righting lever (m) within the stability range, but not greater than 0.1 m

θ_e – final equilibrium angle of heel (^{o})

The stability range is taken maximally up to the angle of heel of 20^o.

The attained subdivision index must be higher that the required one. If that is not the case, some interventions have to be made in the design, either by additional subdivisions, increased freeboard, rearrangement or heightening of hatch coamings or by using some other means.

Appendix A2: MARPOL rules of major importance in the design procedure

Minimum dimensions for the double side and double bottom are established in Chapter II, Regulation 13F. The minimum width (w) of the double side is defined in the following way:

$$w = 0.5 + DW / 20000 \text{ (m), or} \tag{A2.1}$$

w = 2.0 (m) , whichever is the lesser. The minimum value is 1.0 m

where DW (t) is deadweight.

Minimum height (h) of the double bottom is determined in the following way:

$$h = B / 15 \text{ (m), or} \tag{A2.2}$$

h = 2.0 (m), whichever is the lesser. The minimum value is 1.0 m

where B (m) is the moulded breadth of the ship.

Maximum dimensions of cargo tanks are defined in Chapter III. It will be briefly presented with in the following text.

Maximum length of a cargo tank is 10 m or any of the following values, whichever value is greater:

a) for tankers with no longitudinal bulkhead inside the cargo tanks

$(0.5\, b_i / B + 0.1)\, L$ but not to exceed 0.2 L (A2.3)

b) for tankers with a centreline longitudinal bulkhead inside the cargo tanks

$(0.25\, b_i / B + 0.15)\, L$ (A2.4)

c) for tankers with two or more longitudinal bulkheads inside the cargo tanks

(i) for wing cargo tanks: 0.2 L (A2.5)

(ii) for centre cargo tanks:

(1) if b_i / B is equal to or greater than one fifth: 0.2 L (A2.6)

(2) if b_i / B is less than one fifth:

- with no centreline bulkhead:

$(0.5\, b_i / B + 0.1)$ L (A2.7)

- with a centreline bulkhead:

$(0.25\, b_i / B + 0.15)$ L (A2.8)

where b_i is the minimum distance from the ship's side to the outer longitudinal bulkhead of the tank in question measured inboard at right angles to the centreline at the level corresponding to the assigned summer freeboard.

The length of a ship L (m) is defined as 96% of the total length on the waterline at 85% of the moulded depth, or as a distance from the stem to the axis of rudder stock on that waterline, whichever value is greater.

Maximum cargo tank capacity is defined in the way that a hypothetical oil outflow in the case of side damage of the ship O_c or the bottom damage of the ship O_s should not exceed 30,000 m^3 or 400 (DW)$^{1/3}$, whichever value is greater, but subject to a maximum of 40,000 m^3.

Basic calculations of a hypothetical cargo discharge in the case of ship damage are as follows:

(a) for side damages

$$O_c = \Sigma W_i + \Sigma K_i C_i \tag{A2.9}$$

(b) for bottom damages

$$O_s = {}^1/_3\,(\Sigma Z_i W_i + \Sigma Z_i C_i) \tag{A2.10}$$

where

W_i (m^3) = volume of a wing tank assumed to be breached by the damage

C_i (m^3) = volume of a centre tank assumed to be breached by the damage

$K_i = 1 - b_i / t_c$ when b_i is equal to or greater than t_c, K_i shall be taken as 0

$Z_i = 1 - h_i / v_s$ when h_i is equal to or greater than v_s, Z_i shall be taken as 0

b_i (m) = width of wing tank under consideration measured inboard from the ship's side at right angles to the centreline at the level corresponding to the assigned summer freeboard

h_i (m) = minimum depth of the double bottom under consideration

t_c, v_s are assumed damages defined in folowing text.

For the purpose of calculating hypothetical oil outflow following extent of damages are assumed:

(a) Side damage

(i) Longitudinal extent (l_c): ${}^1/_3\, L^{2/3}$ or 14.5 m, whichever is less (A2.11)

(ii) Transverse extent (t_c): B/5 ili 11.5 m, whichever is less (A2.12)

(iii) Vertical extent (v_c): from the baseline upwards without limit

(b) Bottom damage	From 0.3 L from the forward perpendicular	Any other part of the ship
(i) Longitudinal extent (l_s):	L/10	L/10 or 5 m, whichever is less (A2.13)

(ii) Transverse extent (t_s): B/6 or 10 m, whichever is less, but not less than 5 m — 5 m (A2.14)

(iii) Vertical extent from the baseline (v_s): B/15 or 6 m, whichever is less (A2.15)

Damage assumptions and stability criteria are established in Chapter III and Chapter II, Regulation 13F. It will be briefly presented with in the following text.

Damage stability criteria shall aply to:

- tankers of more than 225 m in length, anywhere in the ship's length
- tankers of more than 150 m, but not exceeding 225 m in length, anywhere in the ship's length, except involving either after or forward bulkhead bounding the machinery space located aft. The machinery space shall be treated as a single floodable compartment
- tankers not exceeding 150 m in length, anywhere in the ship's length between adjacent transverse bulkheads with the exception of the machinery space.

Damage cases:

(a) Side damage

(i) Longitudinal extent: $^1/_3\, L^{2/3}$ or 14.5 m, whichever is less; (A2.16)

(ii) Transverse extent: (inboard from the ship's side at right angles to the centreline at the level of the summer load line): B/5 or 11.5 m, whichever is less; (A2.17)

(iii) Vertical extent: from the moulded line of the bottom shell plating at centreline, upwards without limit

(b) Bottom damage

	For 0.3 L from the forward perpendicular	Any other part of the ship	
(i) Longitudinal extent:	$^1/_3\, L^{2/3}$ or 14.5 m, whichever is less	$^1/_3\, L^{2/3}$ or 5 m, whichever is less	(A2.18)
(ii) Transverse extent:	B/6 or 10 m, whichever is less	B/6 ili 5 m, whichever is less	(A2.19)

(iii) Vertical extent B/15 or 6 m, whichever is less, measured from the moulded line of the bottom shell plating at centreline (A2.20)

(c) Bottom raking damage (for oil tankers of 20,000 dwt and above)

(1) longitudinal extent

(i) for ships of 75,000 dwt and above: 0.6 L measured from the forward perpendicular (A2.21)

(ii) for ships of less than 75,000 dwt: 0.4 L measured from the forward perpendicular (A2.22)

(2) transverse extent: B/3 anywhere in the bottom (A2.23)

(3) vertical extent: breach of the outer hull. (A2.24)

Damage stability criteria are as follows:

(a) The final waterline, taking into account sinkage, heel and trim, shall be below the lower edge of any opening through which progressive flooding may take place.

(b) In the final stage of flooding, the angle of heel due to unsymmetrical flooding shall not exceed 25°, provided that this angle may be increased up to 30° if no deck edge immersion occurs.

(c) The stability in the final stage of flooding shall be investigated and may be regarded as sufficient if the righting lever curve has at least a range of 20° beyond the position of equilibrium in association with a maximum residual righting lever of at least 0.1 m within the 20° range; the area within this range shall not be less 0.0175 metre radian.

The requirement for minimum volume of ballast tanks is given in Chapter II, Regulation 13 by a definition of minimum ballast draughts. The minimum moulded amidships draught d_m is given as:

$$d_m = 2.0 + 0.02\, L \text{ (m)} \tag{A2.25}$$

in association with the maximum aft trim of 0.015 L and enabling full immersion of the propeller(s).

Appendix A3: Classification societies' rules having a influence on the general configuration of the ship

Further are presented DNV's requirements, other classification societies have similar requirements.

1) Minimum number of watertight transverse bulkheads

For ships without a longitudinal bulkhead and with the engine room located at the stern, the minimum number of bulkheads is defined by the following table A3.1.

Table A3.1 **Minimum number of transverse bulkheads**
Tablica A3.1 **Minimalni broj poprečnih pregrada**

Length of a ship (m)	Number of bulkheads
85 < L ≤ 105	4
105 < L ≤ 125	5
125 < L ≤ 145	6
145 < L ≤ 165	7
165 < L ≤ 190	8
190 < L ≤ 225	9
L > 225	considered individually

L (m) – length between perpendiculars (it should not be less than 96% or greater than 97% of the water line length at maximum draught).

The number of watertight transverse bulkheads may be lesser than the minimum number required. If that is case, the ship must satisfy the conditions of damage stability, and the problem of general configuration and strength of the ship should be given due attention.

2) Position of collision bulkhead

The position of collision bulkhead defines the length of the fore peak and the cargo space. It is defined as follows:

The distance from the forward perpendicular (x_c) must be within the values stated below:

$x_{c\,(minimum)} = 0.05\,L - x_r$ (m) for $L < 200$ m (A3.1)
$x_{c\,(minimum)} = 10 - x_r$ (m) for $L \geq 200$ m
$x_{c\,(maximum)} = 0.08\,L - x_r$ (m)

where

L – is the length of a ship defined according to ICLL, i.e. 96% of the length overall at 85% of the moulded depth of the ship, or the distance between the stem and the centre of the rudder shaft at the same waterline, whichever length is greater.
x_r – reduction due to bulbous bow, defined as
$x_r = 0$ for a bow without bulb
or, as the least value of the following values for a bulbous bow:
$x_r = 0.5\,x_b$ (m)
$x_r = 0.015$ L (m)
$x_r = 3.0$ (m)

where
x_b – is the length of the bulbous bow.

3) Minimum height of double bottom

The minimum height of double bottom is defined by the requirement for the height of the double bottom centre girder and brackets at the centreline of the ship. The minimum height is defined in the following way:

$$h_{min} = 250 + 20\,B + 50\,d_s \text{ (mm), minimum 650 mm} \quad \text{(A3.2)}$$

where

B – breadth of the ship (m)
d_s – scantling draught (m).

Nomenclature

A	attained subdivision index
b_i	minimum distance from the ship's side to the outer longitudinal bulkhead of the tank in question measured inboard at right angles to the centreline at the level corresponding to the assigned summer freeboard, m
B	maximum breadth of the ship, m
c_{st}	average unit price of steel, US $/t
cGT	compensated gross tonnage
C	consistency level
C_B	block coefficient
C_{BD}	block coefficient at the moulded depth
$C_{B\,0.85D}$	block coefficient at 85% of the moulded depth
C_{CB}	freeboard correction for the block coefficient
C_D	freeboard correction for the moulded depth, mm
C_i	volume of a centre tank assumed to be breached by the damage, m^3
C_{fix}	costs of other material and equipment, US $
C_{fc}	freeboard correction for forecastle, mm
C_L	cost of labour, US $
C_M	cost of material, US $
C_{ME}	cost of main engine, US $
C_{NB}	cost of newbuilding, US $
C_{sh}	freeboard correction for sheer, mm
C_{st}	cost of steel, US $
CSR	continuous service rating, kW
d_s	scantling draught, m
d_m	minimum ballast draught amidships, m
D	moulded depth of the, m
DW	deadweight, t
f_1	factor of influence of high tensile steel on the reduction of steel structure weight (%)
f_2	empirical factor presented in Figures 10 and 11
f_3	addition of the accommodation steel structure mass and specific features of a particular design, t
f_4	empirical factor presented in Figures 12 and 13
f_5	CSR/SMCR ratio
f_6	empirical factor presented in Figures 14 and 15
f_7	addition of the weight of ship equipment which is specific for a particular design, t
F_A	minimum freeboard for ships type A, mm
F_{B-60}	reduced minimum (B-60) freeboard, mm
F_{tA}	tabular freeboard for ships type A, mm
F_{tB}	tabular freeboard for ships type B, mm
F_{tB-60}	reduced minimum (B-60) tabular freeboard, mm
GT	gross tonnage
GZ_{max}	maximum positive righting lever, m
h	height of double bottom, m
I	unit matrix
I_{ME}	identificator of the main engine
IACS	International Association of Classification Societies
ICLL	International Convention on Load Lines
IMO	International Maritime Organization
ITTC	International Towing Tank Convention
l_c	longitudinal extent in the case of side damage, m
l_s	longitudinal extent in the case of bottom damage, m
L	length of the ship, m
L_F	length of the ship for the purpose of minimum freeboard calculation, m
L_{pp}	length between perpendiculars, m
LNG	liquefied natural gas
LPG	liquefied petroleum gas
LS	lightweight of the ship, t
MARPOL	International Convention for the Prevention of Pollution from Ships
MCR	maximum continuous rating, kW
NA	number of attributes
O_c	hypothetical cargo discharge in the case of side ship damage, m^3
O_s	hypothetical cargo discharge in the case of bottom ship damage, m^3
OECD	Organisation for Economic Co-operation and Development
p	importance vector
p_i	importance of attribute i
p_i	probability that only the compartment or a group of compartments under consideration may be flooded
P	preference matrix
P_{cGT}	productivity, working hour/f_cGT
P_{ij}	ratio of importance of attributes i and j
P_{oc}	other costs, US $
R	required subdivision index
s_i	probability of survival probability after flooding the compartment or a group of compartments under consideration
SMCR	selected maximum continuous rating, kW

SOLAS	International Convention for the Safety of Life at Sea
SSPA	Swedish hydrodynamics institute
t_c	transversal extent in the case of side damage, m
t_s	transversal extent in the case of bottom damage, m
U(y(x))	fuzzy function of attribute y
v_c	vertical extent in the case of side damage, m
v_s	vertical extent in the case of bottom damage, m
v_{tr}	trial speed, kn
V	total ship volume, m^3
V_{car}	capacity of cargo holds (tanks), m^3
V_{fc}	volume of the forecastle, m^3
V_{sup}	volume of the accommodation, hatch coamings and hatch covers, m^3
V_{cam}	volume of the camber, m^3
V_D	ship´s volume up to moulded depth, m^3
V_L	unit hourly wage, US $/working hour
VLCC	very large crude oil carrier
w	minimum double side width, m
W_{gst}	gross weight of steel, t
W_i	volume of a wing tank assumed to be breached by the damage, m^3
W_m	weight of machinery, t
W_e	weight of equipment, t
W_{st}	weight of steel structure, t
x_b	length of the bulbous bow, m
x_c	distance from the forward perpendicular, m
x_r	reduction due to bulbous bow, m
γ_{tot}	sea water density including the influence of ship plating and appendages, t/m^3
Δ	displacement, t
θ_e	final equilibrium angle of heel, °
κ	specific voluminosity of the ship
λ_i	eigenvalues of the problem

UDC 629.5.017.22:629.5.015.141

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Parametric Rolling at Main Resonance

Original scientific paper

The parametric resonance of the induced rolling is a typical dangerous situation for the transversal stability of ships operating in longitudinal waves. This paper presents the results of experimental tests regarding the seakeeping performances of a 2700 dwt cargo model at full loading, in regular longitudinal waves. The induced roll motion, in the second instability domain, both at zero and design speed was observed. The conditions for occurrence of the parametric resonance were analysed and the instability domains of motion were identified. The theoretical analysis of the heave and pitch motions in longitudinal waves, in frequency domain, was performed by using a computer code based on the Frank-close fit method. A satisfactory correlation between the theoretical and experimental results was found for heave and pitch motions. In order to simulate the induced rolling a typical differential coupled equations system for heave, induced roll and pitch motions was used. The numerical solution was obtained using the Runge-Kutta method. The simulation results for the ship motions at zero speed, in following regular waves, are presented. A good agreement was obtained between the numerical and experimental results in time domain.

Keywords: *experimental tests, parametric rolling, time domain simulation*

Parametersko ljuljanje pri glavnoj rezonanciji

Izvorni znanstveni rad

Parametarska rezonancija ljuljanja tipična je opasnost za poprečni stabilitet broda na uzdužnim valovima. Članak prikazuje rezultate pokusa ponašanja 2700 dwt modela teretnog broda pod punim opterećenjem na pravilnim uzdužnim valovima. Opažene su pojave induciranog ljuljanja u drugoj domeni nestabilnosti pri nultoj i projektnoj brzini. Analizirani su uvjeti za pojavu parametarske rezonancije te su određena područja nestabilnosti njihanja. Teorijska analiza u frekventnoj domeni poniranja i posrtanja na uzdužnim valovima je provedena na računalu primjenom Frankove metode prilagođavanja. Za poniranja i posrtanje teorijski i eksperimentalni rezultati su u suglasju. Za numeričko rješenje korišten je postupak Runge-Kutta. Prikazani su rezultati njihanja broda na harmonijskim valovima pri nultoj brzini. U vremenskoj se domeni rezultati pokusa i numerički rezultati dobro slažu.

Ključne riječi: *parametarsko ljuljanja, pokusi, simulacija u vremenskoj domeni*

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Received (Primljeno): 2008-04-29
Accepted (Prihvaćeno): 2008-06-08
Open for ciscussion (Otvoreno za raspravu): 2009-31-12

1 Introduction

When the ship runs in longitudinal waves, the representative ship motions are the heave and pitch ones. Due to the ship-waves interaction the transversal metacentre position has a dynamic modification in time domain, the excitation period being equal to the incident wave one [1]. As an effect of lateral perturbations, an induced roll motion may occur. The amplitude of the roll motion can increase, in parametric resonance conditions, according to the relation

$$\frac{T_e}{T_\phi} = \frac{n}{2} \tag{1}$$

where, T_e is the incident wave period and T_ϕ the roll natural period.

The roll parametric resonance is known as a typical dangerous situation for the transversal stability of ships operating in longitudinal waves.

The main causes of the induced roll motion are considered to be:

- the energetic "saturation" phenomena of heave and pitch motion [2];
- the nonlinear coupling of the heave or pitch motions with the induced roll one [3].

Nayfeh considers that the energy put into the pitch and heave motions by the wave excitations may be partially transferred into the roll motions by means of nonlinear coupling among these modes; consequently roll motion can be indirectly excited. The ship will spontaneously develop large amplitude roll motions at parametric resonance.

The first instability domain (n=1) is the most dangerous one for the ship's transverse stability, but large amplitude roll motions can occur in the main resonance domain, when n=2. The seakeeping tests in longitudinal waves of a cargo ship model [4] revealed the occurrence of parametric rolling, both at zero and design speed. The induced roll motion has occurred in a narrow frequency domain of the incident waves. The body plan of the cargo ship is shown in Figure 1. The main particulars and characteristics of the ship and the scaled model, at full loading condition, are listed in Table 1.

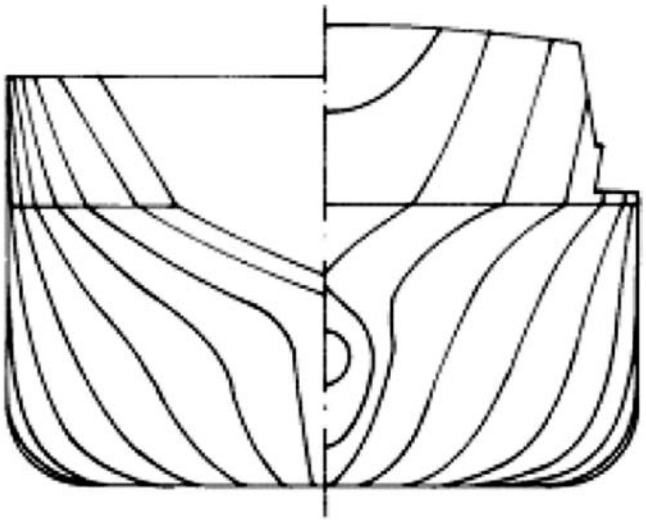


Figure 1 **The body plan of the cargo ship**
Slika 1 **Linije teretnog broda**

Table 1 **Main particulars at full loading condition**
Tablica 1 **Glavne izmjere za potpuno nakrcani brod**

Main characteristics	Full scale	Model scale (1/30)
Length over all, L_{max}	86.04 m	2.864 m
Length between perpendiculars, L	79.84 m	2.661 m
Breadth, B	14.5 m	0.483 m
Volumetric displacement, ∇	4297.0 m^3	0.159 m^3
Mean draught, T	5.245 m	0.175 m
Longitudinal centre of gravity, LCG	38.702 m	1.290 m
Vertical centre of gravity, KG	4.5 m	0.150 m
Metacentric height, GM_T	1.6 m	0.053 m
Natural period of roll motion, T_ϕ	7.55 s	1.38 s
Roll radius of gyration, k_{xx}	4.331 m	0.144 m
Pitch radius of gyration, k_{yy}	18.703 m	0.623 m
Yaw radius of gyration, k_{zz}	18.042 m	0.601 m
Ship speed, U	6.95 m/s	1.27 m/s
Froude number, F_n	0.25	0.25

2 Instability Domains of Induced Roll Motions

The uncoupled equation of the induced roll motion in longitudinal regular waves may be written using the Mathieu formulation [2]

$$(I_4 + A_{44})\ddot{\phi} + D(\phi,\dot{\phi}) + M_r(\phi,t) = 0 \tag{2}$$

where, I_4 is the inertia moment of roll motion, A_{44} is the added mass of roll motion, $D(\phi, \dot{\phi})$ is the roll damping moment, $M_r(\phi, t)$ is the restoring moment in longitudinal waves, ϕ is the roll angle and t is the time variable.

The dependency of the damping moment on the roll angle may be synthetically written as

$$D(\phi,\dot{\phi}) = \sum_{i,j} D_{ij}\phi^i\dot{\phi}^j,\ \text{(i, j=0,1,2,…)} \tag{3}$$

where D_{ij} represents the damping coefficients.

Within the limits of a cubic approximation, the restoring moment may be written as

$$M_r(\phi,t) = gm[GM(t)\phi + k_3\phi^3] \tag{4}$$

where, $GM(t)$ is the time-dependent metacentric height, m is the ship mass and g is the gravitational acceleration.

When the ship runs in longitudinal waves, the metacentric height has a time-dependent harmonic variation expressed as follows

$$GM(t) = GM_m - GM_a \cos\omega t \tag{5}$$

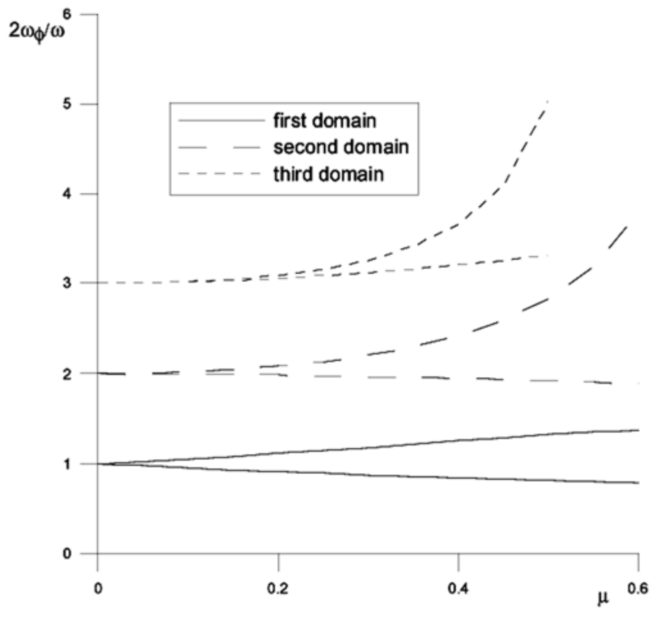
where, GM_m and GM_a are the mean value and the amplitude of the metacentric height and ω is the wave circular frequency. The excitation coefficient for the induced roll motion has the expression

$$\mu = 0.5GM_a / GM_m \tag{6}$$

The instability domains are corresponding to the critical frequency ones, where the ship may develop large amplitude roll motions.

A remarkable property of Mathieu's equation is that the solutions reach infinite values within instability domains. The frontiers of both stability and instability domains are defined by

Figure 2 **The instability domains of induced roll motion**
Slika 2 **Područje nestabilnosti induciranog ljuljanja**



T or 2T periodical solutions of the equation. By imposing Mathieu's equation to have 2T period solutions, the frontiers of the odd instability domain can be obtained. In the second instability domain, Mathieu's equation must have T period solution.

Figure 2 shows the diagram of the first three instability domains of the induced roll motion for the cargo ship. The vertical axis is the $2\omega_\phi/\omega$ ratio, where ω_ϕ, is the roll natural circular frequency and the horizontal axis is the excitation coefficient μ. One can observe that the first instability domain may lead to occurrence of the induced roll motion for small values of the excitation coefficient μ. The instability domains become larger for increased values of the excitation coefficient. The data from Figure 2 are obtained neglecting the damping moment contribution. If the influence of the damping term is considered, the area between the frontiers of the instability domains becomes smaller.

The excitation coefficient is not the only parameter that has an influence on induced roll motions. Another necessary condition may be mathematically written as follows

$$T_\phi = \frac{1}{n} \cdot \frac{4\pi}{\sqrt{\dfrac{2\pi g}{\lambda} - 2\pi \dfrac{U}{\lambda} \cos \alpha_i}}, \quad n \in N^* \tag{7}$$

Figure 3 **The necessary conditions for induced roll motions in the first instability domain**
Slika 3 **Potrebni uvjeti za induciranje ljuljanja u prvom području nestabilnosti**

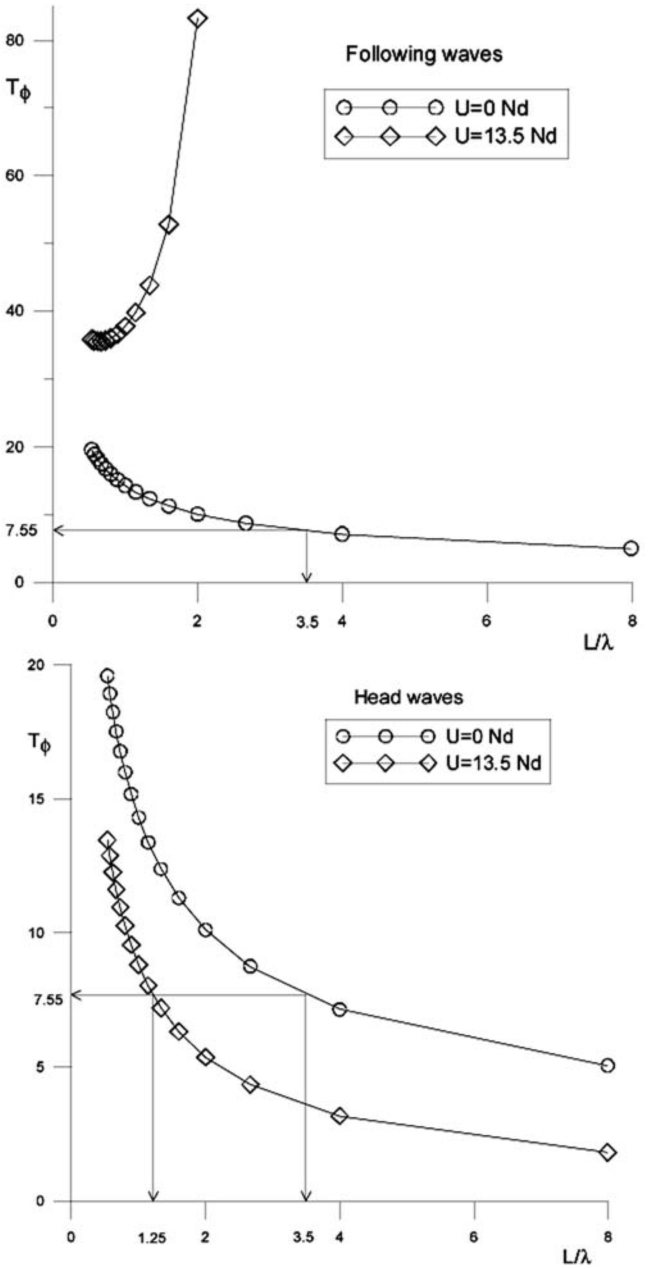
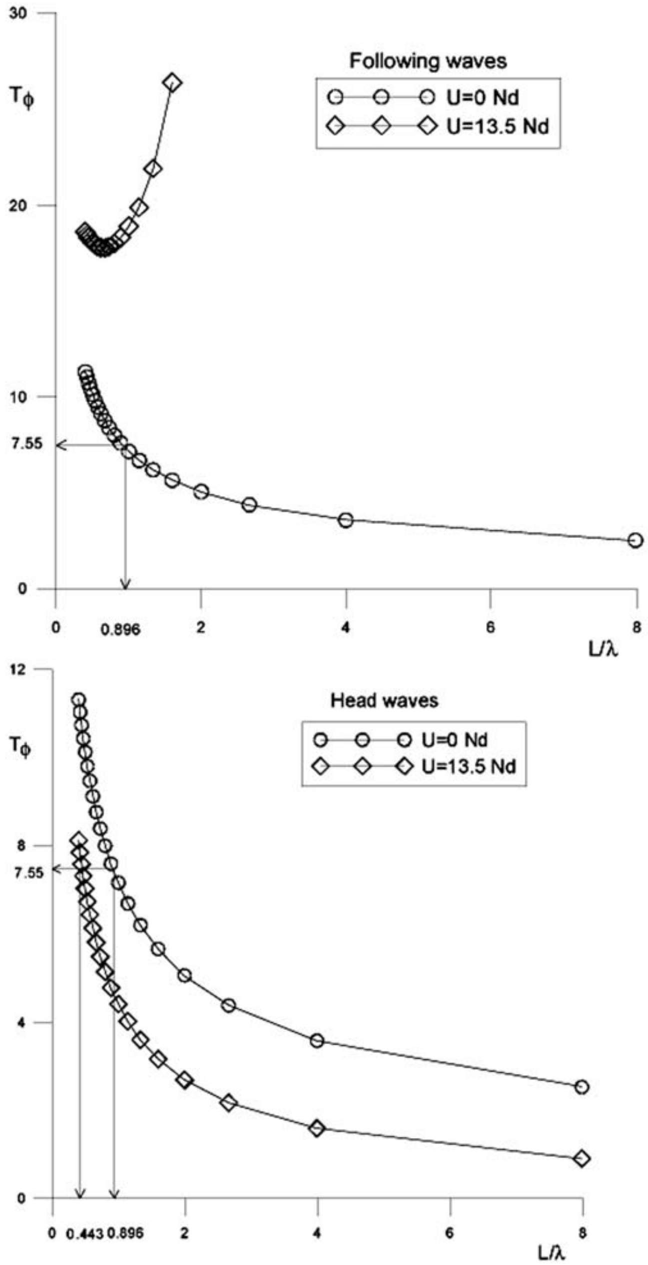


Figure 4 **The necessary conditions for induced roll motions in the second instability domain (main resonance)**
Slika 4 **Potrebni uvjeti za induciranje ljuljanja u drugom području nestabilnosti (glavna rezonancija)**



where, U is ship's speed and α_i is the heading angle. The graphical representation of the condition given by the equation (7) is shown in Figures 3 and 4. On the vertical axis the L/λ ratio is represented, where L is the ship length and λ is the length of the regular wave, and on the horizontal axis the natural roll period T_ϕ is represented. For the studied cargo ship, having the full scale natural roll period T_ϕ = 7.55 s, the probability of a roll motion occurrence in the second instability domain (main resonance) is increased. For the case at zero speed, the first instability domain corresponds to L/λ = 3.5 and the second instability domain to L/λ = 0.896. Regular waves, with small wavelength, do not have enough energy to generate induced roll motion. Consequently, at zero speed, parametric roll resonance in the first instability domain has low probability of occurrence. Moreover, due to the limitations of the wave generator, the experimental tests for f_m = 1.43 Hz (corresponding to L/λ = 3.5) could not be carried out. However, the induced roll motions at zero speed were measured, in the second instability domain, at main resonance, for both head and following waves respectively.

Figures 3 and 4 demonstrate that for both instability domains, the induced roll motion in regular following waves, at design speed, is not possible. When the ship runs at design speed in head waves, the probability of occurrence of roll motion in the second instability domain (L/λ = 0.443) is greater as compared to the first instability domain (L/λ = 1.25), where the amplitudes of motions of the ship model, on short waves, are very small. The main roll resonance for the ship running at design speed, in head waves, was observed and measured during the experimental tests. Comparing with the parametric resonance in the first instability domain, the induced roll motion at the main resonance for the ship in longitudinal waves was not intensively investigated. The literature offers little information on the presence of induced roll motion in the second instability domain [5]. From this point of view, a theoretical and experimental analysis of the main roll resonance represents a necessary extension of the studies on parametric roll resonance.

3 Theoretical and Experimental Determination of Ship Motions in Longitudinal Waves

The theoretical evaluation of ship motions in longitudinal waves has been obtained by using a computer code based on the Frank close-fit method [6]. Figures 5 and 6 show the transfer functions for heave and pitch motions respectively, in regular head waves, as a function of L/λ ratio. Figures 7 and 8 depict the transfer functions for heave and pitch motions respectively, in regular following waves, as a function of L/λ ratio. The non-dimensional formulations are obtained using z/ζ_w and $\theta/(k\zeta_w)$ ratios where, z and θ are the amplitudes of heave and pitch motions, ζ_w is the wave amplitude and k is the wave number.

The comparison between the theoretical and experimental results, for the heave and pitch motions, reveals a good agreement at zero speed. For the design speed the experimental transfer function are smaller than the theoretical predicted ones.

For the ship running in longitudinal waves, the computer code does not calculate a solution of induced roll motion. However, the experimental tests clearly demonstrate the existence of induced rolling at the main resonance. Figure 9 shows the experimental values of non-dimensional transfer functions, $\phi/(k\zeta_w)$ depending on L/λ ratio, for induced roll motions in longitudinal waves.

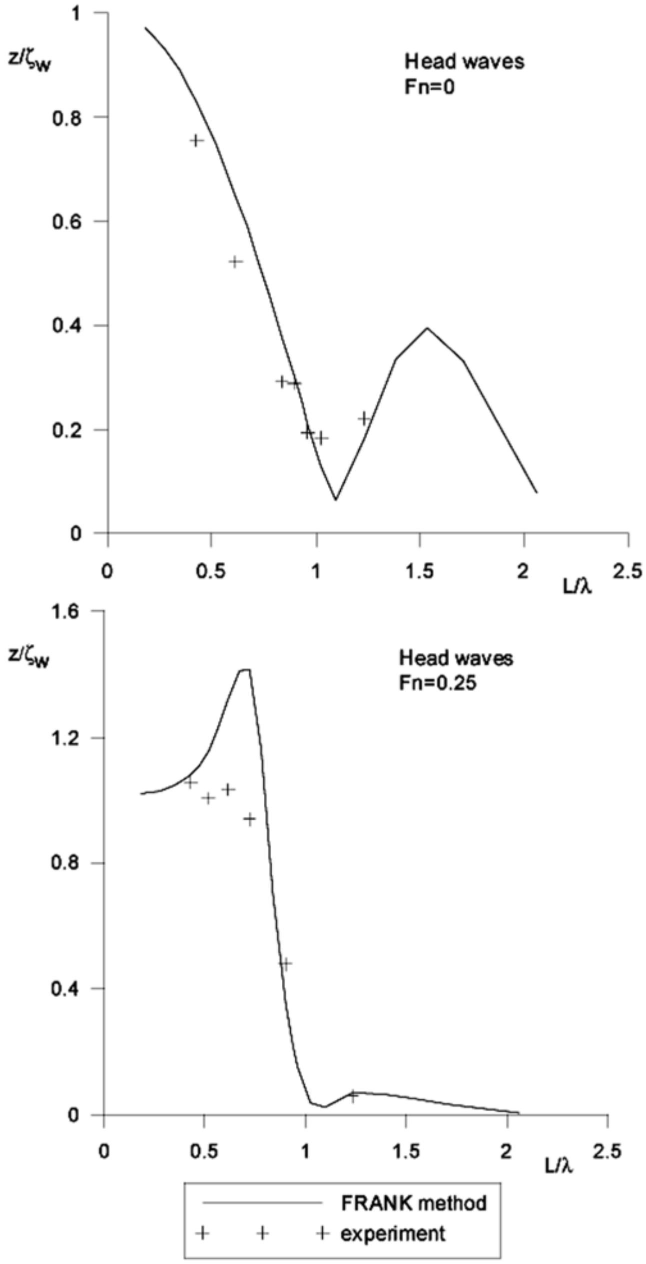


Figure 5 **Non-dimensional transfer functions of heave motion in regular head waves**
Slika 5 **Bezdimenzionalna prijenosna funkcija poniranja na pravilnim valovima u pramac**

The maximum amplitude is obtained at the main resonance, both at zero speed (L/λ = 0.896) and design speed (L/λ = 0.443). Figure 10 shows the experimental test arrangement in regular head waves.

The analysis of the experimental results given in Figure 9 leads to the following observations:

- the roll induced motions cover a small number of L/λ ratios, being a typical phenomena of narrow frequency domain;

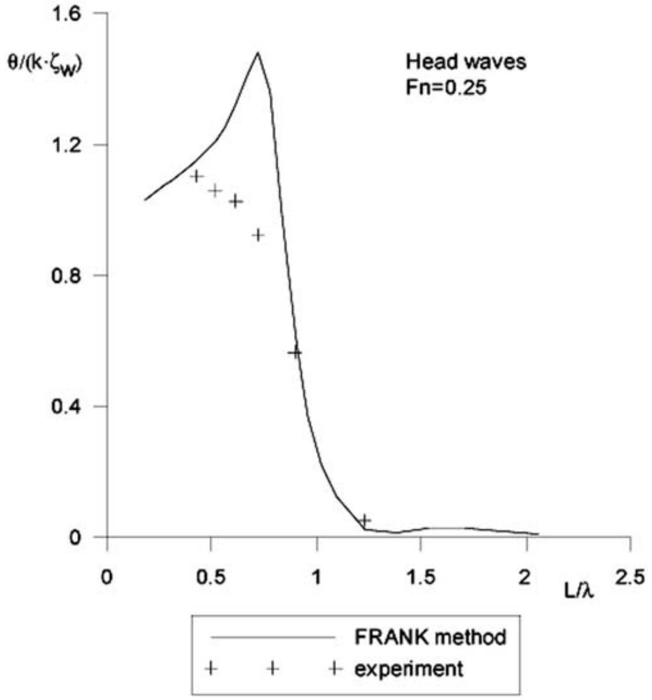


Figure 6 **Non-dimensional transfer functions of pitch motion in regular head waves**
Slika 6 **Bezdimenzionalna prijenosna funkcija posrtanja na pravilnim valovima u pramac**

- the amplitude of the induced roll motion at design speed is smaller than the amplitude at zero speed, in longitudinal waves;
- the model ship behaviour at zero speed in head waves, is similar to the following waves case;
- the coupled induced roll, heave and pitch motions are obtained in the second instability domain.

It is obvious that the development of the theoretical model must consider the induced rolling at the main resonance as being coupled with heave and pitch motions [7].

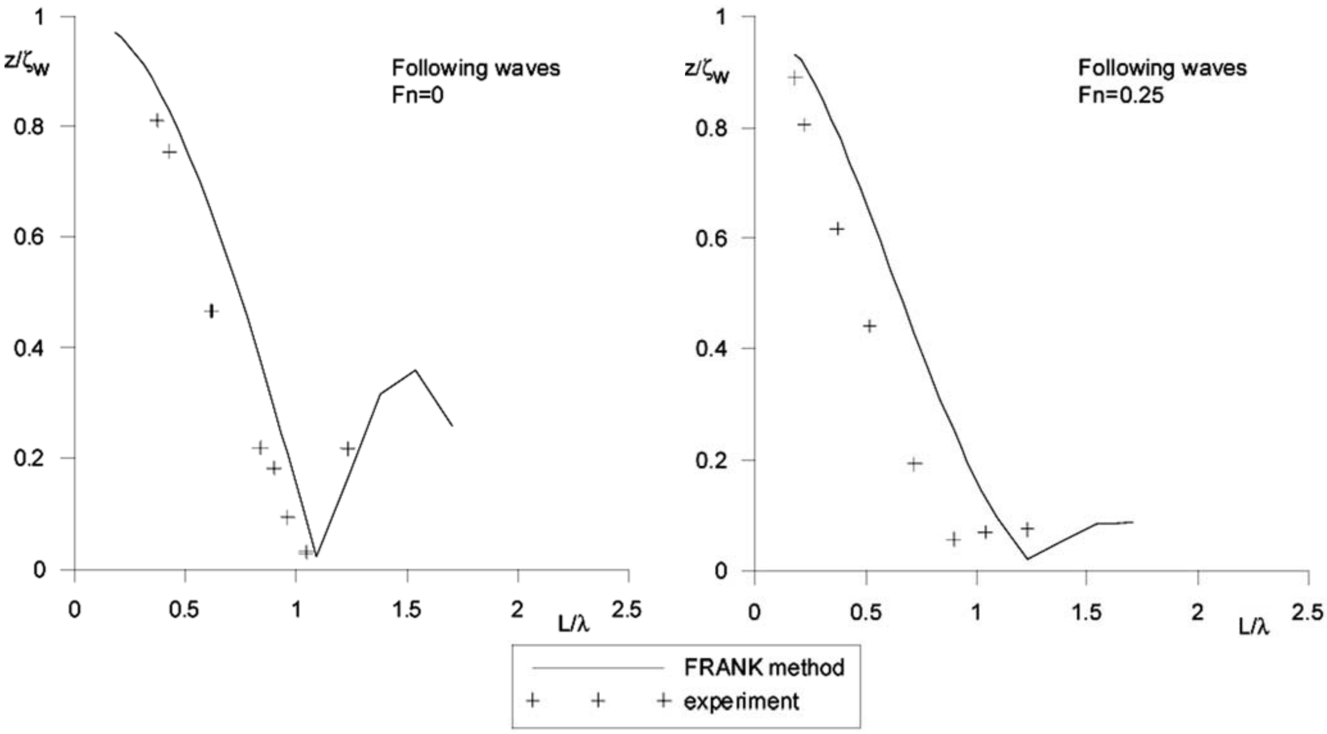
4 Numerical Modelling of Coupled Induced Roll, Heave and Pitch Motions

Considering the ship moving in regular longitudinal waves, heave (z), induced rolling (ϕ) and pitch (θ) motions were determined by numerical solving of the following coupled differential equation system

$$
\begin{aligned}
&(m+A_{33})\ddot{z}+A_{34}\cdot\ddot{\phi}+A_{35}\cdot\ddot{\theta}=q_1\\
&A_{34}\cdot\ddot{z}+(I_4+A_{44})\ddot{\phi}+A_{45}\cdot\ddot{\theta}=q_2\\
&A_{35}\cdot\ddot{z}+A_{45}\cdot\ddot{\phi}+(I_5+A_{55})\ddot{\theta}=q_3
\end{aligned}
\tag{8}
$$

$$
\begin{aligned}
&q_1=F_{3a}\cdot\cos\omega t-(B_{33}\cdot\dot{z}+C_{33}\cdot z+\\
&+B_{34}\cdot\dot{\varphi}+B_{35}\cdot\dot{\theta}+C_{35}\cdot\theta)\\
&q_2=F_{4a}\cdot\cos\omega t-[B_{34}\cdot\dot{z}+(D_{01}+\\
&+D_{02}\cdot|\dot{\varphi}|)\dot{\varphi}+g\cdot m(GM+\\
&+K_3\cdot\varphi^2)\varphi+B_{45}\cdot\dot{\theta}]\\
&q_3=F_{5a}\cdot\cos\omega t-(B_{35}\cdot\dot{z}+C_{35}\cdot z+\\
&+B_{45}\cdot\dot{\varphi}+B_{55}\cdot\dot{\theta}+C_{55}\cdot\theta)
\end{aligned}
\tag{9}
$$

Figure 7 **Non-dimensional transfer functions of heave motion in regular following waves**
Slika 7 **Bezdimenzionalna prijenosna funkcija poniranja na pravilnim valovima u krmu**



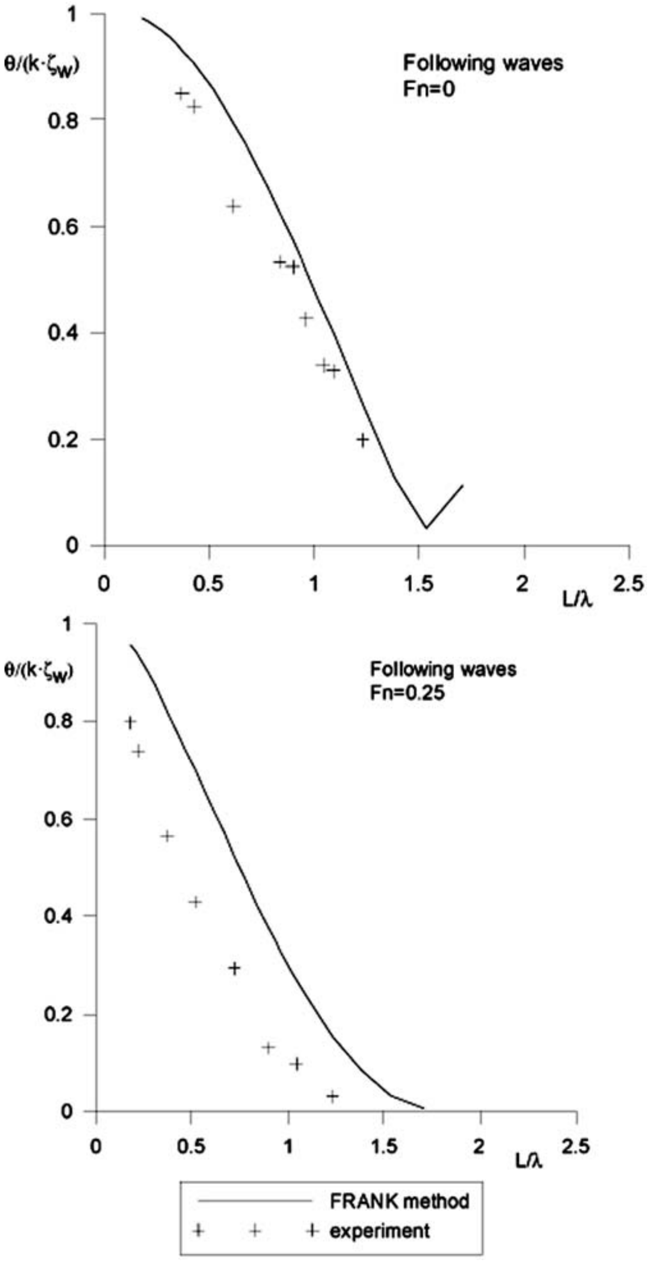


Figure 8 **Non-dimensional transfer functions of pitch motion in regular following waves**
Slika 8 **Bezdimenzionalna prijenosna funkcija posrtanja na pravilnim valovima u krmu**

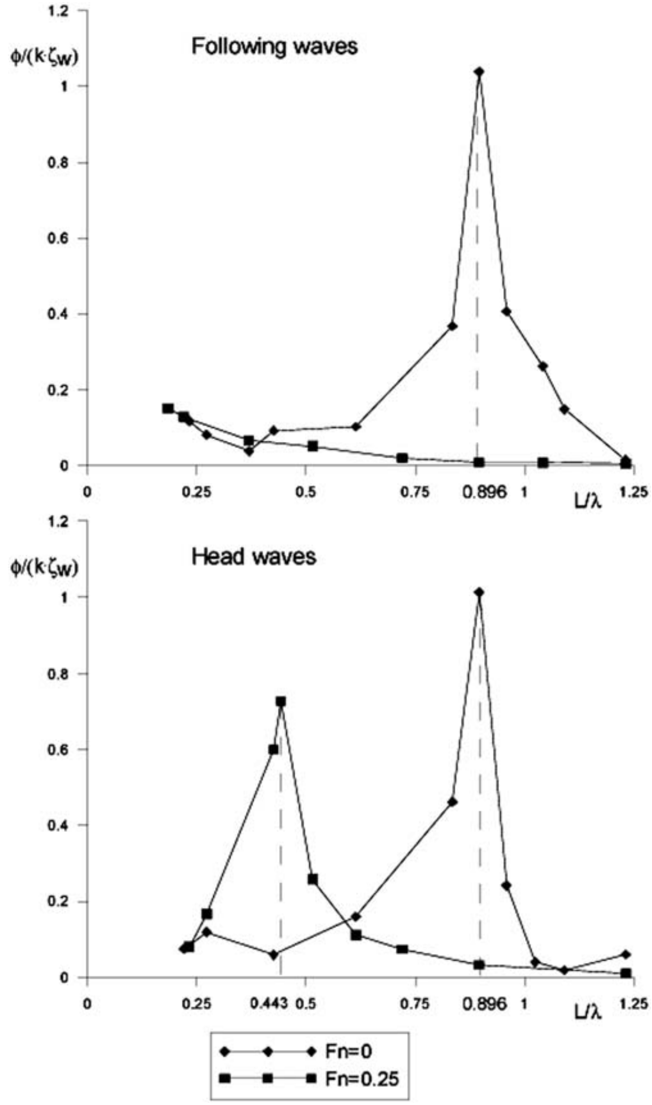
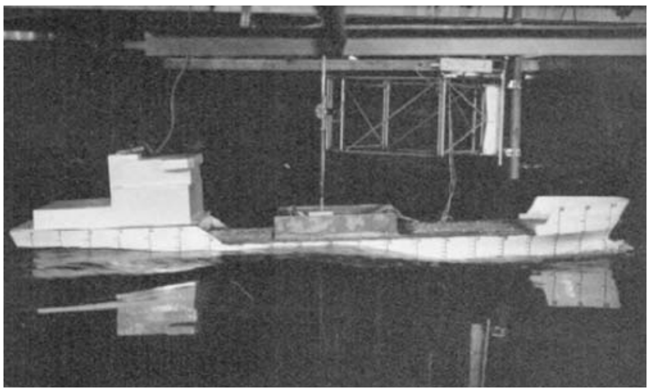


Figure 9 **Non-dimensional transfer functions of induced roll motion in regular longitudinal waves**
Slika 9 **Bezdimenzionalna prijenosna funkcija induciranog ljuljanja na pravilnim uzdužnim valovima**

Figure 10 **Experimental test in regular head waves**
Slika 10 **Pokusi na pravilnim valovima u pramac**



A_{ij} and B_{ij} (i, j = 3, 4, 5) represent the added masses and the damping coefficients, respectively, C_{ij} are the restoring coefficients and F_{ia} (i=3, 4, 5) represent the amplitudes of excitation forces and moments generated by the incident waves.

The evaluation of A_{ij} components is based on theoretical and experimental approach of the radiation problem [8], where the coupled coefficients A_{34}, D_{34}, A_{45}, B_{45} are considered as linear functions of small heeling angle.

Solving the diffraction problem, the amplitudes of F_{ia} force were theoretically and experimentally determined [8]. Moreover, the amplitude of the excitation moment at induced rolling is a linear function depending on the heeling angle

$$F_{4a} = f_{4a}\phi \tag{10}$$

The damping coefficients of induced roll motion in longitudinal waves, D_{01} and D_{02} were estimated based on experimental roll decay tests [4], analysed through energetic method proposed by J.B. Roberts [9].

The hydrostatic and hydrodynamic values of the cargo ship model are presented in Table 2.

Table 2 **The hydrostatic and hydrodynamic values of the cargo ship model**
Tablica 2 **Hidrostatički i hidrodinamički podaci za teretni brod**

Symbol	Ship model values (model scale 1:30)
A_{33}	170.97 kg
B_{33}	682.3 kg/s
A_{55}	64.582 kg·m^2
B_{55}	271.632 kg·m^2/s
A_{44}	0.71 kg·m^2
D_{01}	0.7848 kg·m^2/s
D_{02}	0.00592 kg·m^2
A_{34}	3.6325 · $\lvert\phi\rvert$ kg·m
B_{34}	-13.853 · $\lvert\phi\rvert$ kg·m/s
A_{45}	-0.2913 · $\lvert\phi\rvert$ kg·m^2
B_{45}	-2.142 · $\lvert\phi\rvert$ kg·m^2/s
A_{35}	-4.965 kg·m
B_{35}	-30.082 kg·m/s
C_{33}	10592.65 kg/s^2
C_{44}	85.122 kg·m^2/s^2
C_{55}	4690.35 kg·m^2/s^2
C_{35}	26.13 kg·m/s^2
F_{3a}	33.42 kg·m/s^2
F_{4a}	36.95 · ϕ kg·m^2/s^2
F_{5a}	62.16 kg·m^2/s^2

The numerical solutions of the differential equation (8) were found using the Gill version of the 4th order Runge-Kutta method. The calculations were performed at zero speed, for the case of induced roll motions in regular following waves, at the corresponding values of $L/\lambda = 0.896$ and $h_w/\lambda=1/50$, where h_w is the wave height. These conditions are specific for the parametric resonance of the induced roll motions in the second instability domain (main resonance). The initial conditions used to solve the differential equations took into consideration the zero heeling angle case. The experimental tests carried out under the above mentioned conditions allowed measurement of heave, induced roll and pitch motions, having the following amplitudes: $z = 0.006\ m$, $\phi = 3.8^0$ and $\theta = 1.8^0$.

In Figure 11 the numerical and experimental results of the behaviour of the ship model at zero speed, in regular following waves at the main resonance condition are exemplified. The correlation between the numerical and experimental results is satisfactory. As compared to heave and pitch motions, which are reaching stabilised solutions for a small number of oscillations, the induced roll motion has firstly a transitory phase and then the rolling amplitude slowly increases and becomes close to the experimental measured value. The induced rolling amplitude depends on the unit amplitude of roll excitation moment f_{4a}, thus the excitation coefficient of the induced roll motion can be defined as

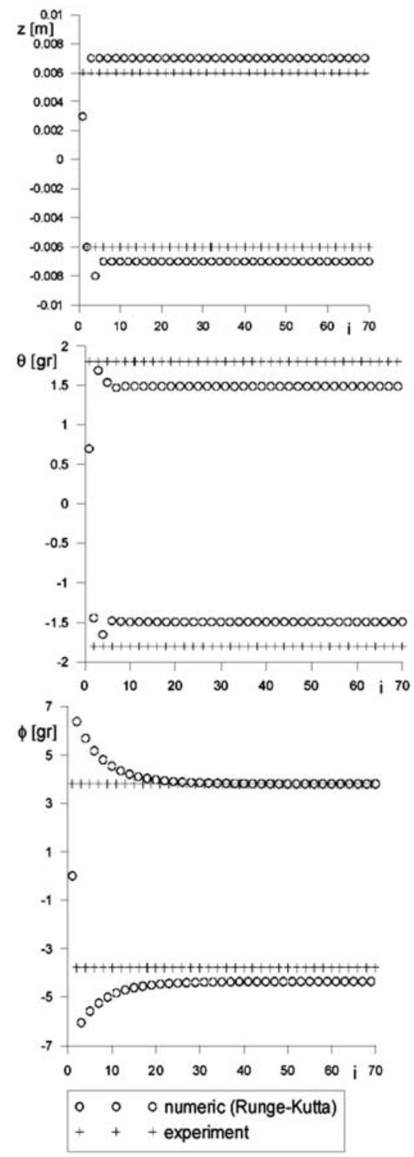
$$\mu = \frac{f_{4a}}{2 \cdot \omega_\phi^2 \cdot (I_4 + A_{44})} \tag{11}$$

where, ω_ϕ is the natural circular frequency.

5 Conclusions

The induced roll motion at parametric resonance represents a complex physical process. The occurrence of induced roll motions

Figure 11 **Numerical and experimental results of the behaviour of the ship model at zero speed, in regular following waves, at main resonance condition**
Slika 11 **Numerički i eksperimentalni rezultat ponašanja modela broda pri nultoj brzini na pravilnim valovima u uvjetima glavne rezonancije**



in the instability domains is a result of multiple interdependence of physical parameters such as: the energy partially transferred to the roll mode of motion, the excitation coefficient depending on the variation of the metacentric height on longitudinal waves, the natural roll period, the ship speed, the wave length, the heading angle, the encountering period, etc. Mention should be made that the diagrams given in Figures 3 and 4 suggest the correlation between parametric resonance condition (1) and the relation between the encountering period and the incident wave period. Based on the above mentioned diagrams and instability domains analysis, the physical conditions of the induced rolling may be identified.

The second instability domain was little analysed. This study was performed by coupling the differential equations of induced rolling with specific equations of heave and pitch motions. Except for the restoring coefficients and the coupled hydrodynamic coefficients, the remaining ones were experimentally measured during diffraction and radiation tests. The numerical solutions were experimentally validated.

Numerical tests were performed in order to identify the physical parameter having the main contribution on induced roll motion occurrence. It was demonstrated that the roll excitation moment had a decisive influence. In the considered case, a 10% reduction of roll excitation moment leads to the damping of induced roll motion. A 10% increment leads to a considerable amplification of ship's response.

The definition of the excitation coefficient, based on the experimental results of the diffraction problem, leads to a better understanding of the physical complex mechanism of the induced roll motions occurrence.

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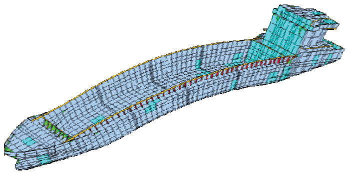

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Jasna PRPIĆ - ORŠIĆ

You should never say something about something that you don't know anything about

Interview with Professor O.M. Faltinsen, *Centre for Ships and Ocean Structures*, Trondheim, Norway

Recently I was invited to stay for two months in Norway at the *Centre for Ships and Ocean Structures* (*CeSOS*) which is located at the *Marine Technology Centre* (*MTC*) in Trondheim. The *MTC* also houses the *Department of Marine Technology* of the *University of Engineering Science and Technology*, as well as the *SINTEF* research institute *MARINTEK*. This location ensures a unique environment for researchers, with access to extensive laboratories, library facilities and other infrastructure.

Since its foundation in 2003, *CeSOS* has developed into a centre with about 80 full or part-time researchers with different educational and cultural backgrounds, half of them from outside Norway. In total, their work represents some 50 man-years annually. The main challenge is to balance the need to reach the goals with that of allowing researchers the freedom for creativity.

A key issue in *CeSOS* strategy is the interaction between researchers with not only different educational backgrounds but also different specialisations in hydrodynamics, structural mechanics and automatic control. Professors Odd M. Faltinsen, Thor I. Fossen and Torgeir Moan coordinate the work in hydrodynamics, automatic control and structural mechanics, respectively. Responsibility for the scientific content is to a large extent delegated to key persons, within the framework of their overall goals and plans. This also includes responsibility for publication in reputed journals and other media.

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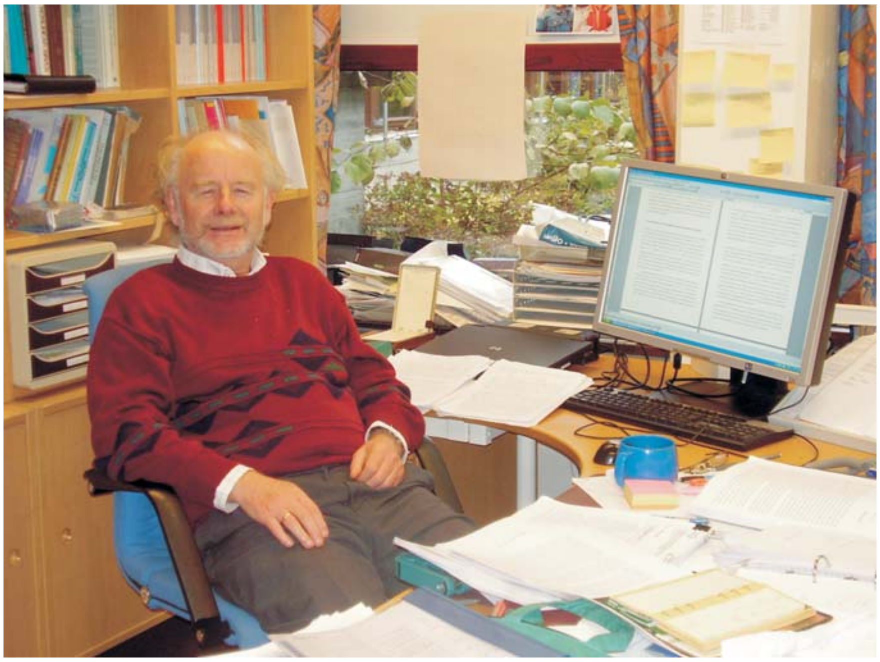

Professor O.M. Faltinsen in his office

The role of the director (Professor Torgeir Moan) and the other two discipline coordinators is to ensure quality of each project and also to ensure that it is in accordance with the Centre's overall plans. A simple organisational structure - based on the director and discipline heads as coordinators, and with key persons taking the main responsibility - gives the best balance between flexibility, goal-orientation and the final outcome.

CeSOS aims to create a daring, demanding and dynamic environment for research and development. At the same time, an important issue is safety, health and the working environment. This provides a framework to ensure physical and mental well-being and safety, especially in laboratory work, and the positive atmosphere of a successful organisation.

The Centre has succeeded beyond expectations in attracting funding in addition to that contributed by the *Research Council of Norway* and *Norwegian University of Science and Technology*, and in producing excellent scientific results.

Two Croatian researchers were also a part of *CeSOS* as PhD students during the past five years and recently got their PhD degrees there. Dr. Damir Radan got his PhD degree with the doctoral thesis *Energy Management of Marine Electrical Power Systems – Control of Integrated, Autonomous Power Systems*, under supervision of Professors Ådnanes, Sorensen and Johansen. Now he works at *Acergy Group* in Stavanger. Dr. Renato Skejić is currently working at *MARINTEK* in Trondheim. He obtained his PhD degree in June, under supervision of Professor Faltinsen. I had a pleasure to attend the successful defence of his doctoral thesis entitled *Manoeuvring and Seakeeping of a Single Ship and of Two Ships in Interaction*.

I also consider myself very lucky to have been able to get a chance to be a part of *CeSOS*, although for a brief period, and to work with Professor Odd Magnus Faltinsen. His work in the field of hydrodynamics of ships and sea structures is worldwide known. He is the author of some 300 scientific papers, two great books, he has been a supervisor to more than 40 PhD theses and with his achievements he has left an indelible trace in this field. He is a member of academies in Norway, USA and has been recently appointed academy member in China also. But what fascinates me most about Professor Faltinsen, besides his huge knowledge, are his remarkable human features. Despite of the planetary status he has retained his simplicity, modesty and kindness, and good humour all the time. For me working with him was a great privilege and pleasure and I carefully cherish the memory of each conversation we had. After I worked with Professor Faltinsen for a while, encouraged by my friends from the Editorial Board of *Brodogradnja* journal, I asked him for a permission to record part of our conversations in order to publish it as an interview afterwards, and he agreed. Here is the result of it …

BRODOGRADNJA : *You are one of the most famous world hydrodynamicists and since 1970 you have been giving continuously enormous contribution to marine hydrodynamics. But you have become worldwide famous very young by developing the strip theory with Professors Salvesen and Tuck. At that time Prof. Salvesen lived in America, Prof. Tuck was in Australia, and you were in Norway…*

FALTINSEN: It was not completely like that. It was like this: after I finished my master's degree in applied mathematics at Bergen I started working in the research department of in *Det Norske Veritas* (*DNV*). At that time Nils Salvesen had a leave of absence from his job at the *David Taylor Model Basin* and was then in *DNV* so we started to work together on the strip theory. Tuck had been already working on this, I believe at the *David Taylor Model Basin* with Nils Salvesen. He had been working on the formulation of the equation of motions; there was a theoretical report and computer program in which Frank (Werner Frank, author's comment) was also involved. What I did then at *DNV* was to generalize these equations by including loads, which is a prime interest for a classification society. Also, due to bad roll predictions I incorporated empirical formulae which were developed in Japan. Also, as a very natural thing to do I included so called end terms in the equations of motion.

BRODOGRADNJA : *You were very young then… It was your first paper?*
FALTINSEN: Yes, I was very young. In my master's thesis I was doing acoustics and then this was presented as a paper and maybe that was the first paper, but actually the paper about strip theory was my first paper in this field. We did a lot of investigations, which was part of the paper, in order to validate the method. Nils Salvesen did the major task in writing this paper. He is extremely good in formulating and also he was guiding me in general. You see, I came from applied mathematics and I had to learn those things. Well, how old was I? I believe … I have to think…

BRODOGRADNJA : *Twenty six?*
FALTINSEN: Yes when the paper was presented, but when I started at *DNV*, that was in 1968, I was then 24 years old and, well, you may say that it is just by chance that I went into this field. After I took my degree in applied mathematics I wanted to work with something which was more real world. I could work in quite different fields. Actually, I joined a very good group in the *DNV* research department where the leader of the group was Nordenstrøm. At *DNV* they know what they want, they want a product and they are supposed to do services, but still they did not require delivering it yesterday. They were patient relative to that, and generally speaking, I think it is an extremely important part of all applied oriented institutions that one should allow people, some people, not everyone, to work with a longer perspective, let's say at the order of a year or so. But I should be very clear, you should create a product, because that is the applied research of industrial companies, but I think that in our field you don't see that in many places. For me *DNV* had a very clear policy about having a research department. Nils Salvesen got me to go to the USA to take PhD degree, but afterwards, when I came back, I continued to work at *DNV* for a couple of years. Then I got a position here at the university and then also I was working on similar types of projects.

BRODOGRADNJA : *Today, the strip theory is still widely used. How do you explain that in spite of very fast accumulation of knowledge and rapid development of computers capabilities this theory still survives?*
FALTINSEN: You may say it like this: the more you know the more complicated you would like to be. I think that it was advantage that at that time I didn't know too much. There are several reasons why the strip theory is still widely in use according to my opinion. First of all the strip theory would not be good when the frequency of encounter is very small, but in that case the problem is quasi-static and the hydrodynamics is not so important. The other aspect of it, I believe, is that global seakeeping predictions are not so sensitive to the details of the hull form as for instance the wave resistance problem. So, I think it is partly luck that it works and that is one aspect. The other aspect is related to the fact that you can use linear theory for much higher wave conditions than you may think in reality. There has been a lot of effort related to having the Green function type of method in three dimensions, but it is extremely hard to calculate and it is very sensitive numerically. Even when one is able to correctly make these calculations you don't get that much difference in results and then you don't want to make that effort. I have been a lot involved with the ITTC (*International Towing Tank Conference*, author's comment), for three periods I was involved with the Seakeeping Committee and we

did a lot of the comparative studies and then it turned out like the following. You had what is called unified theory which was developed by Nick Newman and, for sure, that type of theory can predict more accurately and then in the broader frequency range, which are the added mass and damping coefficients. But you are not really interested in added mass and damping, you are interested in response. When you have a situation where you are in a resonant condition of heave and pitch, the relative motions are relatively large at head sea and with forward speed and in experimental studies you even may have green water, so it happened that the strip theory may give better results than the unified theory. What is really needed is a method that can predict really extreme type of situations where you have green water on deck and slamming incorporated in your calculations but it is extremely hard to do it. It is very demanding task to do. Also, we must not forget that it is extremely important to make simulations in a seaway because you want stochastic result, in a stochastic sea. You want to make long term prediction and so on. And then following straightforward with CFD (Computational fluid dynamics, author's comment) methods is out of the questions but of course you can be smart and then have different type of procedures. To conclude, I think it is partly luck and you don't have to cite me but I often say: *Some people believe in strip theory like some people believe in God*. It is a religious aspect related to it. Of course, we must not forget that this type of analysis is made for operational studies and is useful for structural people, as long as they can get some useful values of it. Of course, it is not perfect and you can generalize the strip theory to different problems like whipping or springing. It doesn't predict that kind of effects so accurately. The other issue is related to what you are working on - the added resistance. For example you have Gerritsma & Beukelman's formula which is very simple and you may want to make it more complicated if you want to, but it works as long as you are in the wavelength regime where you have significant ship motions. Also similarly I was working on low wave length type of theory. It is very simple and that is also working. So the success of the strip theory is a mixture of luck and a mixture of that you don't know too much. If I had known much more then, I would have made it much more complicated.

BRODOGRADNJA: *You have written two very famous books "Sea Loads on Ships and Offshore Structures" and "Hydrodynamics of High-Speed Marine Vehicles". These books are widely used in education of students worldwide. The first one is translated in Korean and Chinese and "Hydrodynamics of High-Speed Marine Vehicles is translated in Chinese".*

FALTINSEN: One of the reasons why these books have been translated into Chinese is also the fact that we have many Chinese students and we see that they have limited background on sea loads which is very important if you deal with offshore structures. The development of the first book is a long story in time starting from when I was employed by the university in 1974 and I had to develop lecture notes related to offshore. When I was at the *MIT* (*Massachusetts Institute of Technology*, author's comment) in 1987-1988, Nick Newman encouraged me to write a book as an editor of the Ocean Technology Series of *Cambridge University Press*. After I wrote that book I told to myself I should never more write books. One cannot imagine how it is because you must continuously control and control. You must create a lot of examples without errors and you must create exercises and so on. It is a full time job. So I said that to myself then. When *CeSOS* started in 2003, we defined different objectives. One is to publish in journals which have very high standard. But I said that one objective also should be that one should try to write books. When I said that then I must do it. Once more I had some material which I used as a part of seakeeping course I had been giving. That was a start, but I had to do a tremendous job because I created the title which was very demanding, and that was "Hydrodynamics of High Speed Vessels". I can't just talk about seakeeping. That means I must talk about all aspects, but I had a very important school for me - I had learned a lot by being a member of committees of the ITTC. I was the Chairman of the High Speed Committee so then I learned a lot about all aspects of it. At that time we had a project in Norway about high speed vessels and we started this FAST conference and there were a lot of materials which I have been used. Writing a book like that it means that I learned a lot myself because you must study all different aspects of it.

BRODOGRADNJA: *You are writing the third book about sloshing… can you tell me something about it.*

FALTINSEN: As you know, I am writing a new book and now I feel trapped, because first it was planned that there would be three authors, and now it turns out that there will be two authors. I didn't think that I should be involved so much, because originally there should have been more authors, but it takes tremendous time. I have been working together with Oleksandr Tymokha a lot about theoretical aspect of more analytically oriented methods. But, in my opinion, it becomes too narrow to talk just about that and then one has to make this scope broader and also make connections to other engineering fields where sloshing is important. One topic which we don't cover is space application. There has been done a lot related to space application, but the reason why we don't cover it is that it would imply situations where you don't have gravity and other physical effect matter. So we don't cover that and there is a very good book by Abramson which I think is very useful in that area. But once more, writing a book means you have to create lot of examples and you must create exercises. One important aspect is control but it must also be understandable. I understand this topic, but I am not writing a book for myself and I must test it out. I have many students and they are helping me in reading and solving the problems, controlling the errors that I am making. I make examples and there is always a bug. Therefore, I have a big network of people who are involved in the books, locally but also internationally. I have colleagues whom I am using in reviewing the book like for articles. All these books are published by *Cambridge University Press* which I have a very good experience with, also the last book which is supposed to be delivered by the end of this year. It is planned that this book will also be translated into Chinese. The *Jiaotong University* in Shanghai will be involved in this and they, of course very professionally, used a lot of students to translate the book but then they control on higher level with professors having detailed knowledge in the field. I can't control what they are writing, so top persons which are very good people have been involved in real quality control of the Chinese version.

BRODOGRADNJA: *You are a member of three Norwegian academies: Norwegian Academy for Technical Sciences, Norwegian Academy of Science and Let-*

ters and the Royal Norwegian Society of Sciences and Letters. You are also a foreign member of National Academy of United States of America and, recently, you are elected member of China Academy.

FALTINSEN: In Norway we have different kinds of academies. Only one is purely in technical sciences. I became a foreign associate of the National Academy of Engineering of the United States of America, Academy in USA in 1991. Then recently I have become a member of the Chinese Academy of Engineering and I was there at the end of June as you know. First of all, there are not a lot of foreign members, only 34 totally in all fields. The number of the Chinese academy members may be of the order of seven hundred let's say in all fields but foreign members are very few. In naval architecture and related fields there are only two international members, I believe. There is a very big difference in how the academy there is working compared to Norway. They have much more direct contact with the government and the members have a very high position in society. What also surprised me, when I was elected member, the Chinese Ambassador invited me with my wife for a dinner. She has been visiting me afterwards and she tells that being a member of that academy has very high status in China. When the ceremony was there the President of China and a large part of the government and important persons were present. For sure it is not the same in Norway, it is far from that, here it is a separate organization. I believe that in the USA they also have much more influence than in Norway. Although they are not part of the government, they help the government in some sense. It was, of course, a nice experience for me. Members of academy in China have so many privileges. But I don't have them they told me, but I don't want to have them either, that is so.

BRODOGRADNJA: *How does the Norwegian shipbuilding industry survive in these days and what kind of ships do they produce? I am asking this because in Croatia we are continuously listening that the shipyards are not profitable and that they should be closed or should produce other things? How do you see the future of European shipyards?*

FALTINSEN: Well, I can go back to that when I was elected a member of Chinese Academy. I don't know why they needed to say this to the newly elected members, but it was like the following: *You should never say something about something that you don't know anything about*. And I agree with that. I don't think I should misuse the title, I should really not say anything concrete about the shipbuilding, because what I am saying I have not background for saying it. I would rather take a longer perspective of it about what can happen but then it depends on the country and on that what kind of maritime field is important for the country. For me it is very important not to narrowing down what is conventional type of ocean application. For me it is a broader aspect, but I am at university. Also, I don't know much about Croatia, but I can say for Norway, here the maritime field is very important. For Norway it is characterized by three major applications. One is related to ship transportation, but when it comes to shipyards they are more specialized in Norway. Then you have companies like DNV which is international, and as major fields you have ships and ship transportation. Then we have offshore, you don't have that in the same extent in Croatia I guess. Fishery and aquaculture is also important for Norway.

BRODOGRADNJA : *We have long tradition in shipbuilding; it is a common opinion that we should tend to build sophisticated ships because otherwise we can't compete with Korea and China.*

FALTINSEN: It is very simple, if it is very work intensive and you can't do this type of jobs automatically, you will go to the cheapest country and that is what has happened. When I was a kid then in Stavanger, in the city where I was living, there was a very big shipyard which was building very big tankers, the biggest in the world I believe. But then this market disappeared from Norway. Then Japan was the major country for this kind of jobs but they also had a problem with Korea. I am sure that we can not compete if it is a question of price and heavy use of labour. But you have for example Finland where they have been specialized for very long time in cruise ships and that has been a very big market. I guess it will continue to be so. It is also a question of other fields of use of the ocean. If you look at Japan, they have been trying very hard to go to other fields. They had very big research project on VLFS (Very Large Floating Structures, author's comment) related to floating airports and they did a lot of serious work in this field. The shipbuilding industry was involved in it but I am not sure what the current status of it is. Norway was lucky you may say because they found all that oil and gas and that industry is very big and advanced and the marine technology field is only one part of it. Then Norway is a very big exporter of fish products and you have a lot of projects with aquaculture. What I have been told is simply that there is not that much marine technology in aquaculture as long as you are in protected areas. It is more biology and things like that, but then it is a question of limited place inshore and maybe in the future they will go offshore and then you have the same type of problems. Also, I have been involved in more specialized type of problem that has to do with floating bridges. There are some floating bridges in Norway but I wasn't involved in that, however, I was involved in some plans to build submerged floating bridges. It has never been built, but it has a lot of advantages as I see it. I don't know how big this market is to be more specialized in reality. Once more, I am not talking about Croatia but I am talking about the general trend. The other dimension is that you have all that development related to oil and gas in the northern part of Europe and also in Russia of course. We have global warming and one is thinking about that there might be traffic from Europe to Asia open from ice. That creates a new dimension related to the ice technology. Then also as a consequence of energy crisis everything dealing with renewal of energy is interesting, wave energy, and in Norway there is an oil company that is involved in plans for floating wind mills providing energy to platforms. The platforms are creating a lot of CO_2 so if they can get energy like this it would help. I think that it must be fair to say that wave energy can never cover everything, it must be just a supplement related to it. It is also interesting to see how history is repeating itself. I was involved in similar type of questions in the1980s because then you also had an energy crisis. And then it disappeared. And now it comes back again. I think one should have much more continuing efforts related to it. I had to make a speech to the audience of about some thousand people when I was at the Academy of Engineering in China. I talked about challenges in ship and ocean technology. I think what is important is first to have a vision, a vision of 20 years. How are we going to use the oceans? That vision is not very good for shipyards which are fighting for survival, but one needs to have a vision. That might be wrong, but in my opinion one has to think like that and then can look into different types of scenario. I had a visitor in the spring, a representative from Japan. They make visits to different places in the world and they ask questions like that. They were supposed to have ideas about what are the different alternatives for the future and I don't know if their report will be open, but anyway what I basically said to them was that I used my presentation for the academy as a basis. They asked me many questions and I said that I can't answer all of it. For instance questions on transport capacity which is not my field. I can have some opinion about it but other people can have an opinion and we have to try to put that together and see where we are going. But it seems so logical that one has to think broader than conventional ships and I think, for sure, that has been the strength of our department here. We are lucky once more because this department has been dying as many west European departments in naval architecture. I mean they are not dead but they have declined severely because they reflect industry, but it is not like that here. I have heard that this year we are starting with over hundred new students in naval architecture or in marine technology. And they are top students, which means they are interested in that. I am talking about Norwegian students; in addition we have an international master program.

BRODOGRADNJA: *In Croatia young people are not so interested in the study of naval architecture although they have good job opportunities after graduation.*

FALTINSEN: Maybe they associated that with something which is not high tech. When you set up a curriculum for naval architecture you must not operate like it had been 30 or 40 years ago. I mean one aspect has to do with automatic control. We have a professor in this area and that has been a very strong subject for electrical engineering. A part of what they are doing has been related to the marine field and I think one should also think about a new type or more advanced type of activities which are needed.

BRODOGRADNJA: *How should we educate a "modern engineer"? What does in this context mean the development of CFD, the CFD application in design offices? What are, according to your opinion, the knowledge and skills that a modern engineer of naval architecture should acquire during study?*

FALTINSEN: So I think what is very important to acknowledge is that no person can know everything within naval architecture, but if you talk about advanced material, then you must have specialists. However, the major challenge is communication and team working. That is a very important aspect. Communication means also that you can understand each other, for example, when I am talking with a specialist in automatic control. Cybernetics is not in naval architecture traditionally, so they have their own language. I think one would benefit from the communication. You need specialists but the very big danger is, of course, that you get so many specialists which can't communicate with other specialists. So I think that a naval architect is not supposed to know everything of course. It is a tremendous field. At our department we have marine engineering, ship design, hydrodynamics and structural mechanics as examples of important fields, but it is difficult to have a very good basis in all that. Our department is divided into two groups. Students select one of these groups at a later stage. One is, I belong to this one, hydrodynamics plus structural mechanics, it also involves marine cybernetics and there is also a professor in nautical studies. We have a special program in nautical studies. That is not a big activity here and that type of job was created because of education of officers for the field, but the tendency is that this type of schools disappears in Norway and the ship owners use foreign people instead. So that is not that big market. But for sure in marine cybernetics you have a lot of industry related to equipment onboard ships and they are involved in, and as I said, most of them are not educated from us. So how do I see CFD in this context, you asked. I guess I said that when I had a lecture down in Zagreb (at the Croatian Academy of Sciences and Arts in spring of 2007, author's comment). It is of course useful, but a big danger one is doing is to believe only in CFD. One advantage is the ability to do the 3D modelling of a system. This could be also done with analytical means, and another aspect is experiment, and the third aspect is CFD. One needs all of them together, but I think it is important that students get knowledge about CFD. We have course in CFD at the fifth year. They don't learn too much relative to it, I think it is important to learn what limitations and capabilities are related to doing that. There has to be a balance, it cannot be like you are only making input and getting output and don't know what it is. I mean, you have an answer but how true it is in reality. It depends on the field of application, I mean. If you talk about ship hydrodynamics there are cases where CFD is very good, but it is very hard in seakeeping world to do calculations like that. If we go back, that is also why strip theory survives. But then I always say if you are designing a new structure you must do experiments. Once more, I was heavily involved in the activities of the ITTC, and going back many years, at the ITTC they were afraid they were going to loose relative to CFD, but they have not lost. I think there is a still big market and they are building new testing facilities, similar like we have here within the ocean environment laboratory, a big basin which is 50 m times 80 m. That type of laboratory you have also in Wageningen. Also, in November there will be an official opening of a facility like that, I believe, in some aspect has more possibilities than one in Wageningen, at *Shanghai Jiatong University* and they are also building one at *Harbin Engineering University*. Seen from the university point of view, I see the clear advantage, like we have also here, in very small facilities, very specialized type of facilities.

When you talk about educational program then I think that the ability to try to solve new problems is very important and that is of course very demanding. If I go myself back when I started I didn't know what problems I was going to work on. Suddenly there was a big interest in high speed vessels. These problems are very unique. It is very hard to let the students be able to solve completely new problems in a 5-year study. In hydrodynamics what we had been trying to do is more to try to simplify the problem. But still it should not be too simplified, so that it doesn't give relevant meaning. I think one learns much more by simplified models. You can do a simple model and do seakeeping analysis. OK, it is far from perfect but I think you get more insight and then you come for a new system whatever it is and then you may be able to deal with wave energy problems, you can deal with fish farms or you can deal with a harbour or whatever. Of course, in practice you must have software and experimental facilities, but for students it is important to get understanding through simple models. That is also important when you are going to check the output that you get from CFD or from testing facilities.

BRODOGRADNJA: *I am witnessing here very devoted attitude towards science. How is research work in the CeSOS financed? I see many PhD students from whole world working here, how are they financed … by the government, by the industry or mixed?*

FALTINSEN: It is mixed. The *CeSOS* is financed by the National Research Council. They decided that they should have a limited number of centres of excellences. They get that money from the government but it is a separate organization. So, in some way you can say that it is the government. When you have this money, then industry also supports it when it is in the engineering field. Who is involved now is oil industry, now it is only one oil company in Norway, and that is *StatoilHydro*. Then *DNV* also provides financial support and *MARINTEK* as well. The budget is about 40 million NOK, I don't have the exact numbers, but let's say is fifty- fifty between industry and Research Council. Also, the university provides extra money, extra support, you get extra positions from the university. The objective of *CeSOS* is to do fundamental research which is relevant and I have to deal with something that has marine applications. Our product is to publish; our product is not making computer program that should be used by the industry. Besides, we are educating people to get PhD level and those people then go out to industry.

BRODOGRADNJA: *You have here more than 50 PhD students. What possibilities of employment they have after achieving PhD level in Norway? Does industry appreciate the PhD level?*

FALTINSEN: Their opportunities in industry are big. It doesn't mean that they do research there. If I look at my PhD students, some of them for sure have gone to *DNV*, *MARINTEK* like Renato (Renato Skejic, author's comment), engineering companies involved in offshore types of activities, some of them are professors here but there are very few of them. The major part is not here and my impression is that they generally have good possibilities and then, of course, they will not do research where they go but they can be hopefully involved at advanced type of projects. That has been very fortunate also for our MSc students. Generally speaking, there has always been good market for them. There have been bad times in general but still, I think, they always manage to find a job. The maritime industry is for sure very broad here and it means a lot for the Norwegian economy.

BRODOGRADNJA: *Behind yourself you have enormous job and achievements: about 300 scientific papers and almost three books, more than 40 doctoral theses. What it next?*

FALTINSEN: I don't know. If you ask my wife, she will say I should retire. Generally speaking, I have activities in the centre here and now I am sitting and writing books. I think I am working too much. I am not stressed, but I don't think is healthy to be too single minded. I don't know what is next but in general what fascinates me are new topics.

BRODOGRADNJA: *What is the best part of the work you do, the part that gives you the most satisfaction? Conversely, what is the downside of your work?*

FALTINSEN: Well, I like very much to teach, I mean teaching at different levels. I like to stand and lecture to students, but I don't want to teach the same things for 30 years. So, I try to get involved in new things. I like very much to be involved with students. A certain thing one has to do, but I am not doing it now, is that one has to be involved with administration and I like that too, you see. But you must select, I think, you cannot do everything. And what I dislike relatively to that is that is the tendency for too many meetings. So, my attitude is that meetings should never last, let's say more than one hour whatever. Professors are so talkative and they are often not result oriented and I don't like meetings. I emphasize very strongly that meetings should not go on very long because they are not productive. Then, I could very well say that I would not like to sit down and grade 100 or 200 exams, but I don't do that, I am not involved in that. What is fascinating for me, as I said before, are new types of problems. I like to do new type of things and also I think I still can learn many things. Our centre is for ten years and now we have 4.5 years left. The idea is that afterwards you have to create something new, so this is not forever. Maybe it is a good idea, because otherwise you take it for granted that you have all that money. So, you have to create something new. For sure, I would never like to sit down and do a lot of calculations with some standard programs as a consultant. People may call me for consulting; I would never do that type of jobs because that type of jobs is done by *MARINTEK*. However, I can be involved as a specialist in one- or two-day type of projects, because you get lot of insight by doing that. It is a question of making priority, for sure. But then I like to joke, you see. People want me to do certain type of things and they tell me: *You just say the amount of money and we will pay*. And I say: *I don't want that, but if you can give me one hour more every day so I have 25 hours to work then I will take the job*. You don't have a job at university if you are interested to be very rich, I don't think so. By working at university you have very big freedom. I have had, but not anymore, many sabbaticals. That has been very interesting because you learn different societies. I have been at the *MIT* then in Japan and so on, so I have the freedom to do that. You can misuse that freedom, but if you use the freedom in positive way, than it's very good.

BRODOGRADNJA: *Professor Faltinsen, thank you very much for your time. It was pleasure talking to you.*

Trondheim with the *Marine Technology Centre* in the foreground

Victor DUBROVSKY

A Few Words on Small-sized Motor Yachts

(or Do You Remember the Yacht *Duncan*?)

1 Introduction

There are some unusual literary heroes in the novels of the French science enthusiast and romantic Jules Verne, like for instance the sail-steam powered yacht *Duncan*. Repeatedly the ship was so reliable shelter and mobile sweet home of Captain Grant's searchers, so that finally she seems as being one of the novel characters.

The yacht was described by Verne as a typical result of contemporary shipbuilding development according to him. Like any sail ship, she has a big range, not so severe motions because of the damping effect of the sails, and permissible level on for the passengers and crew comfort.

It is very good that today there are people who can have yachts and can travel at seas for own pleasure and rest. Today hundreds maybe - thousands, yachts are proposed for lovers of sea travels. It seems no principally new and better options can be proposed. But it is not absolutely so – or even not so absolutely. Let us examine the types of today proposed yachts.

Firstly let us note that a pure sailing yacht, even with a small auxiliary engine, gives, of course, really completely a chance of "immersing into nature". But the increased rate of contemporary life allows assigning not restricted time for sea travel rare enough. And exact planning of the sailing travel time is hard enough in general. Therefore, a motor yacht with high enough – at contemporary conditions - speed seems an optimal option for active owners.

Of course, if a motor yacht has to be a floating presentation residence, no radical developments are needed in a comparison with proposed options. The yacht residence can have speed for smooth water only, and the speed can be selected by the principle "greater than that of the neighbourhood" – it is simple enough for smooth water. Such yachts are usually big and expensive *monohulls*. An artist/image maker plays the most important role in designing of such ships, and not a naval architect. Usually a large enough displacement compensates for the lack of interior space and ensures high performance in small waves. But the main practical goal of such yachts is standing in sheltered bays and impressing passers-by with the refined exterior.

But a desire for long enough sea voyages generates the problem of higher seakeeping performance. It is a serious problem if the yacht's displacement is smaller one. For example, only a higher seaworthiness, and not the power supply, can ensure a high enough average speed in any waves, including severe ones, with the permissible level of comfort. (The analysis of full-scale data shows that the speed in waves is defined by power supply only at waves of height no more than 20% of displacement triple-stage root. Only a higher seaworthiness can ensure high enough speed in more severe sea.)

Seaworthiness can be radically improved with the application of *multi-hull types* of motor yachts, because all multi-hull ships have more or less higher seakeeping performance in comparison with *monohulls*.

On the other hand, the application of a *multi-hull yacht* can decrease building costs for constant inner deck area, because all multi-hulls have a bigger deck area relative to displacement in comparison with *monohulls*.

Today some *catamarans* (twin-hull ships with equal hulls of conventional shape) are applied as motor yachts. Lately, some *proas* (twin-hull ships with a bigger main hull and a smaller outrigger, both of traditional shape) have been built [1], for example, Figure 1.

Figure 1 **Chinese-built *proa* as a motor yacht**

Like all multi-hulls, *catamarans* and *proas* differ from *monohulls* by relatively bigger deck area for constant displacement. As all multi-hulls, these ships have no problem with ensuring of transverse stability, any needed stability (up to that equal to the longitudinal one - for a catamaran) can be ensured by corresponding transverse distance between the hulls. Also, the hulls can have any desired aspect ratio, which is a sufficient advantage for high enough relative speeds from the performance point of view.

A correctly designed *catamaran* has sufficiently smaller amplitudes of roll motion, and approximately the same vertical amplitudes of roll acceleration.

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But the pitch motion of a *catamaran* and *proa* is approximately the same, as the same motion of a *monohull* of the same length and displacement for the same speed. Unfortunately, sufficient mitigation of *catamaran* and *monohull* pitch is practically impossible, because the "arm, by which waves rock the ship", the longitudinal metacentric height, is too big (usually about ship length). Then a mitigating moment must have the order of displacement and length product, i.e. is possible rare enough.

A sufficient decrease of the longitudinal metacentric height (and corresponding disturbing moment) can be achieved by small water-plane hull shape. Usually such a hull consists of a bigger under-water volume, "gondola" or "under-water hull", and thin strut(s), which intersect water free surface, and connect the gondola with the above-water platform. A sufficient, up to 3-5 times, decrease of the longitudinal metacentric height means a corresponding decrease of the disturbing moment and needed mitigating moment from motion mitigation devices.

The foregoing proves to beginning of small-sized motor yacht type selection – for ensuring of high seakeeping and safety, and big enough area of decks relative to the displacement.

2 The small-sized type and characteristics of motor yacht of increased seaworthiness

It seems evident that the small-sized motor yacht of increased seaworthiness has to be a ship with a small water-plane area. However, a more detailed examination of the relative dimensions and speed is needed for the selection of the number of hulls.

Overall dimensions of the above-water platform are defined by the area of the accommodation space for passengers and crew and by service and auxiliary accommodation in addition.

For example, a possible customer wants an accommodation arrangement for 6 passengers in double cabins (including the owner's cabin) with a cabin area of about 15 sq m. The saloon area has to be 40 sq m. For more or less long time permanent sailing, a crew of no less than 9 persons is needed (a captain-mechanic, if it is not the owner, three watches by 2 sailor-motorists, a chief sailor and a cook). In addition, two more persons, navigator and steward, are very much desired in order to relieve the crew of some duties.

To ensure a minimal area of the water-plane and at small enough displacement, the engine rooms have to be placed in the above-water platform. Of course, it means bigger mass of insulation of the living quarters against vibration and noise, but it can increase seakeeping sufficiently. Then the platform area is increased at two engine rooms and corresponding stores.

One of the possible options of the platform arrangement (for the previously noted number of crew members) is shown in Figure 2. Two semi-rigid boats and corresponding crane are placed after the platform stern bulkhead.

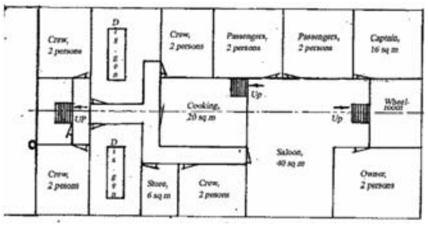


Figure 2 **An option of platform arrangement**

For higher non-sinkability, the platform will have exits to the upper deck only. It means that the yacht will be afloat even if both hulls are flooded. (It means, the windows will be the watertight ones, and air inlet will be on the level of the wheelhouse upper deck. Air inlet for the diesels will be placed near funnels on the upper deck). The yacht can sail at very severe sea, including periodical immersion of the upper-deck level.

Electrical power transmission is assumed for the most simple control and minimal area of water-plane. But it means, of course, more expensive and heavy power plant.

The result overall dimensions of the platform (30 m x 12 m) allow estimation of permissible relative speed and needed type of the ship (number of hulls). If the ship full speed will be about 25 knots, it corresponds to the Froude number of about 0.7. A twin-hull option is preferable for the value of relative speed, because (shorter) three hulls with relative speed of about 0.9, and positive interaction of the hull wave systems are not possible regarding the permissible longitudinal clearance (distance between the hulls). An alternative option is an outrigger SWA ship, but the type has a relatively bigger relative water-plane area, i.e. worse seakeeping. Thus, the twin-hull option was assumed for a small-sized motor yacht.

The design full speed must be achieved at 85% of the remaining power (after power extraction for other ship needs) in Sea State no more than 3. Economy speed (about 12 knots) will be defined as achievable one for the load of one diesel-generator no less than 30-35%.

Endurance by fresh water and food must be 10 days, range at economy speed – about 2500-3000 Nm. Range at full speed will be about 1500 Nm (with 20 % sea supply).

One of the general advantages of all SWA ships is a possibility of wide draught variations, because small enough volume of struts can be compensated by not so big volume of gondola tanks. It means that the design draught with full supplies can be about 1 m, i.e. approximately equal to the gondola draught, and the same minimal water depth. Sailing in smooth water is convenient option for that minimal draught.

Design (bigger) draught (approximately on the half height of the strut) is needed for sailing in waves, and can be achieved by small enough ballasting.

Referring to the practice of the Japan-built passenger ferry, the vertical clearance (the distance between design water-line and platform bottom) must be about 50% of the design wave height. Thus, the clearance no less than 1 m is needed for Sea State 4. The clearance and design draught were assumed equal to 2 m.

The ship survivability and energy supply will be provided even if one hull will be fully flooded. The above-water platform must ensure the ship's floating even if both hulls are flooded; it means, windows must be water-tight ones, and all entrances will be places on the level of the platform upper deck. Ventilation air inlets will be placed on the level of the wheelhouse upper deck.

The ship interior will be specially designed, for example, with luxury hatches in the upper deck for the saloon and kitchen. The upper deck will be equipped with a movable tent and solarium. Stern ends will be equipped with inlets to water and landings for outlets from water. Safety rafts will be placed behind the wheel-house and funnels. Besides, some additional equipment can be provided:

- possibility of light helicopter landing and its refuelling;
- saloon for underwater viewing with outer sources of light;
- cell net between hulls;
- equipment for aqualung service.

The initial transverse stability of the yacht will ensure heel of no more than 10 degrees, at rest on side wind with a speed of 50 knots. The strut area was selected based on these conditions, and will be about 2 x 15 sq m.

Figures 3 and 4 present the results of approximate estimation of the pitch and roll amplitudes of the yacht compared to the full-scale data of a 1000-t *monohull* [2]. (The yacht characteristics were re-calculated from the model test data of a *twin-hull* model with 1.5 times bigger relative area of water-plane.)

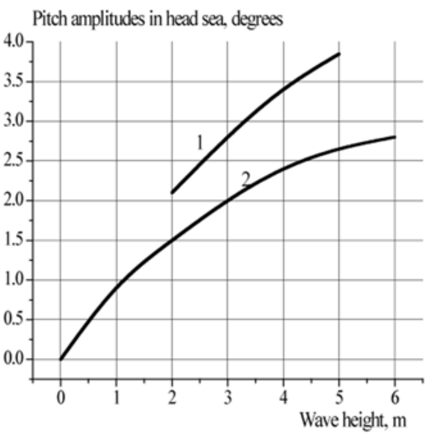
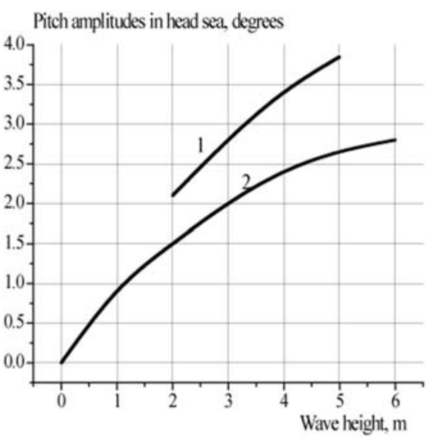


Figure 3 **Pitch amplitude comparison in head sea, speed 10 knots: 1 – monohull, displacement 1000 t; 2 – SWATH, displacement 125 t, without motion stabilizers**

Figure 4 **Roll amplitude comparison in side sea, speed 10 knots: 1 – monohull 1000 t, 2 – SWA yacht, 125 t, without motion stabilizers**



It seems evident that the proposed yacht will have smaller pitch than the *monohull* at 8 times bigger displacement. If the pitch level of about 2 degrees is taken as maximally permissible one, the examined yacht will ensure the level in wave height of about 3 m, and the compared *monohull* – of about 2 m.

It is evident that the proposed yacht even without motion stabilizers has smaller roll than the *monohull* at 8 times bigger displacement. If the roll level of about 6 degrees is taken as the permissible one, the examined yacht will ensure the level in the wave height of about 6 m, and the comparable *monohull* – about 2 m.

Even a lower level of motions can be ensured by controlled foil-stabilizer application. Figure 5 illustrates the influence of foil-stabilizers and relative speed on the pitch and roll amplitudes of a twin-hull prototype (displacement of about 5 t) in irregular waves having a height of 0.7 m (relative height about 0.4). Relative area of foils is about 15% of the water-plane area.

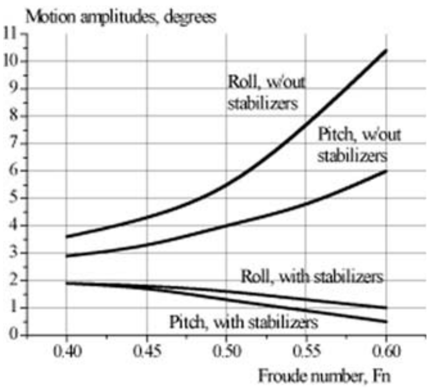


Figure 5 **Foil-stabilizer influence on motions of 5-t twin-hull SWA prototype in wave height 0.7 m [2]**

These data confirm the general specificity of all *SWA* ships: high effectiveness of motion stabilizers because the forces and motions generated by them are comparable with the disturbance forces and moments of hulls with small water-plane area.

The decision concerning the stabilizer application can be made in the later stages of yacht designing – in dependence on the potential owner's desire.

If motion mitigation is needed for stoping regimes in sea, by air tank activations [3] can be applied instead of foil-stabilizers (or additionally to them), because any foil-stabilizers are not effectively enough at zero speed. The volume of the same tanks can be used for draught changing from minimal one to draught needed for waves.

An option of the general arrangement is shown in Figure 6, main dimensions and general characteristics are given in Table 1.

Table 1 **Main dimensions and general characteristics of SWA motor yacht for 6 passengers**

Characteristics	Unit of measure	Values
Overall length	m	30-32
Overall beam	m	12-14
Design draught	m	2
Depth	m	9.5 – 10
Design displacement	t	120-130
Deadweight	t	15-20
Engine power	MW	2×0.5
Endurance	days	10
Range at economy speed	Nm	2500-3000

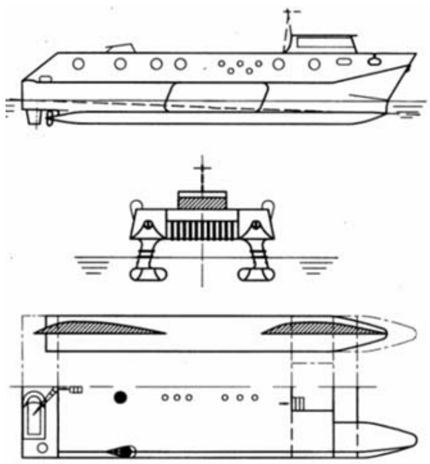


Figure 6 **General arrangement of SWA motor yacht for 6 passengers (displacement about 125 t, full speed about 25 knots, engine power 2 x 0.5 MW).**

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[3] DUBROVSKY, V., MATVEEV, K., SUTULO, S.: "Small Water-plane Area Ships", Backbone Publishing Co., ISBN-13978-09742019-3-1, 2006.