Construction and Shipbuilding

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LABELS

Sign	Unit of measurement	Name
S_W	m^2	Surface of a construction (or any) water line
A_M	m^2	Area of a main frame
A_w	m^2	Effective windward area
B	m	Breadth of a ship
B_{max}	m	Maximum breadth of a ship
B	-	Buoyancy
C_B	-	Coefficient of displacement fullness
C_M	-	Coefficient of main frame fullness
C_P	-	Coefficient of longitudinal fullness
C_{VP}	-	Coefficient of vertical fullness
C_W	-	Coefficient of water line fullness
D	m	Depth of a ship's side
D_L	t	Mass of a ship at full load
f	m	Freeboard
g	$m.s^{-2}$	Acceleration of gravity
G	t	Mass of a useful cargo
H_0	m	Longitudinal metacentric height
h_{fix}	m	Height of a fixed point (from DWL)
h_s	m	Height of a wall
L	m	Length of a ship
L_{max}	m	Maximum length of a ship
m	-	Metacentre

Sign	Unit of measurement	Name
M_{ALd}	-	Tolerable moment for a dynamic heeling
M_{oy}	m^4	Static moment of a water line surface
M_R	kN.m	Righting moment at inclination of a ship
M_{Wd}	-	Moment by a dynamic wind effect
P_W	Pa	Unit wind pressure
S_j	m^2	Surface of a water line
t_S	mm	Thickness of a girder's wall
T	m	Draught of a ship
T_F	m	Forward draught (on a bow)
T_A	m	Aft draught
T_{max}	m	Maximum draught of a ship
v	$km.h^{-1}$	Speed
$V=\nabla$	m^3	Volume displacement
w_θ	sec^{-1}	Angular frequency of a free ship rolling without consideration of resistance forces
Δ, D_w	t	Weight displacement (deadweight)
ΔL	m	Frame spacing
ΔT	m	Water line spacing
θ	°	Heeling angle
θ_{AL}	°	Tolerable angle of a transverse heeling
ρ	$kg.m^{-3}$	Density
σ_{pl} d	MPa	Gliding limit
ψ	rad, °	Inclination angle of a ship
ω_j	m^2	Surface of a frame

LIST OF ABBREVIATIONS

Abbreviation	**Name**
$\otimes$	Main frame
DWL	Construction water line
CL	Axis plane
BL	Basic plane

INTRODUCTION

When we look at a map of the world we can mostly see the surface covered with water. Water transport, mainly sea but also river transport, takes part in cargo traffic to a large extent worldwide. Although the majority of consumers are unaware of the fact, the bulk of cargo is carried by sea transport around the world.

Thanks to two thirds of its outer sea boundaries the European continent is a sea power, in particular after the EU expansion. A long seacoast of Europe and a large number of ports predestine the sea sector to become a valuable alternative to land transport, mostly the road one. Sea coastal transport has been expanding in recent years as intensively as road freight transport and it definitely has even a bigger potential.

Construction and production of transport means is closely related to the development of transport. Watercraft is the oldest means of transport built by a man. The first archeologically documented boats are 8000 years old. We are speaking about hollowed tree trunks - monoxylons (dugout canoes).

During the history watercraft construction has been improving from the point of view of construction as well as drive, from a paddle to an atom energy.

The reason why a solid body floats on the water surface has been a point of people's interest since ancient times. The first known scientific studies about this issue come from the ancient Greece, when a mathematician, physicist, mechanist, inventor, astronomer and philosopher Archimedes of Syracuse (c. 287 - c. 212 BC) formulated the first physical principle from fluid mechanics known as the Archimedes' principle. In 1746 a French mathematician Pierre Bouguer published a theorem "Treatise of Naval Architecture" (Traité du navire, de sa construction et de ses mouvements). In this book Bouguer solved the problem of ships' stability and explained the rules for their buoyancy calculation. Ship theory was further developed by excellent mathematicians Leonhard Euler and Daniel Bernoulli. Their principles regarding ship navigation found their practical application very quickly. The first ship constructors who brought theoretical knowledge of these scientists into real life were: Pett, the Englishman, and Champan, the Swedish. Their "Mercurius" was the first ship built according to the newest rules. It started its navigation in 1747 and it belonged to the category of so called East India ships providing trade connections with the East India and China. Ships of this type were equipped with plenty of sails and armed with cannons to defend against pirates. The construction of "Mercurius" ship reflected a new perspective on design concepts and technologies. The hull was slimmer, extended and the whole design was more compact.

The progress in physics and mathematics as well as changes in technologies brought a need to lay theoretical scientific foundations for shipbuilding. The first foundations of "Ship Theory" laid by French scientists in the 18th and 19th century significantly contributed to the development of such a ship building we are familiar with nowadays.

The textbook "Ship Theory" is intended for students of a Bachelor and Engineer Study studying the field of Water Transport at Faculty of Operation and Economics of Transport and Communications of University of Žilina. It deals with physics of vessels under the conditions of sea and inland navigation. It provides a set of knowledge gained during ship designing, closely related with their operation.

Authors

1. BASIC DATA ON VESSELS

According to Slovak Technical Standard (STN) a vessel is defined as a floating body intended for a certain activity on water. To assess the size, shape and anticipated navigational and utility properties of each vessel basic data, main parameters and ratio indicators are decisive; they are used to assess a vessel from a technical point of view as well as to compare it with other vessels. This data is likewise important from vessel classification, registration and operation aspects. Evaluation criteria and parameters definitions are determined with international and national technical standards. The nomenclature and parameters of ships in the Slovak Republic are defined with the standard STN 32 0000 (enacted on 1 March 1997).

1.1. BASIC DIVISION OF VESSELS

Vessel - the most general body floating on water which is adopted to a certain activity (cargo and passenger transport, machine and mechanism carrying).

Ship - a bigger navigable vessel which is adopted to navigation with good navigational and manoeuvrability properties.

Engineering vessel - any vessel equipped with a mechanical device intended for work on water roads or in ports which is able to move on water either using its inherent propulsion or another vessel (dredger, crane, pump, etc.).

Floating machine - a laid-up vessel located in a certain place which serves for a certain purpose (to moor vessels, floating docks, sheds, etc.).

Rafts - the most primitive type of watercrafts built of floating bodies mutually connected into one whole for the purpose of their own transport.

Sporting and recreation crafts - small crafts intended for sporting and recreation purposes.

1.2. DESIGN GROUPS AND BASIC PARAMETERS OF A SHIP

- Body
- Superstructure
- Energy mechanisms and engine room equipment
- Propulsion devices
- Ship mechanisms
- Ship systems
- Electrical equipment
- Radio devices
- Special devices

Navigational parameters of a ship:

- Buoyancy
- Stability
- Unsinkability
- Seakindliness of oscillation
- Running (operational) properties
- Steerability
- Strength

Operational - economic indicators:

- Tonnage
- Stowage
- Number of crew and passengers
- Speed:
    - design,
    - operational,
    - technical.
- Autonomous operation
- Compliance with operation requirements
- Accommodation conditions

- Price
- Operation costs

1.3. MAIN SHIP DIMENSIONS AND COEFFICIENTS OF THE SHAPE

- Length: *L* [m]
- Breadth: *B* [m]
- Depth of a side: *D* [m]
- Draught: *T* [m]

Main Planes

- *Axis Plane (Centre Line - CL)* - a vertical longitudinal plane of a vessel which is mostly a longitudinal plane of a vessel hull symmetry.
- *Main Frame* - a vertical transverse section through a vessel hull in the centre between perpendiculars which is perpendicular to the axis plane (most frequently in the position of the widest breadth of the vessel hull).
- *Base Plane (Base Line - BL)* - a plane through the lowest point of a vessel hull (from inside of a shell) which is perpendicular to the axis plane and usually parallel to the keel or construction water line.
- *Construction Water Line (Designed Water Line - DWL)* - a plane of the greatest (designed) draught the vessel is designed for.

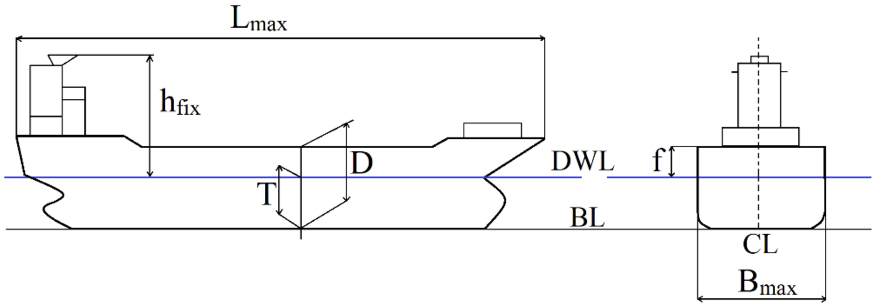

Fig. 1. Main Dimensions of a Vessel [Authors]

Basic Relations

L/B - a ratio of a ship's length to its breadth. It affects the speed, course-keeping stability, transverse strength, steerability, etc. It ranges from 5 to 10.

B/T - an index number which allows to assess the transverse stability, resistance, transverse strength, etc. In comparison of river and sea ships there exists a significant difference in values.

D/T - an index number which affects the transverse stability, hull strength and the unsinkability. It usually ranges from 1.1 to 1.7.

L/D - a ratio of the length of a ship to the depth of a side mainly points out a longitudinal (overall) strength of a ship's hull.

Coefficients of fullness

Coefficient of displacement fullness $$C_B = \delta = \frac{\nabla}{L.B.T} \quad (1)$$

It is defined as a ratio of the displacement ∇ to the volume of a block which is a result of design dimensions of the immerged hull part. It is the most important and most frequently used coefficient of fullness. It points out the overall fullness of the ship's hull. Upon substitution of index numbers L/B and B/T it allows to make a relative image about fineness of shape of the ship's hull, about making the stern and forebody of the ship finer when compared to other ships. Its values range from 0.5 - 0.95.

Coefficient of water line fullness $$C_W = \alpha = \frac{A_W}{L.B} \quad (2)$$

It is defined as a ratio of the area of a main (or any) water plane to the area of a circumscribed rectangle. It affects the watercraft resistance, direction stability, tonnage of a ship, etc. Its values range from 0.7 - 0.95.

Coefficient of main frame fullness $$C_M = \beta = \frac{A_M}{B.T} \quad (3)$$

It characterises the transverse and direction stability, speed, steerability, tonnage of a ship, etc. Its values range from 0.6 - 0.98.

Coefficient of longitudinal fullness $C_P = \phi = \frac{V}{A_M.L} = \frac{C_B}{C_M}$ (4)

It represents a ratio of the displacement to the volume of a prism created from the area of the main frame and length of the ship. It characterises the distribution of the displacement along the length and mainly the sharpening of the forebody and stern.

Coefficient of vertical fullness $C_{VP} = \chi = \frac{V}{A_W.T} = \frac{C_B}{C_W}$ (5)

It represents the fineness of a hull through a main water plane prism and it determines a certain image about the usage of ship's hull space along the length.

1.4. CLASSIFICATION AND DIVISION OF VESSELS

The classification of ships expresses the level a ship or a vessel complies with rules set by a classification organisation. The classification represents a categorisation of the vessel into a class on the basis of classification rules. It is performed during the vessel construction and operation.

1.4.1. Ship Classification Bodies

A supervisory body - classification organisation supervises the construction and operation of a ship. This organisation approves the type and technical documentation for the construction and reconstruction of vessels with the exception of small private watercrafts, and it performs a technical supervision over the construction and operation of vessels. This body has its own classification regulations which apply to:

- passenger and tank ships (intended for transport of flammable and hazardous substances), tow and pusher tugs independent on performance of main engines and ship body dimensions,
- ships with inherent propulsion above 40 kW,
- ships with coefficient of main dimensions $L.B.D \geq 100\ m^3$,

- special designation vessels, such as ferries, floating machines, floating devices, etc., longer than 20 m,
- small watercrafts with their length less than 20 m intended for transport of more than 12 passengers or intended for towing, pushing or leading of a side arrangement,
- recreation watercrafts of any type regardless of the propulsion method with the hull length from 2.5 m to 24 m intended for sporting purposes.

A classification organisation categorises a vessel into a class. All changes in the class or its restoration are made based on the results of a vessel inspection. A classification licence of the vessel is issued regarding the categorisation of the vessel into a class.

Class

A vessel is categorised into a class based on its type, design manufacture, technical-operational properties and shipping area the vessel is intended for. The granted class is marked in a classification certificate; it is a certificate of technical qualification, a class certificate of a small and recreation watercraft in compliance with rules for small ships classification and construction.

The class grant is recorded into a Register Book. The class is renewed every 6 years at most, however, every 4 years for passenger ships, tank ships and icebreaking ships.

Class marks:

"KM" - for ships with inherent propulsion,
"K" - for ships without inherent propulsion.

The classification organisation also deals with other rules which are not presented in this basic division.

The rules include the following parts:

- classification,
- a ship's hull and its equipment,
- fire protection,
- energy mechanisms and systems,
- ship's installations and their equipment,
- electrical equipment,
- radio link devices,
- navigation devices and equipment,
- special requirements on ships transporting hazardous cargo,
- means to avert environment pollutions caused with ships.

From the point of view of wave conditions inland ships are assigned areas "0", "1", "2" and "3" which have the following meaning:

"0" - for ships intended for navigation with waves up to 3.0 m,

"1" - for ships intended for navigation with waves up to 2.0 m,

"2" - for ships intended for navigation with waves up to 1.2 m,

"3" - for ships intended for navigation with waves up to 0.6 m.

From the point of view of wave conditions ships of a mixed navigation (river - sea) are assigned classes which have the following meaning:

"L" - ships with a standard wave height up to 2.0 m,

"S" - ships with a standard wave height up to 2.5 m,

"SM" - ships with a standard wave height up to 3.5 m.

Besides the additional mark "L" a symbol "A" is added at a high degree of equipment automation. If a ship does not comply with classification rules in full extent due to equipment and other design solutions unverified in practice, a mark "E" (experimental solution) is added there before the class symbol; it can be removed after a long-term verification in practice.

The following activities defined in rules are suggested for the function and scope of classification society work (ship register).

Classification process - preparation and issue of rules, inspection and approval of technical documentation needed for designing, construction, reconstruction, modernisation, restoration and repair of ships, production and repair of completing products and production of materials for a ship, assigning classes to ships, confirmation, renewal and grant of a class to a ship based on the ship certification throughout the entire period, operation of each ship until its discard with the preparation and issue of relevant documentation, grant and issue of certificates.

The International Association of Classification Societies (IACS) currently consists of twelve members, each having its own classification rules. The table below mentions four biggest classification societies with their international acronyms and symbols.

Table 1

Classification Society

Classification Society	Acronym	Symbol
American Bureau of Shipping	ABS	ABS ✠ A1
Bureau Veritas	BV	I
Lloyds Register of Shipping	LR	✠ 100A1
Det Norske Veritas	DNV	✠ 1A1

Source: Official websites of the classification societies

Codes used by individual societies are not mutually interchangeable among organisations which means that every letter or number represents different characteristics. Each code can be divided into individual sections which express the ship's properties, equipment and limitations.

Example:

DNV✠1A1 SF DYNPOS-AUTR RP E0 HELDK-SH W1 SBM

DNV✠1A1 - a classification society, a construction symbol and granted class (assigned to all vessels),

SF - requirements on stability at a damage,

DYNPOS - AUTR - Dynamic Positioning with the additional DPS,

RP - additional propulsions,

E0 - outfit of instruments and automation installed in a vessel which enable the operation without the engine room supervision,

HELDK-SH - a helipad or a constructed platform complying with basic requirements for strength and other requirements for safety of ships,

W1 - an integrated navigation system,

SBM - management of safety and protection of the environment at ship operation.

Fig. 2. Example of Classification Organisations

Source: Official websites of the classification societies

The basic division depends on the material used for the design and construction of ships, on propulsion device type, propulsion principle, and the purpose the vessel was constructed for.

Division by the material used for the hull construction:

- steel,
- aluminium and light metals,
- plastics - laminate,
- wooden,
- composite,
- ferroconcrete and other.

Nowadays commercial ships are constructed mainly from steel, even from a special ship steel with an increased strength when speaking about sheets as well as profiles. Aluminium and light metals are used for the construction of small and mainly fast ships, however, also for the construction of superstructures, life boats, etc. Recently they have also been used for facing and furniture production. Plastics and laminates are mostly used for the construction of boats and various sporting and recreation ships.

Wood after inflammable finishing is currently used for the construction of ships only very rarely; it is mainly used in the construction of superstructures where, however, it is largely substituted with light metals and plastics.

Ferroconcrete is used with success even today, not only in time of war. It is mainly utilised for the construction of specialised vessels which are moved with tugs to their destination place and thus have a fixed operation position. Examples include docks, pontoons, botels and restaurants, floating sheds and various special vessels.

Vessels made from composite materials were used in the past when a supporting structure of wooden ships began to be abandoned; it was substituted with a steel structure, whereas the shell remained wooden. Currently composite materials are mainly used for the construction of small and sporting ships.

Division by types of used propulsion devices

Such a division is preceded with a general division of ships into self-propelled and non-self-propelled ones:

Self-propelled ships:

- propeller,
- wheel,
- hydro-reactive,
- with a Voith - Schneider propulsion device,
- untraditional propulsion devices.

Non-self-propelled ships:

- boats.

By engines

- diesel,
- turbine,
- steam,

- steam turbine,
- special.

By the principle of movement

By this principle ships are divided into:

- displacement,
- skimming,
- hydroflights (on submarine wings),
- aeroflights (on air wings),
- air-cushion,
- combined.

By the performed activity

By the performed activity ships are at first divided into transport (commercial - trade), technical, and auxiliary ones.

Transport ships are further divided by the performed type of transport into:

- cargo ships,
- passenger ships,
- tugs:
  - tow,
  - pusher,
- cargo ships intended for dry cargo,
- cargo ships intended for liquid cargo - so called tank ships intended for transport of natural liquids and liquid gases,
- specialised (high-speed ships and others).

By the navigation length

- local navigation (cabotage, coastal shipping),
- offshore navigation.

By the navigation area

This division is determined with ship classification rules and shipping safety rules of the Transport Authority.

2. BUOYANCY

2.1. LINES PLAN

A geometric shape of a body is usually set graphically, i.e. by means of a lines plan which represents projections of sections onto three mutually perpendicular main planes. Nowadays in the era of computing technologies some highly sophisticated ship-building computer programs are commonly used there in the practice thanks to which the lines plan takes on a digital form.

Main (projection) planes are as follows:

- the axis plane,
- the plane of a construction water line,
- the plane of a main frame.

Section planes of a body surface with planes parallel to the axis plane are *sheer plans* and the lines are *longitudinal section lines*. The projection of all longitudinal section lines is called a *theoretical sheer plan of a hull*.

Section planes of a body surface with planes parallel to the construction water line are called *half breadth plans* and the lines are called *water lines*. The projection of all water lines is called a *theoretical half breadth of a ship*. (Half breadth is a term adopted from the practice since as a result of the ship's hull symmetry only half (not entire) water lines are plotted on the lines plan, i.e. water lines on a half breadth of a ship.)

Section planes of a body surface with planes parallel to the plane of the main frame are called *frames* and the lines are called *section frame lines*. The projection of all frames onto the main frame plane is called a *hull* (or *theoretical hull*). On the hull from the right side of the axis plane there are *forward frames* and from the left side there are *stern frames*.

The actual shape of longitudinal section lines is captured only on the projection plane called a *shipboard*; on the other ones they are represented in the form of lines. Water lines have their actual shape captured only on the projection plane called a *half breadth* and frames are captured on the projection plane called a *hull*; on remaining two planes they are represented as lines.

The *"shipboard"* projection is bounded from above with a section of a deck in the axis plane (CL) and side line, on the front with a stem, on the back with a sternpost and from below with a keel line. The projection of a side line is also depicted on a theoretical plane of the main frame, i.e. on the *hull*. On projections of a lines plan there is also usually depicted the shape of a raised forebody and stern, bulwark and often superstructure if they have a complex shape, like for example on lines plans of high-speed ships, etc.

Lines plan of a hull (Fig. 3) is needed for calculation of navigation properties of a ship, such as buoyancy, stability, and unsinkability during the lines plans processing as well as ship's hull construction.

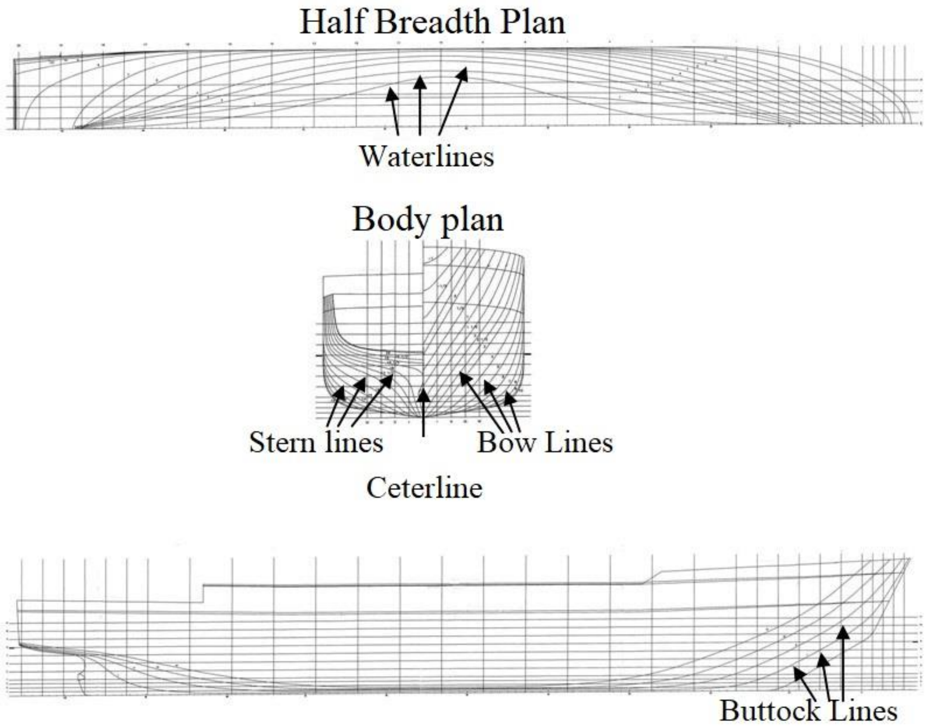

Fig. 3. Lines Plan [https://officerofthewatch.com/2012/02/09/ships-geometry-and-hull-definition/]

The lines plan then determines the shape of girders as well as the configuration of individual hull's plating sheets. A lines plan with scale of 1:1 is often drawn in a special shed of a shipyard called a drawing office for this purpose. In this shed there are also made templates which are used to check shapes of individual construction units. Currently it is realised by means of automatic devices and computers.

The development of a lines plan usually starts with a deposition (creation) of a so called net. There on the half breadth and shipboard projection planes, the designed length L is divided into 20 equal segments using 21 lines - frames (with frame spacing $\Delta L = L/20$). The draught T, or the line segment which represents it between a construction water line and base plane, is there on projection planes of a shipboard and hull divided into at least 4 equal segments and 5 equally distanced water lines are drawn (with water line spacing $\Delta T = T/4$).

The line segment between the axis plane and the right and left side on projection planes of a hull and half breadth is then divided into at least 3 equal segments and curves of longitudinal section lines I, II and III are drawn (with frame spacing $\Delta B = 0{,}5/3$). In case of ships of a complex shape the number of water lines and longitudinal section lines increases especially in forebody and stern parts of a ship; so called half, quarter frames, etc. Frames and water lines are usually marked with Arabic numerals; in our country there has been a convention to mark frames with numerals and water lines with letters.

The width of a lines plan must be defined with rectangles of sides *BxD* on the hull; *LxD* on the side and *LxB* on the half breadth. Then on each projection line of a hull, half breadth and shipboard there are curves representing frames, water lines and longitudinal section lines deposited.

Points of intersection of the above mentioned curves with corresponding line segments of the net are then approved on all three projections in compliance with construction geometry principles.

2.1.1. Characteristics of Buoyancy and Conditions of Balance of a Floating Ship

The ability of a ship to float in required equilibrium with a given load is called *buoyancy*. The term equilibrium (ship seating) means its position in relation to the water surface. The change of the axis plane (CL) position of a ship in relation to water surface, in other words *heeling* (Fig. 4) is given with the angle θ (theta) between a line and a heeled water line on the projection plane of the hull. Heeling to the starboard side is considered positive, heeling to the port side is considered negative.

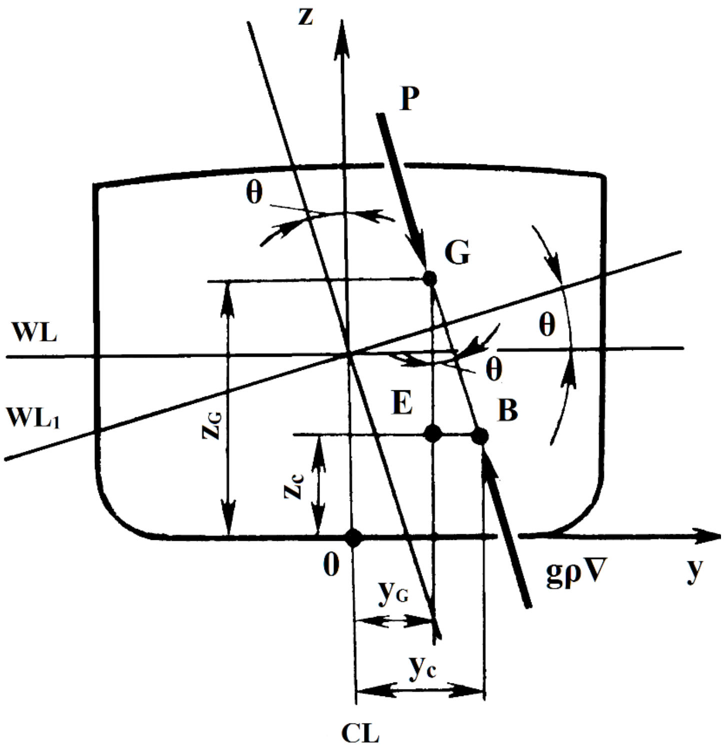

Fig. 4. Heeling [Authors, 2]

The change of a main frame plane position of a ship in relation to water surface or the heeling of a ship in a longitudinal direction is called *inclination* (Fig. 5). The inclination is given with the angle Ψ (psi) between a line and a heeled water line on a projection plane of a shipboard, or with the linear scale factor Δ (delta) which equals the difference of forebody and stern draughts. Inclination to the forebody is positive, inclination to the stern is negative. There exists a dependency between the inclination Δ and the angle of inclination *Ψ*: $\Delta = L.tg\psi \approx L.\psi$.

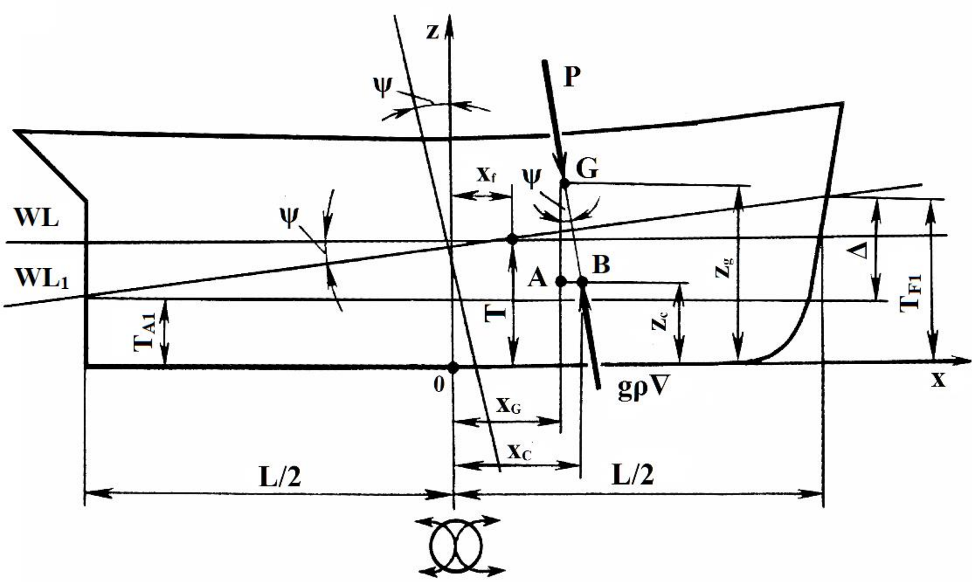

Fig. 5. Inclination [Authors, 2]

State of equilibrium (seating) of a ship may be set with three values:

- midship draught,
- heeling angle θ,
- inclination angle ψ or inclination Δ.

At free floating in tranquil waters there act gravitation forces and hydrostatic pressure forces of water on a ship. The resultant of gravity forces (often called *weight of the ship*) acts in the centre of gravity of the ship (point *G*). The vertical component of compensating forces of hydrostatic pressure is the force of buoyancy - lifting force. Horizontal components attached to each ship's side are in a mutual balance and are not taken into account in the ship theory.

The volume of the immersed hull part $V = \nabla$ [m^3] is called a *volume displacement*. The force acting in the centre of volume displacement is marked as

F_B. The force of buoyancy has its point of action in the point B - *in the centre of volume displacement*.

For the state of equilibrium of the floating ship it is necessary that the weight of the ship $P(N)$ is equal to the force of buoyancy $V.\rho.g$, and that the centre of weight and the centre of buoyancy lie on one vertical (ρ - water density [$t.m^{-3}$], g - gravity acceleration [$m.s^{-2}$]).

If we choose a coordinate system with x, y, and z axes representing lines of the intersection of characteristic planes CL, BL and $\otimes$ and if we mark the centres of gravity as x_C, y_C, z_C, we can write the following three equations - equilibrium conditions:

$$P = \rho.V.g \, , \tag{8}$$

$$tg\theta = \frac{y_C - y_G}{z_G - z_C} \, , \tag{9}$$

$$tg\psi = \frac{x_C - x_G}{z_G - z_C} \, . \tag{10}$$

The equations for $tg\theta$ and $tg\Psi$ express conditions for the calculation of the ship's centre of gravity G and centre of buoyancy B. While the first force equation expresses the relation between the weight and buoyancy size, the other two equations express the second part of the Archimedes' principle, i.e. the centre of gravity and buoyancy must lie on one vertical.

If the heeling and the inclination equal 0 (i.e. $\theta=0$ and $\Psi=0$), i.e. the ship floats on an *even keel*, then the following holds true:

$$P = \rho.V.g \, , \tag{11}$$

$$x_C = x_G \, , \tag{12}$$

$$y_C = y_G = 0. \tag{13}$$

If we substitute the value of the volume displacement into the force formula, we will get the equation of buoyancy.

$$P = \rho . C_B . L . B . T . g \tag{14}$$

The expression $\rho . V = \rho . C_B . L . B . T = \Delta$ has a value equal to the weight which would coincide with volume of water determined with the immersed part of the ship's hull. It is called a weight displacement (deadweight), i.e. $\Delta = P/\rho$. That means that the weight displacement equals the ship's weight P/ρ. From theoretical mechanics it is known that the centre of a body's weight is identical with the centre of gravity of the body's weight. That means that the ship's centre of gravity is also identical with the centre of its weight.

The equations of equilibrium conditions contain ten unknown values. For them to be solved some values are required to be defined in advance. To find out the ship's weight it is necessary to calculate its displacement since $P = g . D$.

The calculation of the volume displacement and coordinates of the centre F_B is done on the lines plan, most frequently using a table method.

2.1.2. Calculation of the Weight Displacement and the Centre of Gravity of the Ship

Since the ship's weight during static navigation always equals the weight displacement, it can be calculated as a sum of ship's individual parts' weights.

$$D_L = \sum_{i=1}^{n} m_i \tag{15}$$

Here m_i stands for the weight of an element of the hull, superstructures, machines, devices, systems, etc., which are determined on the base of ship's plans, mainly the general plan, construction designs, ship's installation catalogues, etc. Together with determination of the weight the plans also define centres of individual partial masses x_{gi}, y_{gi}, z_{gi} representing individual ship's installations. Then coordinates of the centre of gravity of the ship *G* as a whole are:

$$x_g = \frac{\sum m_i . x_{gi}}{\sum m_i}, \quad y_g = \frac{\sum m_i . y_{gi}}{\sum m_i}, \quad z_g = \frac{\sum m_i . z_{gi}}{\sum m_i}. \qquad (16, 17, 18)$$

The calculation of the weight is done by individual subgroups sorted into a so called book of weights.

Table 2

Book of Weights

Weight Item	**Weight m_i [t]**	**Lever Arms**		**Static Moments**	
		from ⊗ x_{gi}	from CL z_{gi}	$m_i\ x_{gi}$	$m_i\ z_{gi}$
I (A) Ship and its Installations 1. Material of the hull and superstructures 2. Inboard skeleton 3. Rooms equipment 4. Coatings and insulations 5. Fixed fitting 6. Ship's installations 7. Deck mechanisms 8. Ship systems 9. Electrical equipment and radio devices 10. Stock and equipment of a ship					
∑ of the ship's hull	$\mathbf{m_t}$	$\mathbf{x_{gt}}$	$\mathbf{z_{gt}}$	$\mathbf{M_{xt}}$	$\mathbf{M_{zt}}$
II (B) Engine Room 11. Main engines 12. Propulsion device and line shafting 13. Auxiliary mechanisms 14. Engine room lines 15. Service parts and engine inventory 16. Water, fuel, oil in the line system					
∑ of the engine room	$\mathbf{m_s}$	$\mathbf{x_{gs}}$	$\mathbf{z_{gs}}$	$\mathbf{M_{xs}}$	$\mathbf{M_{zs}}$
∑ of the empty ship	$\mathbf{D_0}$	$\mathbf{x_{g0}}$	$\mathbf{z_{g0}}$	$\mathbf{M_{x0}}$	$\mathbf{M_{z0}}$

Table 2 (continued)

Book of Weights

Weight Item	**Weight m_i [t]**	**Lever Arms**		**Static Moments**	
		from ⊗ x_{gi}	from CL z_{gi}	$m_i\ x_{gi}$	$m_i\ z_{gi}$
III (C) Useful Ballast 17. Fuel and oil 18. Crew 19. Passengers 20. Cargo 21. Water 22. Foodstuffs 23. Polluted water					
∑ of useful ballast	$\mathbf{D_W}$	$\mathbf{x_{gd}}$	$\mathbf{z_{gd}}$	$\mathbf{M_{xd}}$	$\mathbf{M_{zd}}$
∑ of ship at full load	$\mathbf{D_L}$	$\mathbf{x_g}$	$\mathbf{z_g}$	$\mathbf{M_x}$	$\mathbf{M_z}$

Source: Authors

In principle there are the following types of the weight displacement:

1. unloaded displacement D_0 (Δ_0) = weight of the completed ship in an operating state (e.g. in a shipyard at its delivery);

2. fully loaded displacement $D_L = D_0 + D_W$, where D_W - the weight of useful ballast: passengers and crew, foodstuffs, supplies of fuel and oil, drinking water, dirty water in the hull and bottom, consumption or variable weights. Part of the deadweight = weight of a useful cargo G is useful (net) capacity of the ship;

3. displacement at partial ship load, e.g. in case of a ship without a load m_0 with the crew and/or passengers. With full or partial supplies of fuel, oil and other stock items. A loaded ship, however, with minimum fuel, oil and stock supplies (e.g. for one day navigation), etc.

2.1.3. Calculation of Water Lines and Frames Elements

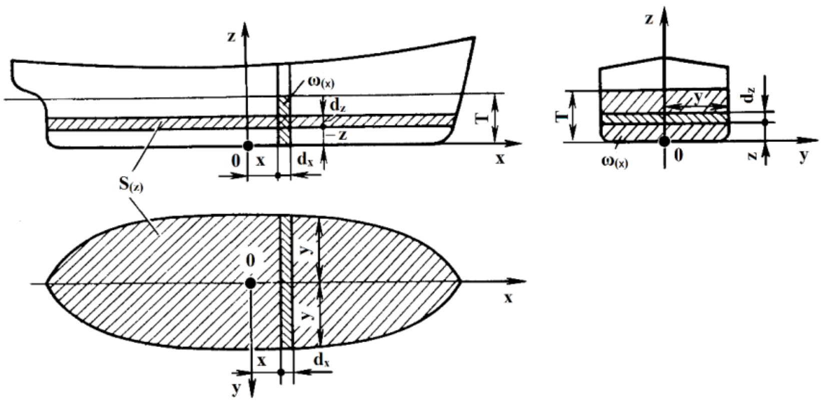

Fig. 6. Representation of Water Lines and Section Frame Lines for Volume Displacement Determination [Authors, 13]

In concordance with the sketch (Fig. 6) the volume displacement V = ∇, when the ship floats on an even keel with the draught *T*, is determined using the following integral formulae:

$$\nabla = \int_{-L/2}^{L/2} \varpi . dx, \tag{19}$$

$$\nabla = \int_{0}^{T} s . dz. \tag{20}$$

The coordinates of the immersed volume centre can be calculated as follows:

$$x_C = \frac{\int_{-L/2}^{L/2} \omega . x . dx}{V}, \tag{21}$$

$$x_C = \frac{\int_{0}^{T} s . z . dz}{V}, \tag{22}$$

where:

ω and *s* are immersed areas of the frame and water line in the *x* and *z* distance from the main frame and basic plane.

The **surface area of a water line** can be expressed with a definite integral:

$$S = 2.\int_{-L/2}^{L/2} y.dx \qquad (23)$$

The centre of this area is located in the x_f distance from the main frame plane and it is determined with the following expression:

$$x_f = \frac{M_{oy}}{S} = \frac{2.\int_{-L/2}^{L/2} y.x.dx}{2.\int_{-L/2}^{L/2} y.dx} = \frac{\int_{-L/2}^{L/2} y.x.dx}{\int_{-L/2}^{L/2} y.dx}, \qquad (24)$$

where:

M_{oy} - static moment of a water line surface area,

S - surface area of a water line.

The dependency *y = f (x)* determines the coordinates of the water line. This curve is specified with the lines plan, and integrals are calculated using approximating methods known from mathematics, such as the trapezoidal rule, Simpson's rule, Chebyshev's theorem, etc.

Since the lines plan is evenly divided, the most frequent manual calculation method is the trapezoidal rule. Each segment of a water line between two adjacent frames sections is substituted with a line which then leads to a trapezoid with the height *ΔL*. The sum of surfaces of all trapezoids is approximately equal to the water line surface.

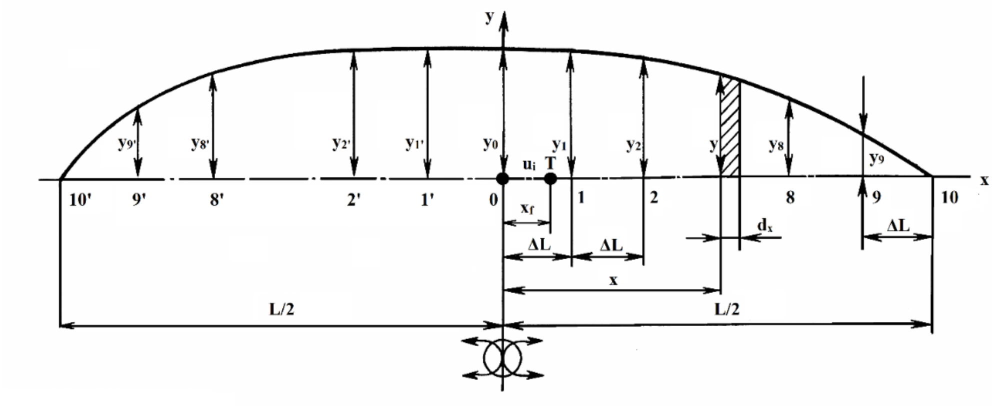

Fig. 7. Integration Method of a Water Line for Determination of its Surface and Centre of Gravity [Authors, 13]

$$S = 2.\left(\frac{y_{10}^{'}+y_9^{'}}{2}.\Delta L + \frac{y_9^{'}+y_8^{'}}{2}.\Delta L+\ldots..+\frac{y_8+y_9}{2}.\Delta L + \frac{y_9+y_{10}}{2}.\Delta L\right). \quad (25)$$

or in general:

$$S = 2.\Delta L\left(\sum_{i_A}^{i_F} y_i - \frac{y_A+y_F}{2}\right) = 2.\Delta L\sum_{i_A}^{i_F} corect.y_i \quad (26)$$

where:

$y_A^{'}$ and $i_A^{'}$ - the end corrected coordinate of the afterbody and its number,

$y_F^{'}$ and $i_F^{'}$ - the end corrected coordinate of the forebody and its number.

A correction is a value ε which equals the half sum of end (often corrected) coordinates.

$$S = \varepsilon = \frac{1}{2}(y_A' + y_F') \quad (27)$$

The difference between the full sum of coordinates and the correction is called a corrected sum of coordinates:

$$\sum_{i_A}^{i_F} corect.y_i = \sum_{i_A}^{i_F} y_i - \varepsilon \quad (28)$$

Numeration of coordinates is in concordance with the plan; forebody part 0 - 10 and afterbody part 0 - 1′ to 10′.

The static moment of a water line surface will then be as follows:

$$M_{0g} = 2.\int_{-L/2}^{L/2} x.y.dx = 2.\int_{-L/2}^{L/2} u.dx = 2.\Delta L\left(\sum_{i'_A}^{i_F} u_i - \frac{u'_A - u'_F}{2}\right). \quad (29)$$

The values u_i and u_i' (products of y_i coordinates in their distance from $\otimes$ $x_i = \pm i.\Delta L$) can also be presented as $u_i = i.\Delta L.y_i; u_i = -i'.\Delta L.y_i'$ and then the static moment will be:

$$M_{0g} = 2.\Delta L\Big[0 + \Delta L(y_1 + y_1') + \Delta L(2y_2 - 2y_2') + \ldots + \Delta L(i_{\check{c}}.y_{\check{c}} - -i_{k'}.y_{k'}) - \frac{i_{\check{c}}.y_{\check{c}}.\Delta L - i_{k'}.y_{k'}.\Delta L}{2}\Big], \quad (30)$$

or in general:

$$M_{0g} = 2\Delta L^2\left[\sum_{i_A}^{i_F}(i.y_i - i'.y_{i'}) - \frac{i_F.y_F.\Delta L - i_{A'}.y_{F'}.\Delta L}{2}\right]. \quad (31)$$

Then the final expression for the centre of water line will be as follows:

$$x_f = \Delta L.\frac{\sum_{i'_A}^{i_F}(i.y_i - i'.y_{i'}) - \frac{i_F.y_F.\Delta L - i_{A'}.y_{F'}.\Delta L}{2}}{\sum_{i'_A}^{i_F} y_i - \frac{y_F - y_{A'}}{2}}. \quad (32)$$

When calculating the stability there are also moments of inertia of water line surface with regard to the *x* and *y* axes important; according to theoretical mechanics rules these are as follows:

$$I_x = \int_{-L/2}^{L/2}\frac{(2.y)^3.dx}{12} = \frac{2}{3}\int_{-L/2}^{L/2} y.x^2.dx \quad . \quad (33)$$

For the main central moment of inertia of the water line surface with regard to the transverse axis passing through its centre the following expression is true:

$$I_f = I_y - S.x_f^2 \quad . \quad (34)$$

Using the rectangular rule for moments of inertia it is possible to write the following:

$$I_x = \frac{2}{3}.\Delta L\left(\sum_{i'_A}^{i_F} y_i^3 - \frac{y_F^3 - y_{A'}^3}{2}\right), \tag{35}$$

$$I_y = 2.\Delta L^3\left[\sum_{i'_A}^{i_F}\left(i^2.y_i + i'^2.y_{i'}\right) - \frac{i_F^2.y_F + i'^2_{A'}.y_{A'}}{2}\right]. \tag{36}$$

The calculation of geometrical characteristics of water lines is done using the table method; the number of tables then equals the number of water lines.

Table 4

Calculation of Geometrical Characteristics of Water Lines

The number of the main frame towards the forebody i and the afterbody i´	**i [-i´]**	**i^2 = $[-i´]^2$**	**y-coordinates on the i and i´** y_i	$y_{i´}$	**$i.y_i$ [m]**	**$\lvert i´\rvert.y_{i´}$ [m]**	**$i^2.y_i$ [m]**	**$\lvert i´\rvert^2.y_{i´}$ [m]**	**y_i^3 [m^3]**	**$y_{i´}^3$ [m^3]**	**Charact. of a Water Line x_{fi}; S_j; I_{fj}; I_{xj}**
1	2	3	4	5	6	7	8	9	10	11	12
0,⊗	0	0	y_0	-							
1, 1´	1	1	y_1	$y_{1´}$							
2, 2´	2	4	y_2	$y_{2´}$							
. . .											
9, 9´	9	81	y_9	$y_{9´}$	$9.\ y_9$	$9.\ y_{9´}$	$81.\ y_9$	$81.\ y_{9´}$	y_9^3	$y_{9´}^3$	
10, 10´	10	100	y_{10}	$y_{10´}$	$10.\ y_{10}$	$10.\ y_{10´}$	$100.\ y_{10}$	$100.y_{10´}$	y_{10}^3	$y_{10´}^3$	
Sums of columns $\sum´_k$	X	X	$\sum´_4$	$\sum´_5$	$\sum´_6$	$\sum´_7$	$\sum´_8$	$\sum´_9$	$\sum´_{10}$	$\sum´_4$	
ε_k = 1/2 of end (bottom) number of	X	X	$y_F/2$	$y_{A´}/2$	$i_F.y_F/2$	$i_{F´}.y_{A´}/2$	$i_F^2.y_F/2$	$(i_{A´})^2.y_{A´}/2$	$y_F^3/2$	$y_{A´}^3/2$	
Correcting sums $\sum_k = \sum´_k - \varepsilon_k$ where k = 4 to 11	X	X	$\sum_4$	$\sum_5$	$\sum_6$	$\sum_7$	$\sum_8$	$\sum_9$	$\sum_{10}$	$\sum_{11}$	

Source: [Authors, 2]

Coordinates of every jth water line (forebody's y_i and afterbody's $y_{i'}$) are scanned from the lines plan, multiplied with its measure and put into the table in [m] with the precision of two decimals. To increase the precision of the calculation the end y_F and $y_{A'}$ coordinates are usually taken not equal, but corrected in concordance with the following figures.

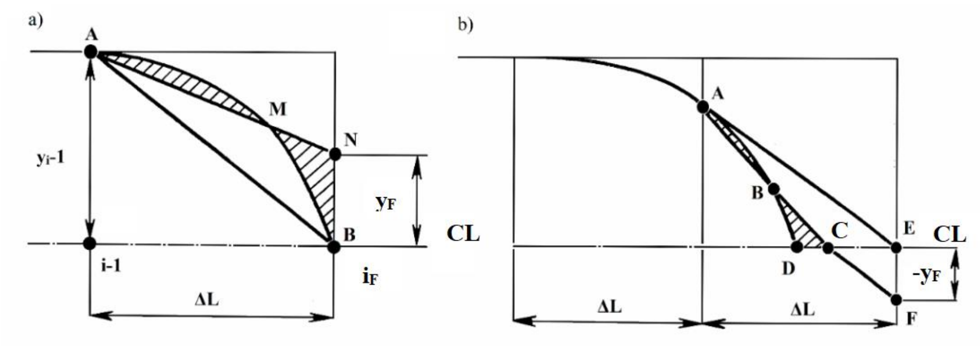

Fig. 8. Ways of Coordinates Corrections [Authors, 2]

If end coordinates are not available, then the relevant box in the table will be struck through.

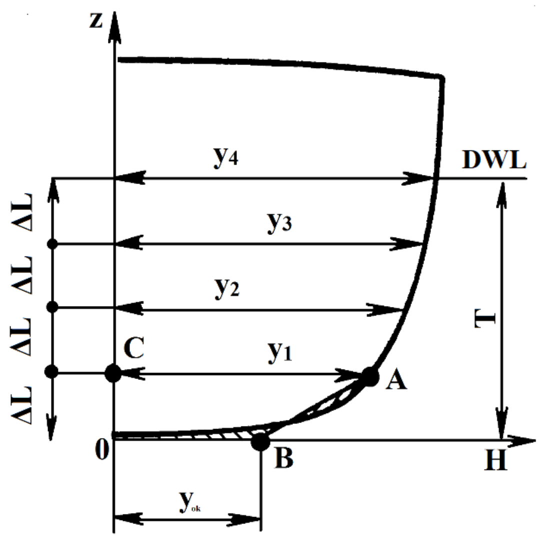

Fig. 9. End (Calculating) Coordinate [Authors, 2]

The area of an immersed frame can be found using the expression:

$$\omega_i = 2\int_0^T y.dz \quad . \tag{37}$$

In case of using the trapezoidal rule:

$$\omega_i = 2.\Delta T \left[\sum_{j=0}^{n} y_{ij} - \frac{(y_{kr}+y_n)}{2}\right], \tag{38}$$

where:

ΔT - distance between water lines,
y_{ij} - coordinate of the ith frame on the jth water line,
j - number of a water line j = 0, 1, 2... ω.

To correct the end coordinate, please see Fig. 9.

2.1.4. Calculation of the Volume Displacement and its Coordinates, Curve of Buoyancy and Initial Stability

For the functions *ω (x)* and *S (z)* visualisation sake they may be graphically represented on the basis of an approximate calculation of frames areas ω_i and water lines surfaces S_j. The curve $\omega = \omega (x)$ which shows the change of frames area on the ship's length is called a **curve of buoyancy according to sectional areas** (Figure 10).

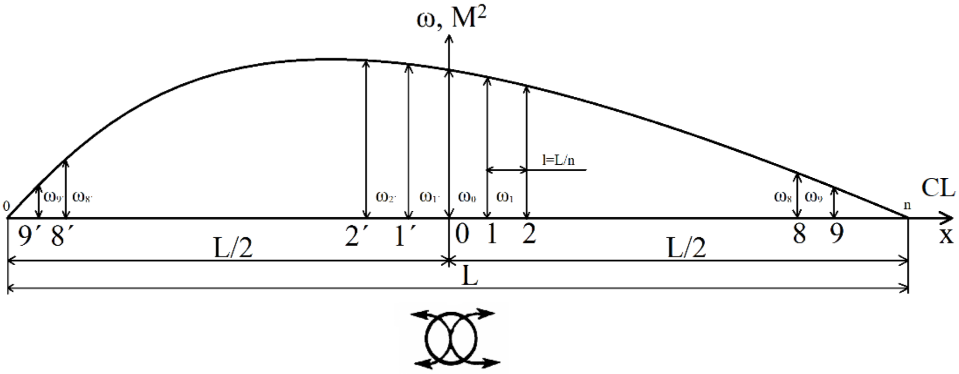

Fig. 10. Curve of Buoyancy according to Sectional Areas [Authors, 3]

The curve $S = S\ (z)$ which expresses the dependency of water lines surfaces on the draught is called a **curve of vertical displacement distribution** (Figure 11).

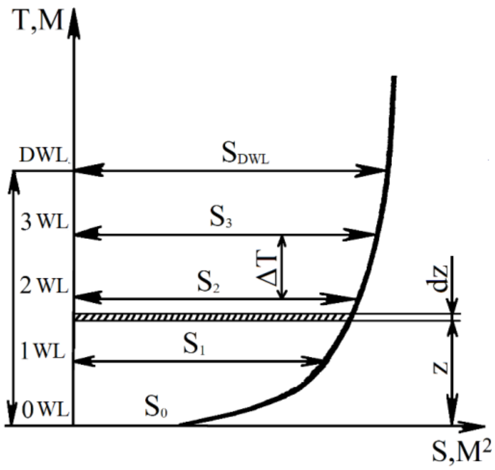

Fig. 11. Curve of Vertical Displacement Distribution [Authors, 3]

The area F_ω bounded with the curve of buoyancy according to sectional areas and the area F_s bounded with the curve of vertical displacement distribution and corresponding coordinate axes are numerically equal to the volume displacement, and the coordinates of the centre of these areas are corresponding coordinates of the displacement's centre x_c and z_c.

The area fullness coefficient of buoyancy according to sectional areas is equal to the coefficient of longitudinal fullness of the ship.

$$C_{p_\omega} = \frac{F_\omega}{L\omega_\otimes} = \frac{V}{L\omega_\otimes} = \frac{C_B.L.B.T}{L.B.T.C_M} = \frac{C_B}{C_M} = C_p \tag{39}$$

Exactly this way we can prove that the area fullness coefficient of a vertical displacement distribution is equal to the vertical prismatic coefficient C_{VP}.

If we use the trapezoidal rule to calculate the displacement from water lines surfaces, then the ship's displacement up to the jth water line is given as a sum of trapezoids inscribed into curves of vertical water lines distribution, or after modifications:

$$\nabla_j = \frac{\Delta T.(S_0 + 2.S_1 + 2.S_2 + \dots + 2.S_{j-1} + S_j)}{2}. \tag{40}$$

The expression in parentheses is called an integral sum, and then the shortened expression for the displacement may be written as follows:

$$\nabla_j = \Delta T \left(\sum_{j=0}^{n} S_j - \frac{S_0 + S_n}{2} \right). \tag{41}$$

The vertical coordinate of the centre of buoyancy z_c can be calculated using the trapezoidal rule as follows:

$$z_{cj} = \Delta T \frac{[S_0.0 + 2.S_1.1 + 2.S_2.2 + \dots + 2.S_{j-1}(j-1) + S_j.j]}{(S_0 + 2.S_1 + 2.S_2 + \dots + 2.S_{j-1} + S_j)}. \tag{42}$$

To get a higher precision the vertical coordinate of the displacement for the first water line is determined as a centre of the trapezoid with sides S_{okor} and S_1 and the height ΔT using the following formula:

$$z_{c1} = \frac{1}{3} \Delta T. \frac{S_{0ok} + 2.S_1}{S_{0ok} + S_1}, \tag{43}$$

where:

S_{0ok} is a corrected length of the zeroth water line surface from the diagram of the altitudinal distribution of displacement.

To calculate the altitudinal coordinate of the displacement x_c at a ship's draught $T=Z$ it is useful to apply integration not by a frame at the length x, however, by a quotient of the water line's static moment at the height z. Then the x_c coordinate will be as follows:

$$x_{cj} = \frac{\int_0^z S.x_f dz}{\int_0^z S dz}, \qquad (44)$$

or using the trapezoidal rule:

$$x_{cj} = \frac{S_{0kor}.x_{f0} + 2S_1.x_{f1} + ... + 2S_{j-1}x_{f_{j-1}} + S_j.x_{f_j}}{S_{0kor} + 2S_1 + ... + 2S_{j-1} + S_j}. \qquad (45)$$

Formulae for V_j, z_{cj} and x_{cj} allow for approximating the integrals with a variable upper limit using the trapezoidal rule. The variable upper limit means that the values V_j, z_{cj} and x_{cj} are not calculated for a specific water line, but for each water line one by one starting from the zeroth water line and ending with any jth one. The calculation of the volume displacement and coordinates of the displacement using the trapezoidal rule is done using a table method.

Except for displacement and displacement coordinates the computing tables also contain metacentric radiuses r and R which will be discussed later. Next there are values of displacement coefficients C_B and water line coefficient C_ω.

Results of table calculations have the form of the following functional dependencies on the height (of the draught):

$V(z)$; $S(z)$; $z_C(z)$; $x_C(z)$; $x_f(z)$; $r(z)$; $R(z)$; $C_B(z)$; $C_\omega(z)$; $I_x(z)$; $I_f(z)$ which may be presented in a diagram of so called **curves of buoyancy and initial stability**.

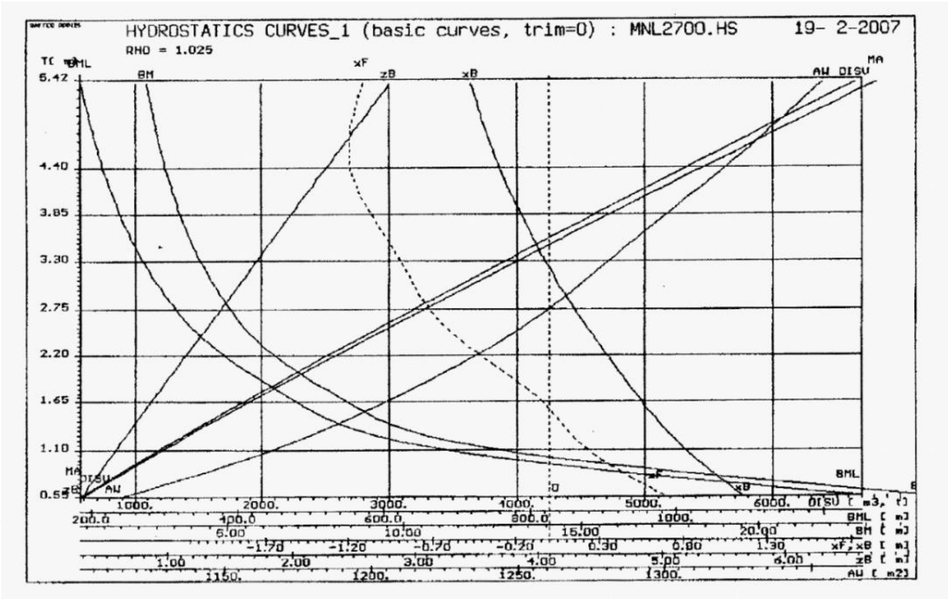

Fig. 12. Diagram of Curves of Buoyancy and Initial Stability MNL 2700 [6]

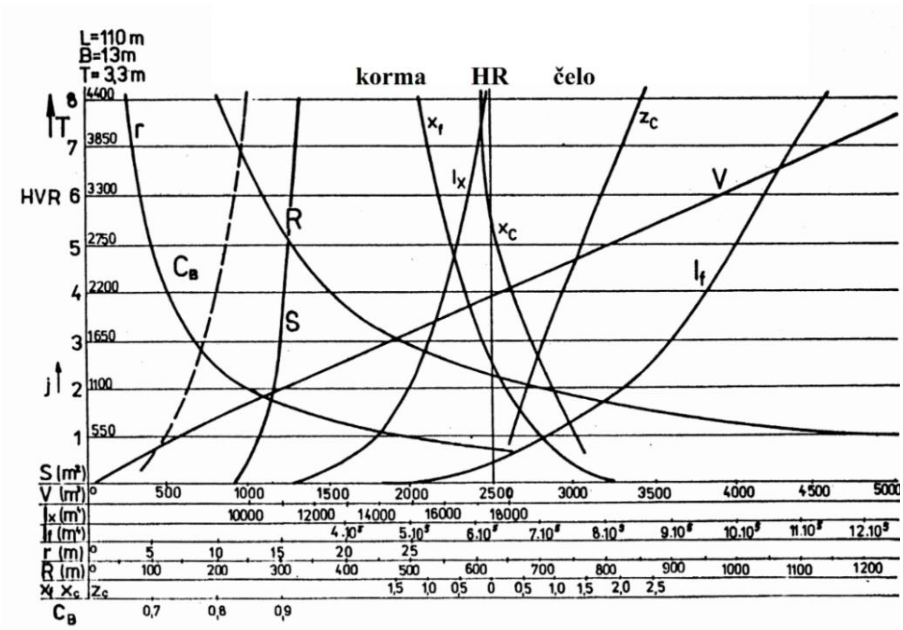

Fig. 13. Calculating Diagram of Buoyancy and Initial Stability [6]

Curves of buoyancy and initial stability allow for an easy determination of characteristics at any draught (water line) on an even keel *(T = z).*

If a ship floats with a trim, the volume displacement is also calculated by means of integration by the trapezoidal rule.

$$V = \nabla \int_{-L/2}^{L/2} \omega . dx \approx \Delta L \left(\sum_{i=-10'}^{i=10} \omega . i - \frac{\omega_{10}+\omega_{10}}{2} \right) \quad (46)$$

In this case it is necessary to calculate and construct a diagram of dependency of the immersed area change $\omega_i = \omega_i(z)$ on the draught individually for each frame.

Such a summary diagram is called Bonjean diagram, named after a French engineer Bonjean who suggested it for such calculations.

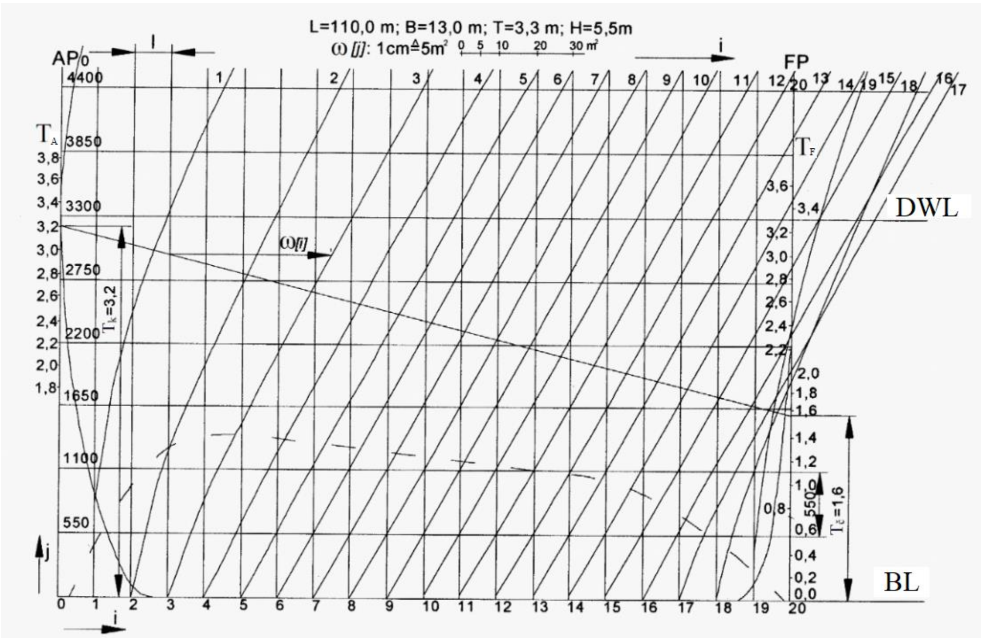

Fig. 14. Bonjean Diagram of Frames Areas MNL 2700 [6]

To construct this diagram frames areas are calculated using a known relationship.

$$\omega_i = 2. \Delta T \left[\sum_{i=0}^{\omega} . y_{ij} - \frac{(y_F+y_n)}{2} \right] \quad (47)$$

According to the known values of the afterbody's and forebody's draught the inclined water line is marked into the Bonjean diagram and values ω_i of frames areas immersed up to this water line are subtracted. Knowing the value ω_i the displacement *V* is calculated as it has already been mentioned for the inclined water line, and the displacement x_c-coordinate is calculated as follows:

$$x_c = \Delta L \frac{\left[\sum_{i=10'}^{i=10} i(\omega_i - \omega_{i'}) - \frac{10.\omega_{10} - 10.\omega_{10'}}{2}\right]}{\left(\sum_{i=10'}^{i=10} \omega_i - \frac{\omega_{10} + \omega_{10'}}{2}\right)} . \qquad (48)$$

Similar dependencies can be calculated in advance and marked into a so called Diagram of Firsov (Fig. 15) in which the T_F and T_A axes display curves of fixed values V, x_c, $z_m = r + z_c$.

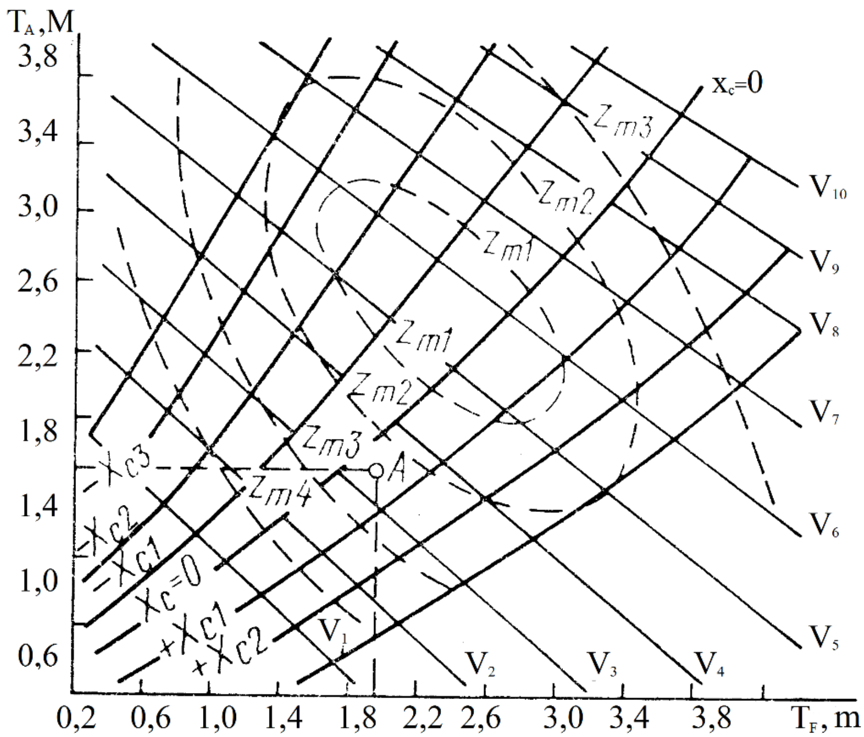

Fig. 15. Diagram of Firsov [6]

According to given draughts the corresponding values are determined using the method of interpolation. In this way according to given draughts T_F and T_A the point *A* is determined, and using the interpolation values *V*, x_c and z_m are determined.

2.1.5. Determination of a Ship's Draught at Loading and Unloading a Small and Big Cargo - the Check And Standardisation of Buoyancy

The term small load means a load for which after loading or unloading, i.e. with the change of a water line its area will be only slightly different from the initial one (the difference of water lines can be disregarded).

For example if we allow the difference in water line surface to be 2 % - 2.5 %, then the small load represents accepting the weight from 20 % - 25 % of the weight at initial displacement for engine cargo ships and cargo ships. In case of passenger ships and pusher tugs the small load means values 10 % - 15 % of the weight displacement.

The information mentioned above implies that methods used for the change of a small load predominate in calculations of a majority of practical cases.

Thus in order for a ship to achieve only a draught change ΔT after receiving or removing a small load, the centre x_z of this load must lie on a perpendicular passing through the centre of displacement's additional volume. This means the longitudinal coordinate of the load must be equal to the longitudinal coordinate of the initial water line surface x_f, i.e. $x_z = x_f$ (Fig. 16).

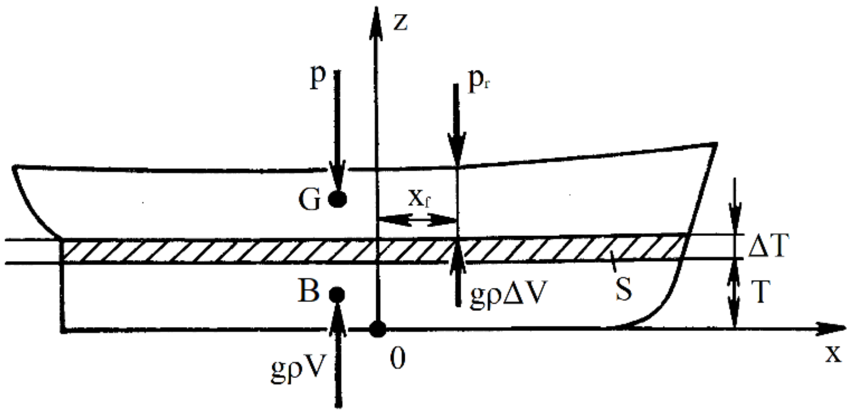

Fig. 16. Distribution of Forces at Receiving a Small Load [2]

If the weight of a small load is marked as m_z and the change of the weight displacement is marked as ΔD, then:

$$m_z = \Delta D. \tag{49}$$

If the supplemental displacement or supplemental hull volume immersed into water is marked as ΔV, we will get $x_z = x_f$ and then

$$m_z = \Delta D = \rho.\Delta\nabla = \rho.\Delta T.S, \tag{50}$$

where:

S represents the surface area of the initial water line.

The character + marks the reception of a load, and the character - marks the removal of a load from the ship.

The loading tonnage per 1 cm of draught, which is marked as q $[t.cm^{-1}]$, will be:

$$q = \frac{\rho.S}{100}. \tag{51}$$

If we know the value of q it is easy to determine the change of the draught after the reception or after the removal of the load using the following expression.

$$\Delta T = \pm \frac{\Delta D}{q} \tag{52}$$

The change of the draught after the reception or after the removal of a big load is determined using a "principal dimension" or more frequently using a loading scale.

Loading scale is a curve showing the dependency of the weight displacement on the draught:

$$D = \rho.V = f(z). \tag{53}$$

In case the ship floats in fresh (river) water, where $\rho = 1$, this curve is identical with the curve of dependency of the volume displacement on the draught, i.e. $V = f(z)$.

Loading scale is often introduced in the form of a monogram which functionally binds the weight displacement, draught, ballast (deadweight), ship's height above water and tonnage appertaining to 1 cm of draught with each other (Fig. 17).

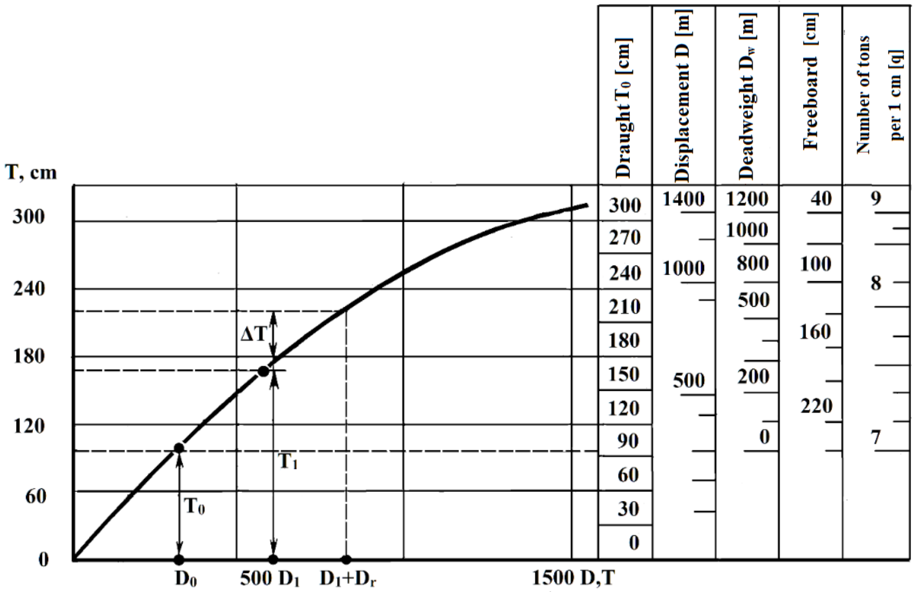

Fig. 17. Monogram of a Loading Scale [Authors, 2]

For the specified mean midship draught - in case the ship is without any inclination (trim), the loading scale can be used to determine the weight displacement, the ballast D_W and the tonnage per 1 cm, i.e. q. To check the draught there is placed a draught scale with 5 cm division on sides of the bow (forebody) and stern as well as in the middle of the ship.

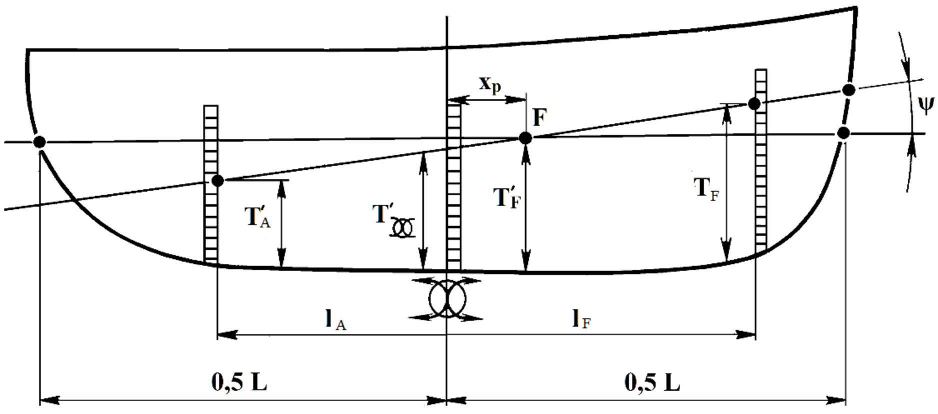

Fig. 18. Scale of Draughts (Dipping)

Then it is true:

$$T_f = T'_A + \psi.\left(l_A + x_f\right) = T'_A + \frac{T'_F + T'_A}{l_F + l_A}.\left(l_A + x_f\right) = \frac{T'_F.(l_A + x_f) - T'_A.(l_F + x_f)}{l_F + l_A}, \quad (54)$$

where:

T_F and T_A - are draughts read from the forebody and stern scale,

l_F and l_A - are distances between the main frame and the forebody and stern draught scale,

$x_f = f(T_{mid}) \cong \frac{T'_F.l_A + T'_A.l_F}{l_F + l_A}$ - can be determined by curves of the lines plan at T_{mid}.

Each ship must have a certain **reserve of buoyancy** which can be determined by means of the volume of the waterproof hull part above water in percentage out of the displacement. This reserve is needed for the safety of the navigation, to ensure ship's buoyancy at its overload, to ensure unsinkability in case of the hull's perforation, etc. Depending on the hull and the ship determination the reserve of buoyancy changes in a wide range and represents 10 % - 100 % out of the full displacement.

The reserve of buoyancy depends on the height of a **freeboard** which is standardised per the classification society rules. The height of each ship's freeboard is determined on the base of assessing its strength and unsinkability depending on the hull length and the ship class. Moreover the classification society also standardises the height of the side sheer, raised forward sheer and after sheer (superstructures), loading hatch moulding and other equipment ensuring non-flooding of the ship's hull.

The tolerable height of a freeboard is inspected according to specialised load lines which are put (welded) onto the midship length (on the main frame) on both sides of the ship. For ships with side wheels there are two lines put on each ship's side in approximately 1/3 of the ship's length from the forebody and the stern. When marking the load line away from the main frame the sheer of the deck must be taken into account.

The load line represents a circle with a horizontal line through it to show the allowable draught.

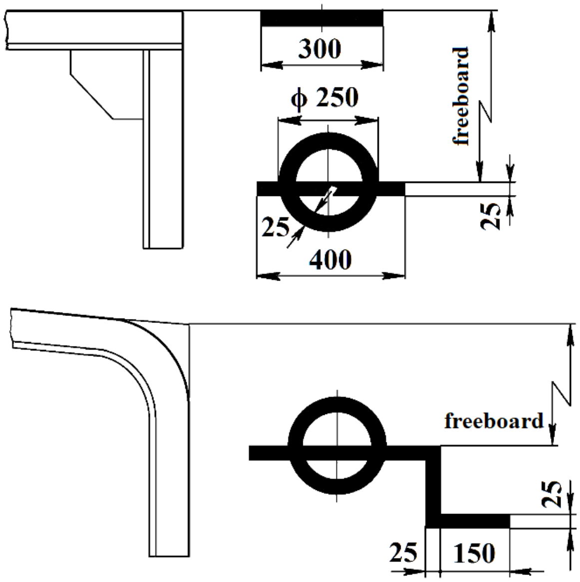

Fig. 19. Load Line [Authors, Slovenský Lloyd]

A pattern of the line according to the relevant classification society for a sea ship can be found in Fig. 20.

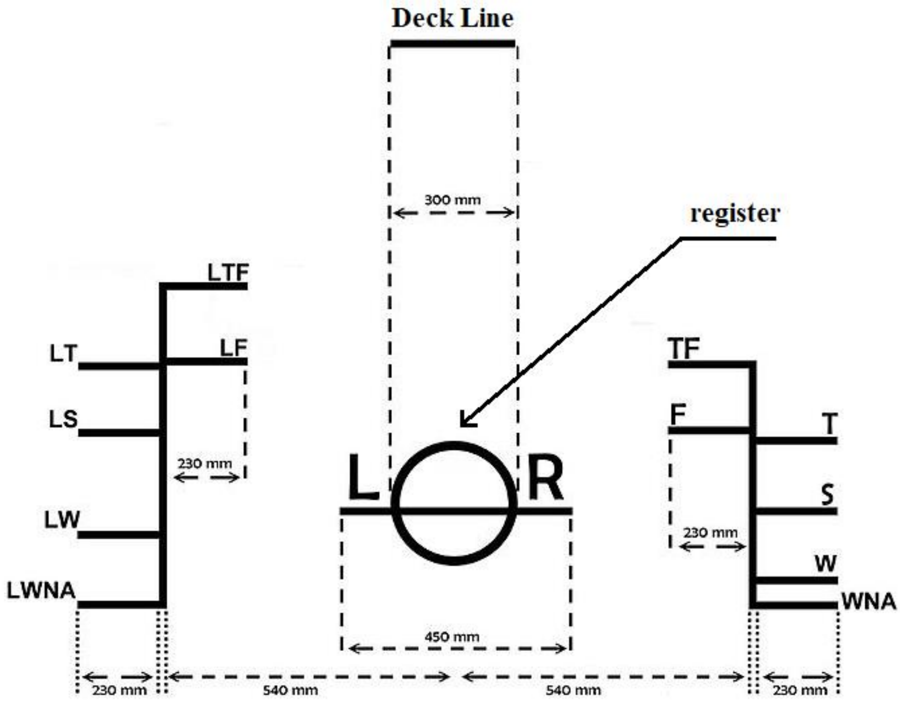

Fig. 20. Load Line of a Sea Ship [https://www.marineinsight.com/marine-navigation/introduction-ship-load-lines/]

(S - Summer, T - Tropical, W - Winter, WNA - Winter North Atlantic, F - Fresh Water, TF - Tropical Fresh Water)

On the side of the line there are classes of the ship labelled with relevant lines of the allowable draught. The ship's freeboard is indicated from the allowable draught line up to the joint of the deck and the side.

The load line on steel ships is realised either through welding strips, protruding welds, or it is marked with a punch and coated with a paint contrasting with the undercoat.

3. STABILITY

Stability is one of the most important criteria affecting the navigability of a ship.

Stability is the ability of a ship to return to its initial position after the effects of (force) moments causing this heeling terminate.

Stability depends on the shape of the hull and the position of the **ship's centre of gravity**. Thanks to this the sufficient stability can be ensured if a correct shape of the body's hull is chosen and the load is correctly distributed on the ship.

3.1. STABILITY AT SMALL ANGLES OF HEELING

We distinguish a transverse stability (at a heel) and a longitudinal stability (at an inclination, i.e. at a trim). Depending on the ship's angle of heel the transverse stability can be divided into stability at small angles of yaw (10° ÷ 15°), which is often called the initial stability, and stability at large angles of yaw.

The longitudinal stability is always considered the initial one as a result of the fact that the angles of ship's inclination are always comparatively small.

Depending on the character of effects of the heeling moment and the inclination moment the stability can be divided into a static and dynamic one. At a static heeling of the ship the external moment slowly increases from zero up to the final value, causing no angular acceleration of the ship. At a dynamic heeling external forces act on the ship with an impulse and cause its angular acceleration.

Yaws are equivolume if the volume of the underwater hull's part at a heel does not change, i.e. it remains constant. This happens for example in case of the heel and inclination caused with a wind, rather small waves, shifting the load in the ship, movement - crowd of passengers on one side, etc. The above mentioned causes are almost all causes which do commonly appear in the practice. Water lines corresponding to equivolume yawing are called equivolume water lines.

In concordance with Euler's theorem two equivolume water lines when a ship yaws at a very small angle intersect in a line passing through centres of gravity of their surface areas. This line is called the axis of ship's static yaw.

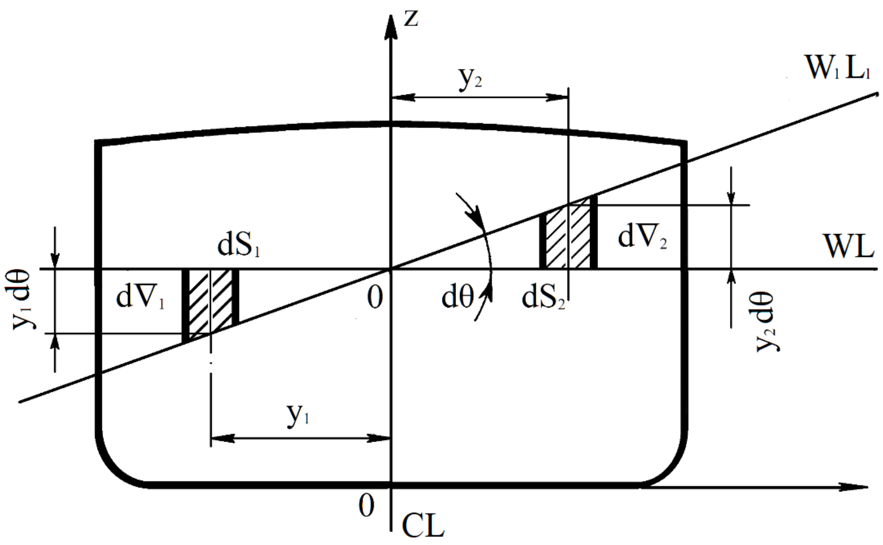

Fig. 21. Dipping at Equal Volume of Ship's Inclination at Small Angles of Heel [Authors, 1]

In compliance with Fig. 21 at the equivolume heeling of a ship at the angle $d\theta$ there on the starboard side of the ship a chock of the volume dV_2 immerses and there on the port side of the ship a chock of the volume dV_1 emerges from water, i.e.:

$$dV_1 = dV_2 \,. \tag{55}$$

If there on the emerging chock in the distance y_1 from the axis plane is defined a prism with the base surface dS_1 and the height $y_1 d\theta$, then the volume of the resulting chock will be:

$$dV_1 = \int_{S_1} y_1 . d\theta . dS_1 . \tag{56}$$

Likewise the volume of the chock immersed into water will be:

$$dV_2 = \int_{S_2} y_2 . d\theta . dS_2 . \tag{57}$$

Thus in case of the equivolume heeling the following is true:

$$\int_{S_1} y_1 . d\theta . dS_1 = \int_{S_2} y_2 . d\theta . dS_2, \tag{58}$$

where:

S_1 and S_2 are surface areas of the water line coming out and coming in water.

Both sides of this equation represent static moments of each half of the water line's (WL) surface area with respect to the line - its intersection with the water line WL_1; however, the equality of static moments of two halves of the surface area with respect to the one axis is possible if and only if this axis passes through the centre of gravity of the entire surface, which is a proof of the Euler's theorem.

In case of ships with vertical sides the Euler's theorem is valid for any final angles of heel if the water line does not come out of these vertical sides. With a certain tolerable accuracy this theorem is also applicable for small final heeling angles up to 10º - 15º of ships with skew sides which is with priority true in the theory of initial stability.

In case of the equivolume heeling at a small angle $d\theta$ the volume of the immersed part of the hull changes its shape. The centre of buoyancy as well as the centre of underwater volume is moved from the point B to the point B_1 along a certain curve which is called the trajectory of the centre of buoyancy.

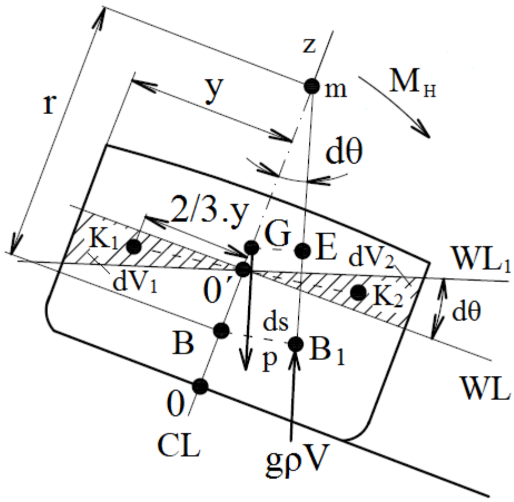

Fig. 22. Determination of a Transverse Metacentric Radius [Authors, 1]

The centre of the trajectory curvature of the centre of buoyancy (point *m*) is called a transverse metacentre. At small angles of heel the trajectory of the centre of buoyancy can be shown as part of a circle and we may assume that the transverse metacentre is located in one point m in the axis plane of the ship.

The distance - a segment from the transverse metacentre to the centre of buoyancy B: $r = m.B = m.B_1$ is called a transverse or a small metacentric radius. In accordance with the given representation it is clear that the metacentric radius is bound to the movement of the centre of buoyancy *ds* with the following dependency which is accurate for an indefinitely small angle of heel and approximate for a small angle of heel θ:

$$dS = r.d\theta \, . \tag{59}$$

From theoretical mechanics it is known that in an equilibrium system of forces the change of a resulting force moment is equal to the sum of changes of individual forces moments with respect to the one point.

In our case it means:

$$\text{g.}\rho\text{.V.dS} = \overline{K_1.K_2}.dV_1.g.\rho, \tag{60}$$

where $\overline{K_1.K_2}$ is the distance between centres of displacement chocks dV (coming into water and coming out of water).

Solving both equations we get:

$$r = \frac{K_1.K_2.dV_1}{V.d\theta}. \tag{61}$$

Fig. 22 makes clear that:

$$K_1.K_2.dV_1 = 2.K_1.0.dV_1. \tag{62}$$

For the elementary chock of the length:

$$dV_1 = \frac{1}{2}.y.y.d\theta.dx = \frac{1}{2}.y^2.d\theta.dx, \text{ i.e.} \tag{63}$$

$$K_1.O'.dV_1 = 2.\int \frac{2}{3}.y.\frac{1}{2}.y^2.d\theta.dx = \frac{1}{3}.y^3.d\theta.dx. \tag{64}$$

Integrating this expression by the entire length of the hull L and comparing this expression with the expression for I_x we will get:

$$K_1.K_2.dV_1 = 2.\int_{-L/2}^{L/2} \frac{1}{3}.y^3.d\theta.dx = d\theta.\frac{2}{3}.\int_{-L/2}^{L/2} y^3.dx = I_x.d\theta. \tag{65}$$

Comparing this expression with the expression for a small metacentric radius we will get the following formula for a transverse metacentric radius:

$$r = \frac{I_x}{V}. \tag{66}$$

The proof and the formula for r are also valid for a ship with any initial angle of heel θ_0, provided I_x means a moment of inertia of the initial water line.

If we repeated the proof for the longitudinal inclination of the ship, we could prove that the expression for a big longitudinal metacentric radius R will be:

$$R = \frac{I_f}{V}, \tag{67}$$

where:

R - distance between the longitudinal metacentre M and the centre of buoyancy B is called a longitudinal metacentric radius.

I_f - main (central) moment of inertia of the water line surface with regard to the transverse axis of the ship's inclination passing through the centre of water line; its value is determined using the Steiner's theorem as follows: $I_f = I_y - S.x_f^2$.

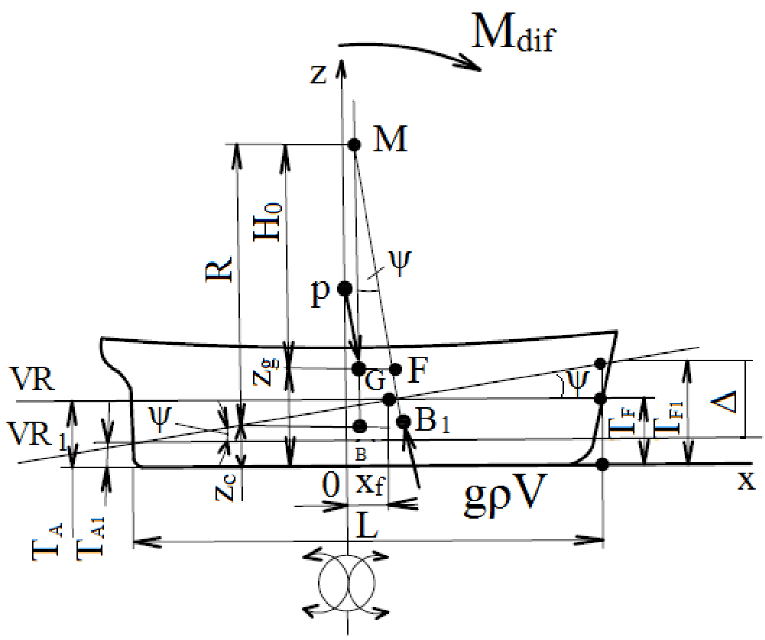

Fig. 23. Determination of a Longitudinal Metacentric Radius and Longitudinal Metacentric Height [Authors, 1]

Empirical calculations of main central moments of inertia I_x and I_f are made using the schemes introduced in the Buoyancy chapter. Moreover metacentric radiuses r and R will be determined in tables for calculating the curves of buoyancy and initial stability.

3.1.1. Metacentric Formula for Stability and Metacentric Height

Let a ship under a statically acting heeling moment M_H heel at equivolume from its straight state at a small angle θ and the position of the transverse metacentre m does not change. As a consequence of moving the centre of buoyancy the ship's weight P and the buoyancy force $g.\rho.V$ will not act on one line. These forces create a pair of forces whose moment M_R is called a **righting moment**.

If the acting of the heeling moment stops, then the righting moment returns the ship into its initial position.

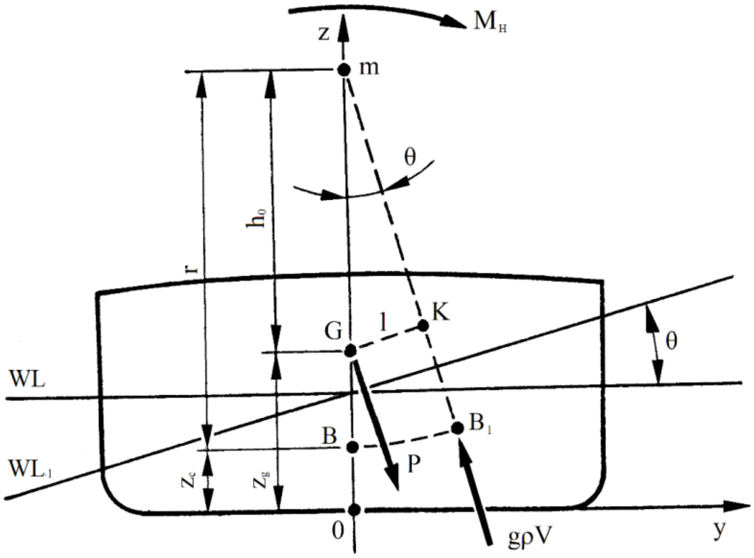

Fig. 24. Acting of a Heeling Moment on a Vessel [Authors, 1]

The figure makes it clear that the righting moment is:

$M_R = P.l.$ (68)

The value $l = /GK/$ is called a lever arm of static stability, or righting lever arm.

The value of righting moment is the measure of static stability of the ship,

$l = h_0 . \sin\theta,$ (69)

and then the righting moment is:

$M_R = P . h_0 . \sin\theta.$ (70)

The value $h_0 = \overline{m.G}$ represents the initial transverse metacentric height, i.e. the height of a transverse metacentre above the ship's centre of gravity in its straight position. The value h_0 is a weighted characteristics of the ship's initial stability.

The formula for the righting moment is often called a formula for a transverse metacentric stability. For small angles of heel it is true: $\sin\theta = \theta$ and the formula for the transverse metacentric stability can be written as follows:

$$M_R = P.h_0.\theta. \tag{71}$$

The metacentric formula for stability in case of common ships brings accurate values at infinite small angles of heel and approximate values at finite angles of heel. The application of the formula is limited for angles at which either the deck comes into water, or the ship's bilge (bottom) begins to emerge, i.e. the shape of water line, and thus the position of the metacentre and the value of the metacentric radius start to change suddenly. The calculating accuracy by the metacentric formula increases for ships with such a shape of frames which approaches a circle. The formula is frequently used for practical calculations of static stability.

Heeling moments are considered to be acting statically if their value grows from zero (from the moment the force acts on a ship) up to a computing (maximum) value slowly, at least over a time segment which exceeds the period of free rolling of the ship.

In this case moments of inertia forces may be disregarded and the condition for a static heeling is expressed with the equation of the heeling moment M_{H1} and the righting moment M_R, i.e.:

$$M_{H1}(\theta) = M_H(\theta). \tag{72}$$

The heeling moment is labelled as M_H.

If instead the righting moment M_R we substitute its value using the formula for a transverse metacentric stability, then the static heeling angle of the ship will be:

$$\theta = \frac{M_H}{P.h_0}. \tag{73}$$

Now we can formulate the required condition for a positive stability of a ship.

A ship has a positive stability when the righting moment acts in an opposite direction to the heeling moment and if the metacentre is located higher than the centre of gravity of the ship, or if the metacentric height is positive ($h_0 > 0$).

If the metacentre is located below the centre of gravity of the ship, the moment of pair of forces *P* and $\rho.g.V$ will be negative, i.e. it promotes the ship to heel further. Such a ship is not stable and in that case the metacentric height is negative ($h_0 < 0$).

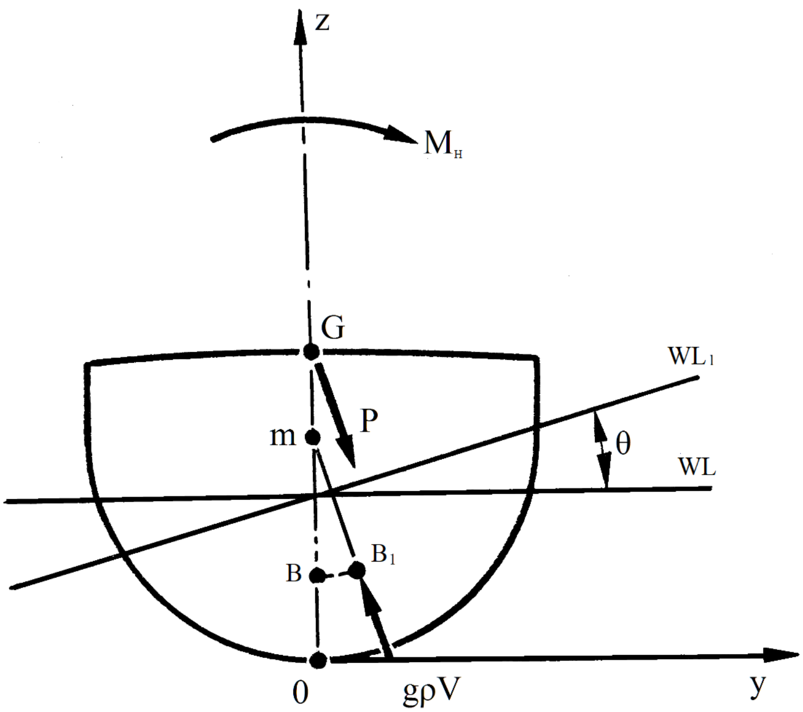

Fig. 25. Acting of a Heeling Moment on a Vessel [Authors, 1]

If the metacentre is identical with the centre of gravity, then the metacentric height and the righting moment are equal to zero. The ship is not able to return to its initial position. The ship is not in equilibrium and it is also considered unstable.

The metacentric height is a qualitative characteristics of stability. However, it is not possible to assess the degree of ship's stability according to its absolute value. A quantitative measure of stability is represented with the product $P.h_0$. This product points out a degree of a righting action of the ship against the heeling moment and it is called a **coefficient of a transverse stability**. It means, the bigger the coefficient of stability is, the more stable the ship is.

If the heeling angle is $\theta = 1°$ we will find a moment which heels the ship in 1°:

$$M_{HT} = \frac{P.h_0}{57{,}3} = 0{,}0175.P.h_0. \qquad (74)$$

The coefficient of stability $P.h_0$ and the unit heeling moment M_H are basic characteristics of the initial ship's stability with its weight *P*.

In concordance with Fig. 23 in order to determine the longitudinal metacentric radius and the longitudinal metacentric height the metacentric formula for the longitudinal stability for the inclination Ψ can analogically be written as follows:

$$M_{RS} = P.H_0.\sin\psi = P.H.\psi, \qquad (75)$$

where:

M_{RS}	- righting moment at inclination of a ship [kN.m],
H_0	- longitudinal metacentric height which equals the camber of the longitudinal metacentre above the ship's centre of gravity [m],
Ψ	- inclination angle - trim of a ship [radian].

If for the inclination (trim) of the ship it is true that $\Delta = L.tg\psi \approx L.\psi$, then the metacentric formula for the longitudinal stability may be written as follows:

$$M_{RS} = \frac{P.H_0.\Delta}{L}. \qquad (76)$$

Assuming $\Delta = 1cm = \frac{1}{100}m$ we will find the inclination moment which causes the inclination (trim) of the ship per 1 cm:

$$M_{RS} = \frac{P.H_0}{100.L}. \qquad (77)$$

Inclination in [cm] at a given inclination moment M_d will then be:

$$\Delta = \frac{M_d}{M_S} = \frac{100.M_d.L}{P.H_0} \quad . \tag{78}$$

As a result of small angles of inclination - trim the formula brings an almost accurate result.

Initial metacentric heights which are substituted into formulae for metacentric stability, as can be seen from the above mentioned schemes, are:

$$h_0 = r + z_c - z_g \, , \tag{79}$$

$$H_0 = R + z_c - z_g \, , \tag{80}$$

where:

r and R	- are metacentric radiuses for a straight position of the ship,
z_c and z_g	- are coordinates of the centre of gravity at the given displacement of the ship on an even keel.

Values of metacentric radiuses r and R and the value z_c are determined using the diagram of curves of buoyancy and initial stability. The height coordinate of the centre of gravity z_g is either determined using the book of weights or it is determined practically from the heeling test.

In initial stages of ship design and while solving a series of practical tasks curves of buoyancy and initial stability are not known and thus the given values are determined on the basis of approximate formulae.

Therefore for a ship which has the shape of a straight block (pontoon) the volume displacement, height coordinate of the buoyancy centre, and metacentric radiuses are determined using the following formulae:

$$V = L.B.T, \tag{81}$$

$$z_c = 0{,}5.T, \tag{82}$$

$$r = \frac{I_x}{V} = \frac{L.B^3}{12.L.B.T} = \frac{B^2}{12.T}, \tag{83}$$

$$R = \frac{I_f}{V} = \frac{B.L^3}{12.L.B.T} = \frac{L^2}{12.T}. \tag{84}$$

There exist some approximate formulae for calculating height coordinates of the centre of buoyancy and metacentric radiuses for ships with sharp shapes of the forebody and stern.

For example for the calculation of a height coordinate of the centre of buoyancy the formula introduced by V. L. Pozdjunin is recommended:

$$z_c = \frac{\alpha.T}{\alpha + \delta} = \frac{c_W.T}{c_W + c_B}. \tag{85}$$

Metacentric radiuses can be determined using the formulae by the professor Van der Fleet:

$$r = \frac{c_W^2}{11{,}4.c_B}.\frac{B^2}{T}, \tag{86}$$

$$R = \frac{c_W^2}{14.c_B}.\frac{L^2}{T}. \tag{87}$$

Or using the formulae by the professor V. G. Vlasov:

$$r = \frac{1}{c_B}\left(0{,}09.c_W - 0{,}02\right).\frac{B^2}{T}, \tag{88}$$

$$R = \frac{1}{c_B}\left(0{,}1070.c_W - 0{,}0378\right).\frac{L^2}{T}, \tag{89}$$

or by other authors, such as A. B. Karpov, Normand, Howgard and others.

The height coordinate of the centre z_g, when a more accurate determination is not required, will be:

$$z_g \approx k_g.D, \tag{90}$$

where: k_g - is a coefficient, value of which is determined from designs of similar ships in the already mentioned relation,

D - side depth of a ship (depth of a ship's side).

The formulae for metacentric radiuses make it clear that relations between characteristics, longitudinal and transverse stability have the following order of values:

$$\frac{P.D_0}{P.\square_0} \cong \frac{D_0}{\square_0} \cong \frac{R}{r} \cong \left(\frac{L}{B}\right)^2. \tag{91}$$

3.2. THE CHANGE OF BUOYANCY STATE AND INITIAL STABILITY OF A SHIP WHEN MOVING THE LOAD IN THE SHIP

Often it is required to determine the angle of heel and inclination as well as the change of ship's stability due to moving a load in the ship. There may also arise some requirements - in which direction and how large load it is necessary to move from one place to another in the ship so it gains the required heeling or inclination, e.g. due to removing the ship from shallow water or settling the ship on an even keel, due to emerging of screw propellers or rudders, due to emerging of a certain part of the ship's shell in order to perform an inspection, or due to repairing the ship's hull directly in water, etc.

In all reasons given above the problem in its most general form can be formulated as follows: to determine the change of a metacentric height of the angle of ship's heeling or inclination if a small load weighting $P_z = m_Z.g$ with the centre of gravity located in the point with coordinates x_0, y_0, z_0 moves to the point with coordinates x_1, y_1, z_1. The solution of such a general problem is done in three stages in which only one coordinate changes at a time.

Vertical Transfer to Δz

Let the load weighting P_z be moved from the point A_0.($x_0, y_0, z_0)$ in a parallel direction to the *z-axis* into the point A_1.($x_0, y_0, z_1)$ (Fig. 26).

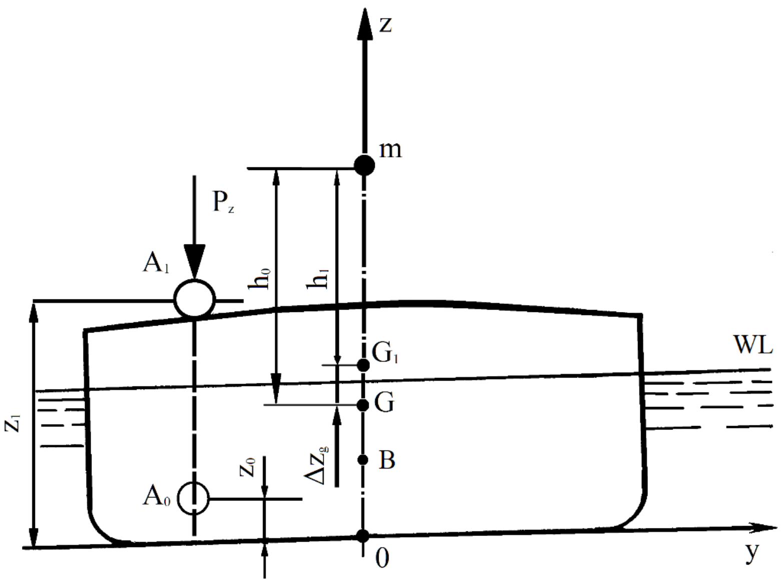

Fig. 26. Vertical Transfer of a Load [Authors, 1]

From theoretical mechanics it is known that during the movement of a partial load the resulting centre of gravity of the loads system will be shifted in that direction in which the partial load has been moved, and the value of the transferring lever arm Δz_g is directly proportional to the relation of the load weight and the entire system weight and the lever arm of the load transfer Δz_c, i.e.:

$$\Delta z_g = \overline{GG_1} = \frac{P_z}{P}(z_1 - z_0) = \frac{P_z}{P}(\pm \Delta z_z) . \quad (92)$$

Changes of metacentric heights in accordance with known relations will be equal to the sum of increments of individual components:

$$\Delta h = \Delta r + \Delta z_c - \Delta z_g , \quad (93)$$

$$\Delta H = \Delta R + \Delta z_c - \Delta z_g \quad . \tag{94}$$

Since the displacement has not changed in our case, it means that neither the values r,R,z_c have changed, and thus

$$\Delta r = \Delta R = \Delta z_c = 0 \quad . \tag{95}$$

Then the following must be true:

$$\Delta h = \Delta H = -\Delta z_g \quad . \tag{96}$$

As a result of a vertical transfer of a load the following will be true:

$$h_0' = h_0 + \Delta h = h_0 - \frac{P_z}{P}(z_1 - z_0) = h_0 - \frac{P_z}{P}(\pm \Delta z_g) \quad . \tag{97}$$

The positive value $+\Delta z_g$ means lifting the load up, and the negative value $-\Delta z_g$ means dropping it down.

Similarly:

$$H_0' = H_0 + \Delta H = H_0 - \frac{P_z}{P}(z_1 - z_0) \approx H_0 \quad . \tag{98}$$

In this case ΔH is a very small value when compared to H_0.

The equations mentioned above imply that transferring the load upwards causes the decrease of transverse stability of the ship, and vice-versa transferring the load downwards causes its increase. The longitudinal stability at a relatively small load does not practically change upon its transfer.

If the ship had an initial inclination θ_0 which was not caused with a vertical transfer of the load, then the new ship's heeling will be changed proportionally to the relation of metacentric heights. Then in accordance with the equation for a heeling and righting moment the following is true:

$$P.h_0.\theta_0 = P.h_0'.\theta_0' \quad . \tag{99}$$

Wherefrom the new initial angle of heel will be:

$$\theta_0' = \frac{\theta.h_0}{h_0'} \quad . \qquad (100)$$

Transverse Horizontal Transfer of a Load to Δy

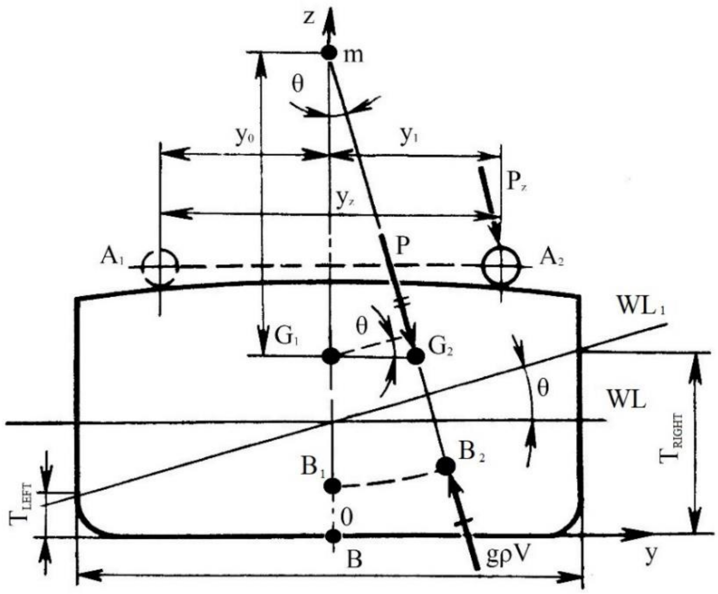

Fig. 27. Transverse Transfer of a Load [Authors, 1]

A load P_z is moved from the point A₁.(x_0, y_0, z_1) in a parallel direction to the *y*-axis to the point A₂.(x_0, y_1, z_1) for a distance $\Delta y_2 = y_0 - y_1$, which results in moving the centre of gravity of the ship and developing the heeling moment

$$M_H = P_z.\Delta y_z.\cos\theta. \qquad (101)$$

In a new position of equilibrium the heeling moment is equal to the righting moment which is determined using the metacentric formula for stability:

$$P_z.\Delta y_z.\cos\theta = P.h_0'.\sin\theta \quad . \qquad (102)$$

Wherefrom:

$$tg\theta = \frac{\sin\theta}{\cos\theta} = \frac{P_Z.\Delta y_Z}{P.h_0'} \cong 0. \qquad (103)$$

If the ship has already had an initial angle of heel θ_0', then the angle of heel will be as follows after the vertical and horizontal transfer of the load:

$$\theta_1 = \theta_0' + \frac{P_Z}{P.h_0'} \Delta y_z , \quad (104)$$

where: h_0' - is a metacentric height after the vertical transfer of the load.

The transfer on the starboard (right) and port (left) side of the ship in accordance with Fig. 27 will be:

$$T_{RIGHT} = T + 0.5.B.\tan\theta \quad (105)$$

$$T_{LEFT} = T - 0.5.B.\tan\theta. \quad (106)$$

Longitudinal Transfer of a Load

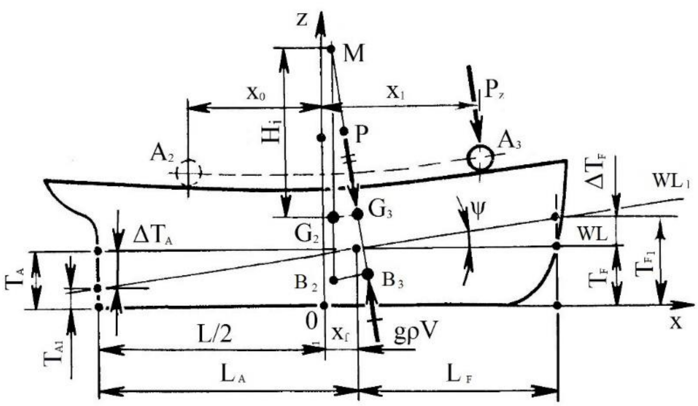

Fig. 28. Longitudinal Transfer of a Load [Authors, 1]

During the transfer of a load P_z from the point $A_2.(x_0, y_1, z_1)$ in a parallel direction to the x-axis to the end point $A_3.(x_1, y_1, z_1)$ the ship gains the inclination Ψ. If we again apply the considerations which were applied in case of the transverse transfer of a load, the expression for the angle of inclination while tolerating small angles($tg\psi \cong \psi$) will be as follows:

$$\psi = \frac{P_Z\left(x_1 - x_0\right)}{P.H_1} \quad . \tag{107}$$

Disregarding the change of the longitudinal metacentric height the expression for the ship's inclination will be as follows:

$$\Delta = \frac{P_Z\left(x_1 - x_0\right)}{P.H_0}.L \quad . \tag{108}$$

Then new forward draught and aft draught after the vertical and longitudinal movement of the load P_z will be:

$$T_{F1} = T_F + \left(\frac{L}{2} - x_f\right)\frac{P_Z.(x_1 - x_0)}{P.H_0}, \tag{109}$$

$$T_{A1} = T_A - \left(\frac{L}{2} - x_f\right)\frac{P_Z.(x_1 - x_0)}{P.H_0}. \tag{110}$$

3.3. DETERMINATION OF STABILITY AND CHANGE OF BUOYANCY STATE OF A SHIP AFTER THE RECEPTION OR REMOVAL OF A SMALL LOAD

The reception or removal of a small load weighting P_z in the point with coordinates (x_z, y_z, z_z) affects the change of buoyancy state and ship's stability. (The removal of a load will be marked with a minus sign.)

The process of receiving (removing) a small load comprises three stages.

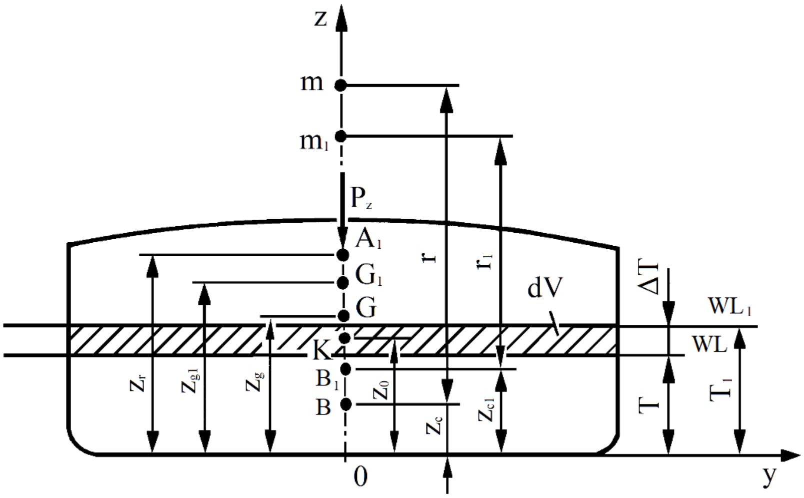

Fig. 29. Change of Elements of Buoyancy and Stability after the Reception of a Small Load [Authors, 1]

In the first stage the reception of a load is understood not to change the initial stability. It is clear that such a condition can only be met if the load is received into the centre of supplemental volume displacement ΔV (point K with coordinates (x_f, 0, z_0)); however, since the weight of the received load and supplemental buoyancy force are as if received in one point, the following relations must be valid:

$$P_1 = P \pm P_Z \, , \tag{111}$$

$$V_1 = V \pm \Delta V = \frac{P \pm P_Z}{\rho . g} \, . \tag{112}$$

Values of coefficients of stability remain unchanged, and thus the following is true:

$$P.h_0 = (P \pm P_Z).h_0' \, , \tag{113}$$

$$P.h_0 = (P \pm P_Z).H_0' \, , \tag{114}$$

and for this reason values of metacentric heights after receiving the load into the point K (x_f, 0, z_0) will be:

$$h_0' = \frac{P.h_0}{P \pm P_Z}, \tag{115}$$

$$H_0' = \frac{P.H_0}{P \pm P_Z}. \tag{116}$$

The change of a draught after receiving or removing the load will then be as follows:

$$\Delta T = \pm \frac{m_Z}{\rho.S} = \frac{P_Z}{g.\rho.S}. \tag{117}$$

In the second stage there is determined the effect of a vertical shift of the load P_z from the point K into the point A_1 (x_f, 0, z_z) on stability. Thus the segment of trajectory of the load transfer $\overline{A_1K}$ will be of the following length:

$$A_1K = z_Z - z_0 = z_Z - \left(T + \frac{\Delta T}{2} \right). \tag{118}$$

From the point of view of keeping the meaning (sign) during the load transfer its negative value is applied, and thus:

$$A_1K = T + \frac{\Delta T}{2} - z_Z. \tag{119}$$

Taking into account values for metacentric height determination after receiving the load into to the point K and after its transfer into the point A_1 we will get the value of metacentric height including receiving or removing the load as follows:

$$h_1 = h_0' + \Delta h = \frac{P.h_0}{P \pm P_Z} + \frac{\pm P_Z}{P \pm P_Z} \left(T \pm \frac{\Delta T}{2} - z_Z \right). \tag{120}$$

Or after a modification the expression for a metacentric height will be:

$$h_1 = h_0 + \frac{\pm P_Z}{P \pm P_Z}\left(T \pm \frac{\Delta T}{2} - h_0 - z_Z\right). \tag{121}$$

The plus sign applies to the case of receiving the load, and the minus sign applies to the case of its removing.

If the value h_0 is unknown, then h_1 is either taken directly from curves of buoyancy, or it is determined using the following formula $h_0 = r + z_c + z_g$ in a functional relation $h_1 = f(T_1, z_{g1})$.

The increment of a coefficient of stability after receiving or removing the load will then be as follows:

$$\Delta(P.h) = (P + P_Z)h_1 - P.h_0 = (P + P_Z)\left[P.\frac{h_0}{P + P_Z} + \frac{P_Z}{P + P_Z}(T \pm \Delta T - z_Z)\right] - P.h_0 =$$
$$= \pm P_Z.\left(T \pm \frac{\Delta T}{2} - h_0 - z_Z\right) \tag{122}$$

The third stage represents the transfer of the load from the point A_1 into the point $A_2.(x_z, y_z, z_z)$ according to an analogical process of a longitudinal load transfer.

The above mentioned formula for a coefficient of stability allows for stating the following conclusions:

1. if $|z_Z| \leq \left(T + \frac{\Delta T}{2}\right)$, then $\Delta(P.h) > 0$ after receiving the load, and $\Delta(P.h) < 0$after removing the load. In other words if the centre of the received load P_z is lower than the "acting" water line, then the static stability of the ship will increase after receiving the load and decrease after removing it;
2. if the centre of the received load P_z is higher than the "acting" water line, then the stability of the ship will decrease after receiving the load and increase after removing it.

The equation for a longitudinal metacentric height is practically analogical to the equation for the transverse one, i.e.:

$$H_1 = H_0 + \frac{\pm P_Z}{P \pm P_Z}\left(T \pm \frac{\Delta T}{2} - H_0 - z_Z\right). \tag{123}$$

In case of majority of ships the value of the expression $\left(T \pm \frac{\Delta T}{2} - z_Z\right)$ when compared to H_0 can be disregarded, so then:

$$H_1 = H_0 \pm \frac{\pm P_Z}{P \pm P_Z}\left(-H_0\right) = H_0\left(1 - \frac{\pm P_Z}{P \pm P_Z}\right), \tag{124}$$

or the following relation is true:

$$\left(P \pm P_Z\right)H_1 = P.H_0, \tag{125}$$

i.e. the coefficient of a longitudinal stability after receiving or removing a small load can be considered fixed.

The heeling angle by the received load is in accordance with the formula for the load transfer (there is the equality of the heeling and righting moment)

$$\theta = \frac{\pm P_Z . y_Z}{\left(P \pm P_Z\right).h_0'}. \tag{126}$$

If the ship had an initial heeling θ_0 up to the moment of receiving the load, then the heeling angle after receiving the load will be as follows:

$$\theta_1 = \theta + \theta_0 \frac{h_0}{h_0'}. \tag{127}$$

The inclination moment, total inclination (trim) and the ratio of the forebody to the stern since the load reception will be:

$$M_S = \pm P_Z\left(-x_Z - x_f\right), \tag{128}$$

$$\Delta = \pm \frac{\pm P_Z (x_Z - x_f)}{(P + P_Z).H_1}.L = \frac{\pm P_Z (x_Z - x_f).L}{P.H_0}, \tag{129}$$

$$T_{F1} = T_F \pm \Delta T + \frac{\pm P_Z.(x_Z - x_f)}{P.H_0}.\left(\frac{L}{2} - x_f\right), \tag{130}$$

$$T_{A1} = T_A \pm \Delta T + \frac{\pm P_Z.(x_Z - x_f)}{P.H_0}.\left(\frac{L}{2} + x_f\right). \tag{131}$$

3.4. STABILITY OF A SHIP WHILE MOVING A SUSPENDED MASS OR LIQUID CARGO

3.4.1. Static Heel of a Floating Crane

A burden weighting P_z, with its centre of gravity in a straight position located in the point *B*, is suspended in the point *A*.

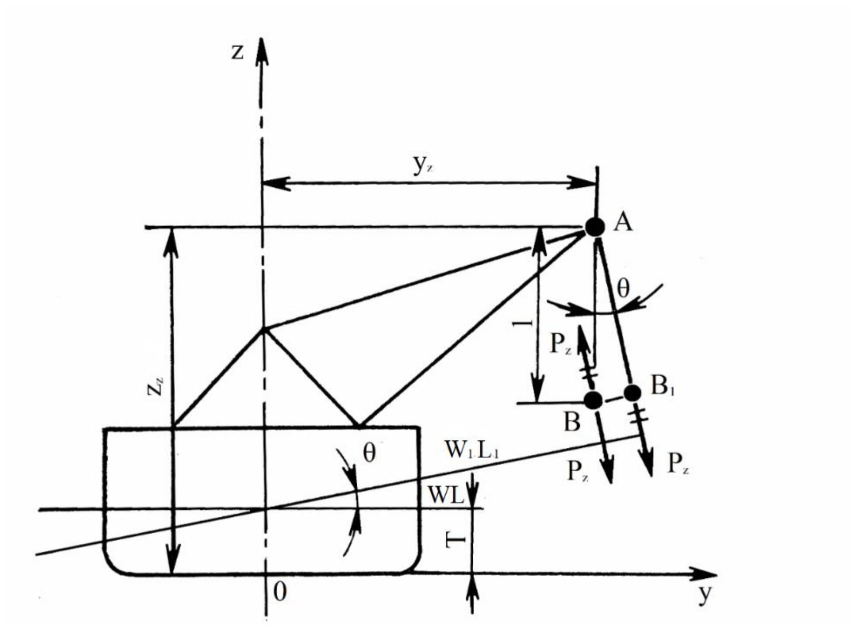

Fig. 30. Stability of a Ship with a Suspended Not Fixed Burden (Load) [Authors, 1]

In case of a heeling caused with the heeling static moment M_H at the angle θ this burden (if not fixed) is moved from the point B to the point B_1. If in the point B there are two forces acting against each other P_z parallel to the direction $\overline{B_1A}$, then we will reduce the force action from the burden P_z in the point B towards the supplemental heeling moment.

$$\Delta M_H = P_Z . l . \sin\theta, \tag{132}$$

where: l - is the distance between the centre of burden suspension A above the actual position of the centre B.

The equation of the equilibrium while heeling and righting moments are acting, will then be:

$$M_H + P_Z . l . \sin\theta = P_Z . h . \sin\theta, \tag{133}$$

or

$$M_H = \left(h_0 - \frac{P_Z}{P} . l_1\right) . \sin\theta, \tag{134}$$

where: h_0 - metacentric height provided that the centre of load P_z is located in the point *B*.

If we mark:

$$h_0' = h_0 - \frac{P_Z}{p}.l \quad , \qquad (135)$$

we will get the following:

$$M_H = M_R = P.h_0'.\sin\theta \approx P.h_0'.\theta. \qquad (136)$$

If we compare the gained formula with the formula for a vertical transfer, we may say that the effect of a suspended not fixed burden on the stability is the same as stable loads (of a small weight) P_z, centre of which is as if it has been transferred from the point *B* into the point of suspension *A*, since the change will only influence the metacentric height in terms of its decrease down to the value $\Delta h = \frac{P_Z.l}{P}$.

Thus after receiving or removing a not fixed suspended load (e.g. in case of floating cranes) the metacentric height can be calculated directly using the formula for the reception or removal of a small load:

$$h_1 = .\frac{P.h_0}{P \pm P_Z} + \frac{\pm P_Z}{P \pm P_Z}\left(T \pm \frac{\Delta T}{2} - z_Z\right) , \qquad (137)$$

and the heeling angle:

$$\theta_1 = \frac{\theta_0.h_0}{h_0'} + \theta, \qquad (138)$$

when coordinates of the point of reception and removal of the load y_z, z_z are equal to coordinates of the top point of the boom, i.e. they match the point of suspension or the point *A*.

The change of the distance *l* between the point of suspension *A* and the centre of burden after the load (burden) is received will not affect the value of the metacentric height h_1 if the point *A* does not change its position.

3.4.2. Free Water Surfaces

If there is a space in a ship where a liquid cargo weighting P_z and having a free water surface is poured, then after the ship's heeling the centre of liquid cargo is shifted from the point K to the point K_1 in the distance of Δl_2.

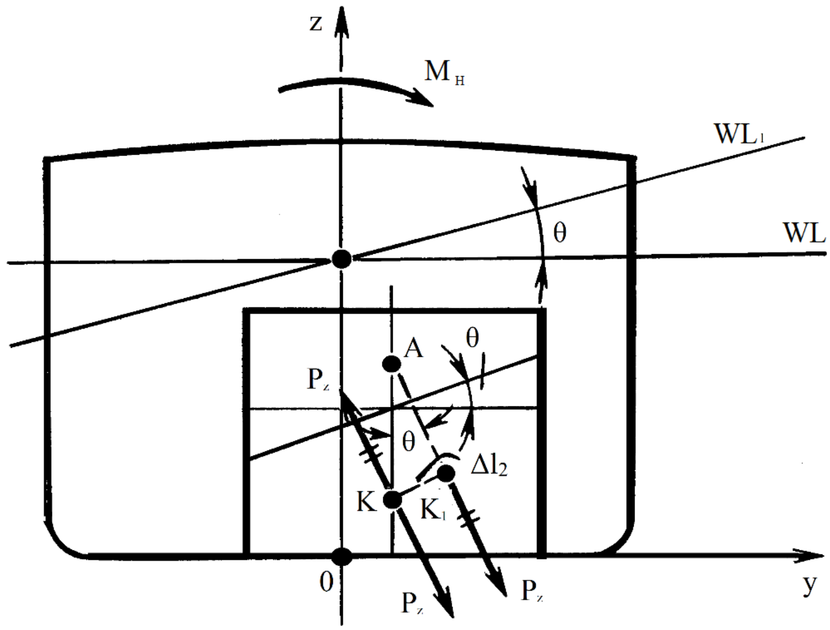

Fig. 31. Stability of a Ship with a Free Water Surface of Liquid Cargo [Authors, 1]

If we use the same method as in case of a suspended load, we can substitute the effects of the liquid movement with a supplemental heeling moment

$$\Delta M_H = P_Z . \Delta l_2 \tag{139}$$

The point K can be considered a centre of buoyancy of the liquid volume V_t. Then the point A can be considered or likened to the metacentre V_t and the distance $\overline{AK}$ can be considered or likened to the metacentric radius r_t. Based on the already known relations the following may be written:

$$AK = r_t = \frac{i_x}{V_t}\,, \tag{140}$$

where: i_x - moment of inertia of a horizontal position of liquid cargo's free water surface in a given ship's room with regard to the central axis of the heeling of this surface at the ship's heeling [m^4].

Fig. 31 makes clear the following:

$$\Delta l_2 = AK.\sin\theta = \frac{i_x}{V_t}\sin\theta\,, \tag{141}$$

$$M_H = P_Z.\frac{i_x}{V_t}.\sin\theta = \rho_t.g.V_t.\frac{i_x}{V_t}.\sin\theta = \rho_t.g.i_x.\sin\theta\,. \tag{142}$$

The equation of the equilibrium of a ship heeled at a small angle will be as follows:

$$M_H + \Delta M_H = M_V = P.h_0.\sin\theta \tag{143}$$

or

$$M_H = M_v' = P.\left(h_0 - \rho_t.g.\frac{i_x}{P}\right).\sin\theta. \tag{144}$$

The value given in parentheses can be considered a new metacentric height. Then it is true:

$$h_0' = h_0 - \rho_t.g.\frac{i_x}{P} = g.\rho.\frac{I_x}{P} + z_c - z_g - g.\rho_t.\frac{i_x}{P} = \frac{g.\rho.I_x - g.\rho_t.i_x}{P} + z_c - z_g\,, \tag{145}$$

where:

h_0 - the initial metacentric height calculated on condition that the centre of liquid P_t is fixed and is located in the point K (i.e. as if the liquid did not have a free water surface) [m],

ρ_t - the liquid density [kg.m^{-3}],

ρ - the water density [kg.m^{-3}].

Thus a free water surface of liquid cargo, which is able to overflow at the ship's heeling, decreases the metacentric radius and the metacentric height of the ship:

$$\Delta h_t = -\frac{g.\rho_t.i_t}{P} \quad . \tag{146}$$

If several compartments (tanks) are filled with liquid cargo, then it is necessary to calculate corrections of metacentric heights for each compartment and subsequently to sum them:

$$\Delta h_t = -\frac{\sum_{j=1}^{n} g.\rho_{t_j}.i_{x_j}}{P} \quad . \tag{147}$$

Then

$$h_0' = h_0 - \frac{\sum_{j=1}^{n} g.\rho_{t_j}.i_{x_j}}{P} = \frac{g.\rho.I_x - \sum_{j=1}^{n} g.\rho_{t_j}.i_{x_j}}{P} + z_c - z_g \quad . \tag{148}$$

To factor the effects of a free water surface of liquid cargo on the longitudinal stability some analogical expressions may be written:

$$\Delta H_t = -\frac{\sum_{j=1}^{n} g.\rho_{t_j}.i_{y_j}}{P} \quad , \tag{149}$$

$$H_0' = H_0 - \frac{\sum_{j=1}^{n} g.\rho_{t_j}.i_{y_j}}{P} = \frac{g.\rho.I_f - \sum_{j=1}^{n} g.\rho_{t_j}.i_{y_j}}{P} + z_c - z_g \quad , \tag{150}$$

where:

i_{yi} - the moment of inertia of a free water surface of liquid cargo in the yth compartment with regard to the inherent transverse axis of the heeling [m^4],

I_f - the moment of inertia of the acting water line surface with regard to the transverse central axis [m^4].

In order to allow for eliminating the effect of liquid cargo on the transverse stability some longitudinal watertight bulkheads are placed there in given sections (tanks). The supplementation of transverse bulkheads will lead to the elimination of liquid cargo effect on the longitudinal stability.

E.g. in case of liquid cargo when the length of the compartment (tank) is l_0 and its breadth is B, the reduction of a transverse metacentric height due to a free water surface effect will be:

$$\Delta h_t' = -\frac{g.\rho_t.i_x}{P} = -g.\rho_t \frac{l_0.B^3}{12.P} \quad (151)$$

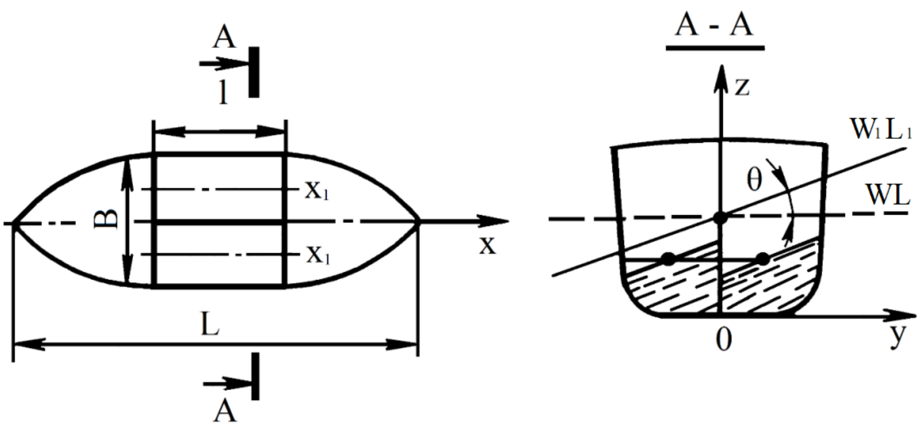

Fig. 32. Determination of an Effect of a Free Water Surface of Liquid Cargo on a Vessel Stability [Authors, 1]

After placing a longitudinal bulkhead the relevant section (tank) is divided into two parts. Then the metacentric height is amended as follows:

$$\Delta h_t'' = -\frac{\sum_{1}^{2} g.\rho_t.i_{x_j}}{P} = \frac{2.g.\rho_t.l_0.\left(\frac{B}{2}\right)^2}{12.P} = -\frac{g.\rho_t.l_0.B^3}{48.P} . \qquad (152)$$

After such a modification the metacentric height is four times decreased.

It may be stated that the placement of watertight bulkheads eliminates the negative effect of a free water surface of liquid cargo on the stability proportionally to the quadrat of number of newly created sections within the given tank.

All considerations above are true if the free water surface of liquid at heeling does not flow over the placed straight bulkheads within the tank or outside the tank.

If a small amount of liquid cargoes with the total weight P_t is placed in the ship, the metacentric height must be calculated using the formulae for a load reception with regard to the effect of a free water surface of individual liquid cargoes:

$$h_1' = h_0 + \Delta h + \Delta h_t = h_0 + \frac{P_t}{P+P_t}\left(T + \frac{\Delta T}{2} - h_0 - z_t - \frac{\sum_{j=1}^{n} g.\rho_{t_j}.i_{x_j}}{P_t} \right), \qquad (153)$$

where:

z_t - is the height coordinate of the centre of received liquid cargo.

The longitudinal metacentric height after receiving liquid cargo with a free water surface will then be:

$$H_1' = \frac{P.H_0 - \sum_{j=1}^{n} g.\rho_{t_j}.i_{y_j}}{P+P_t} = H_0 - \frac{P_t}{P+P_t}\left(H_0 + \frac{\sum_{j=1}^{n} g.\rho_{t_j}.i_{y_j}}{P_t} \right). \qquad (154)$$

Similarly, like liquid cargoes affect the stability, so do grain cargoes. Their movability (angle = natural angle of repose) is usually taken into account in calculations, e.g. in case of transport of fresh fish put into ship's rooms with a free surface and without any barriers against the side shift such a cargo is considered as if liquid.

Typical examples of the application of the above mentioned formula:

- the heeling test,
- the transit of a ship from a river to the sea,
- lifting up the ship's forebody (at repairs of a propulsion-rudder complex directly in water),
- ship striking a shoal,
- icebreaker striking ice,
- lifting up the ship in a dock, etc.

3.5. STABILITY AT LARGE ANGLES OF SHIP'S HEEL

3.5.1. Lever Arms of Static Stability

At large angles of heel the symmetry of displacement chock coming into water and coming out of water with regard to the axis plane and heeled water line changes significantly. This change is particularly evident on the forebody and stern of the ship.

For these reasons the line of intersection of two equal volumes of displacement of the water line is moved from the point $0^{'}$ to the point 0_{θ} and the trajectory of the centre of buoyancy $\overline{BB_{\theta}}$ in the plane of the main frame cannot be considered part of a circle.

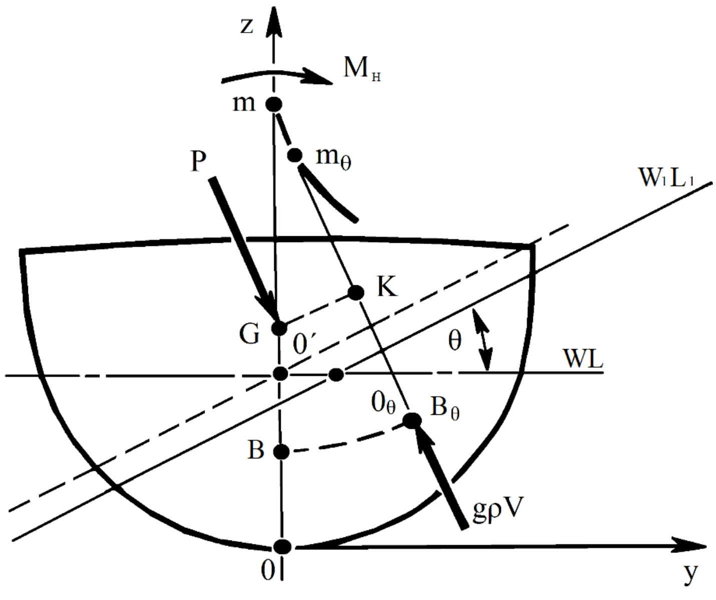

Fig. 33. Heeling of a Ship at Large Angles of Heel [Authors, 1]

These considerations are identical with a statement that the transverse metacentre at a large angle of heel does not remain in the same point in the axis plane, as it is in case of small angles of heel, but it is moved into the point m_θ along a curve called a curve of metacentres (metacentric evolute).

The curve of metacentres is a geometrical place of centres of the trajectory curvature of the centre of buoyancy which may be marked as a curve of centres of buoyancy (or a displacement evolute).

Apparently the righting lever arm $\overline{BK}$ will not be determined using the metacentric formula for stability (lever arm $l \neq h_0 . \sin\theta$) which is applicable in case of a circular trajectory of displacement at small angles of heel. Therefore it is required to find a more accurate expression for *l*.

3.6. DIAGRAM OF STATIC STABILITY

The diagram of static stability represents the dependency of the righting lever arm *l* or the moment M_v itself depending on the angle of heel θ for a ship equalising on an even keel at a certain draught T and certain height of the centre z_g over the basic plane. An analytical expression of this dependency is given with the formula $l = y_{b,90}.\cos\theta + \left(z_{b_\theta} - z_b\right).\sin\theta - \alpha.\sin\theta$.

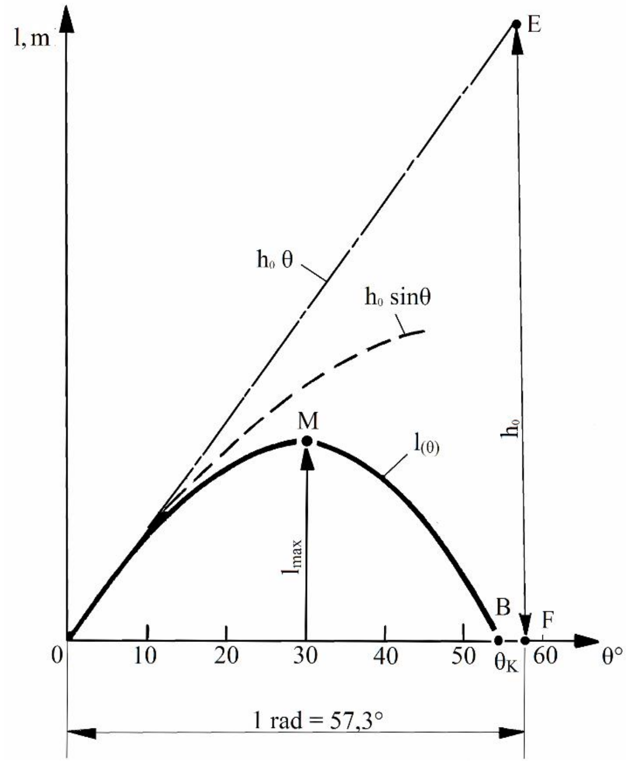

Fig. 34. Diagram of Static Stability [Authors, 1]

The diagram of static stability is a curve with a graphical expression of the maximum. Three characteristic points can be highlighted on it:

- point 0 - the origin of the coordinate system, where $l = M_v = 0$,
- point M - where the lever arm or the moment achieves its maximum value,

- the point of curve K zenith (in which $l = M_v = 0$) which determines the theoretical limit heeling angle θ_K.

When the ship heels at angles $\theta \geq \theta_K$, righting moments are negative which results in turning the ship bottom up.

In Fig. 34 there are (besides the curve of static stability) also shown curves of righting lever arms calculated for the same ship using formulae for small angles of heel; in mutual comparison we can see that in the range of angles of heel from $\theta = 0°$ to $\theta \leq (10° \div 15°)$ the dependency of the lever arm *l* on the angle of heel has a linear character, and they are thus determined using the metacentric formulae for transverse stability with a sufficient accuracy. At large angles of heel this dependency has a nonlinear character and can be determined only using the diagram of static stability. That is why diagrams of static stability represent very important ship's documents which are needed for the ship's stability determination.

There exists a rule how to check the correctness of a static stability diagram design; the differential (gradient) of the righting moment at the angle of heel $\theta = 0°$ is equal to the metacentric height h_0:

$$\left| {dl}/{d\theta} \right|_{\theta=0} = tg\alpha_0 = \frac{EF}{OF} = \frac{h_0}{1rad} = h_0 . \tag{155}$$

The correctness of this statement can easily be checked through differentiating the formula for *l*.

$$l = y_{b_\theta}.\cos\theta + \left(z_{b_\theta} - z_b\right).\sin\theta - \alpha.\sin\theta, \tag{156}$$

$$\frac{dl}{d\theta} = \frac{dy_{b_\theta}}{d\theta}\cos\theta - y_{b_\theta}.\sin\theta + \frac{d.\left(z_{b_\theta} - z_b\right)}{d\theta}.\sin\theta + \left(z_{b_\theta} - z_b\right).\cos\theta - \alpha.\cos\theta. \tag{157}$$

As a result of the fact that $\frac{dy_{b_\theta}}{d\theta} = r_0.\cos\theta$; $\frac{d.\left(z_{b_\theta} - z_b\right)}{d\theta} = r_0.\sin\theta$ when $\theta = 0$, $z_{b_\theta} - z_b$; $y_{b_\theta} = 0$ we will get:

$$\left(\frac{dl}{d\theta}\right)_{\theta=0} = r_{0.}\cos^2\theta + r_0.\sin^2\theta - \alpha = r_0 + z_b + z_g = h_0. \quad (158)$$

This relation is a proof that if a segment $OF = 1rad = 57{,}3°$ is put there on the horizontal axis of the static stability diagram and from the point F a segment $EF = h_0$ is put there on the vertical axis, then the line OE must be a tangent to the diagram in the origin of the coordinate system.

If there in a static stability diagram in vertical direction not the lever arm, but the righting moment $M_V = P.l$ is put, which differs from the lever arm only in the measure, then the differential by the righting moment:

$$\left(\frac{d.M_v}{d\theta}\right)_{\theta=0} = P.h_0 \text{ is a coefficient of stability.} \quad (159)$$

Diagrams of static stability are needed for the determination of the angle of a static heeling for the given heeling moment, or vice-versa, for finding the heeling moment for the given allowable angle of heel θ_{All}. In other words, to get a graphical solution the diagrams of static stability use equations for a static heeling angle $M_H(\theta) = M_R(\theta)$, which are frequently used in the form of equity of the heeling lever arm l_H and the righting lever arm l_R corresponding to a unit of ship's weight.

$$\frac{M_H(\theta)}{P} = l_H(\theta) = l_R(\theta) = \frac{M_R(\theta)}{P} \quad (160)$$

The following lines will introduce some practical examples in more detail:

1. The heeling moment M_H or its lever arm l_H increases progressively until it achieves the final value in the point 1: $l_H \le l_{Hmax}$

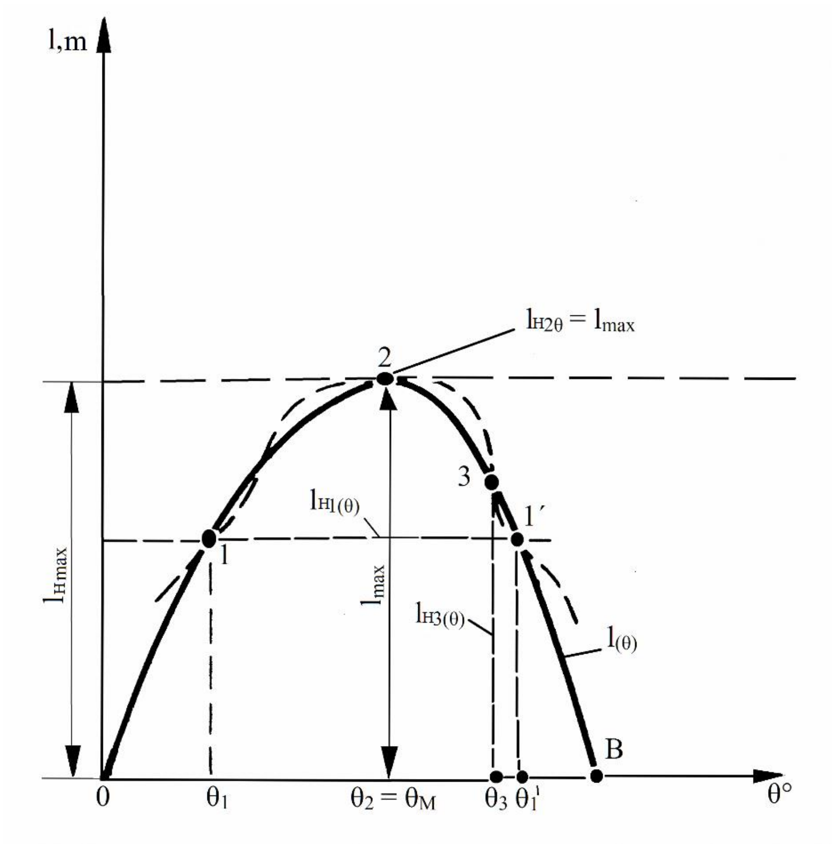

Fig. 35. Diagram of Static Stability (while solving problems when the initial angle of heel is not known for the ship) [Authors, 1]

and it remains constant at this value. The ship heels at the angle θ_1 and will be in a stable state. The ship cannot heel at the angle $\theta_1^{'}$, where it is also valid that $l_{H1} = l$, because if $\theta_1^{'} \geq \theta_1$, then $l_{H1} \leq l$. This example of a static heel is frequently repeated in practice: heeling by load transfer or by movement of passengers onto another side of the ship, by wind flowing with a constant speed, etc.

2. The heeling moment or its lever arm $l_{H_2(\theta)} = \frac{M_H}{P}$, which increases progressively, will achieve its maximum value l_{Hmax} (point 2) and remain the same. At this time the ship heels at the angle $\theta_2^{'} \geq \theta_{max}$, which leads to an unstable position because any heeling (to the right) will evoke the ship's overturning.

3. The moment or its lever arm $l_{H_3(\theta)}$ increases progressively, passing through the maximum point, and then it increases to the point 3 which corresponds to the theorem $l(\theta)$, and from the point 3 it decreases to zero. In this case the ship heels at the angle $\theta_3 \geq \theta_{max}$ and disregarding the transition beyond the maximum point it will occur in a stable position (if, of course, the angle θ_3 is less than the angle of flooding at which the hull starts to be flooded through not closed openings).

Such a case of heeling moments acting may happen in a ship taken down the stream under the bridge when its hinge is higher than the lower fixed point of the bridge (Fig. 36).

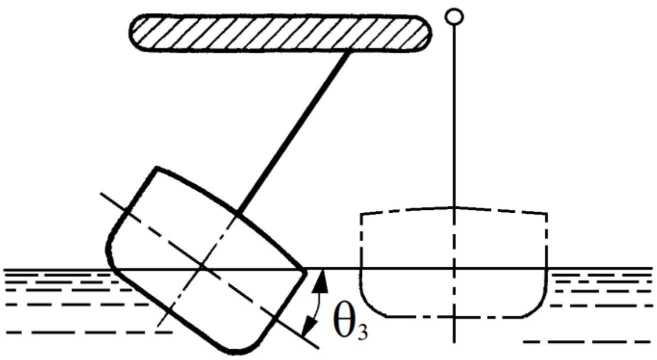

Fig. 36. Heeling of a Ship at Large Angles of Heel (static heeling angle of a ship taken down the stream under the bridge) [Authors, 1]

A similar case is that one of heeling of small ships, such as yachts, at such large angles so it is possible to perform coating, e.g. of the bilge part of the hull. For this case a burden with a precisely calculated length of burden underslung P_g is suspended on a swinging derrick so the burden can create a heeling moment a bit larger than the maximum righting moment ($l_{H1} \geq l$). As soon as the burden touches the shore $M_H = P_g.b = 0$, any further heel is stopped and the ship remains in a heeled state.

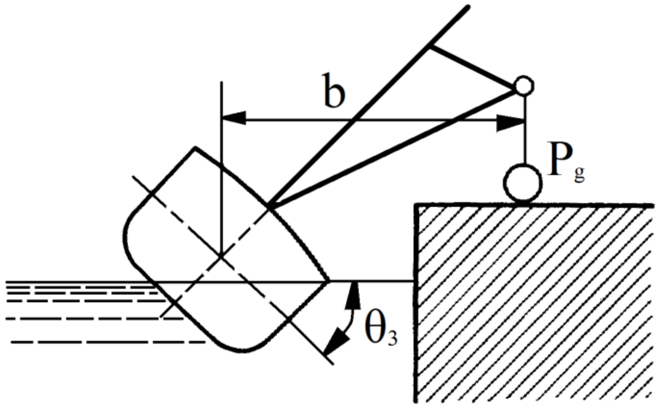

Fig. 37. Heeling of a Ship at Large Angles of Heel (heeling of a ship for the purpose of the hull side treatment [Authors, 1]

Based on the analysed examples the following rule may be formulated: If the curve of a heeling moment or its lever arm l_H crosses the diagram (curve $l(\theta)$) from the outer to the inner part (point 1 or 3, Fig. 35), then the heeled ship will be in a stable position; and if the crossing starts from the inner to the outer part (point 1', Fig. 35) or $l_{H(\theta)}$ only touches the curve $l(\theta)$ (point 2, Fig. 35), then the ship will be in an unstable position.

4. If the statically applied heeling moment M_H or its lever arm l_H acts on a ship which has the initial angle of heel $\pm\theta_0$, then the construction design of the static stability diagram in order to find out the final angle θ_1 is made analogically to the process when coming from zero; however, zero is the value of this angle $\pm\theta_0$ and the angle θ_1 is subtracted from the origin 0.

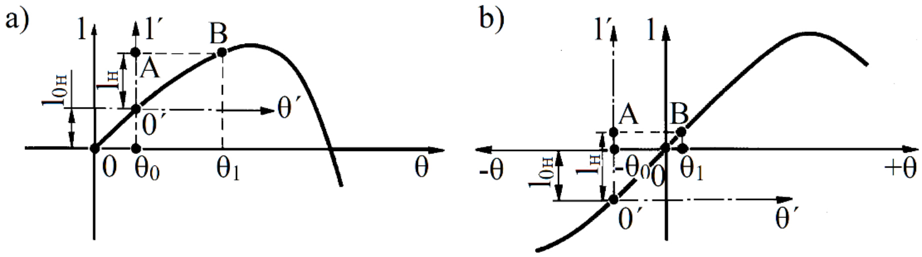

Fig. 38. Diagram of Static Stability for a Ship with the Initial Angle of Heel θ a) to the starboard side (+θ0), b) to the port side (-θ0) [Authors, 1]

3.6.1. Impact of Movement, Load Reception and Free Liquid Surface on the Diagram of Static Stability

Previous considerations imply that the diagram of static stability is calculated and constructed for a ship with a certain volume displacement V and certain position of the centre of weight (z_g; $y_g = 0$). If a certain load is shifted in the ship or if a supplemental load is received (removed), the diagram of static stability must be reconstructed.

Change of the Diagram of Static Stability by Transferring a Load

Let the load weighting P_z be shifted across the ship in the distance $\Delta y_z = y_1 - y_0$ and in the vertical direction in the distance $\Delta z_z = z_1 - z_0$. The load transfer along the ship in any distance will only cause a change of the inclination (trim) and will not impact the transverse stability. The ship's centre of gravity will be moved from the point G to the point G_1 and its coordinates will gain some increments:

$$\Delta y_g = \frac{P_z . \Delta y_z}{P}; \Delta z_g = \frac{P_z . \Delta z_z}{P}. \quad (175)$$

A new lever arm of static stability l_1 after the load transfer will be as follows:

$$l_1 = l - \Delta y_g . \cos\theta \pm \Delta z_g . \sin\theta = l - \frac{P_z . (\Delta y_z . \cos\theta \pm \Delta z_g . \sin\theta)}{P}. \quad (176)$$

The + sign for Δz_z is applied when lifting the load up, and the - sign is applied when dropping the load down.

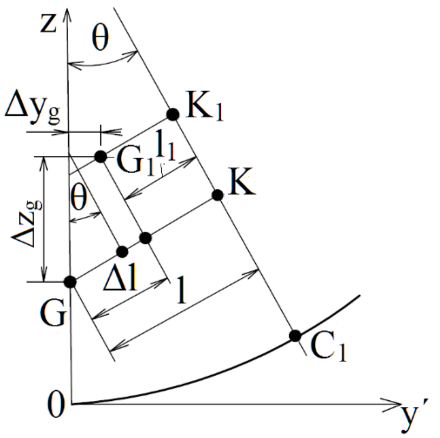

Fig. 39. Change of the Diagram of Static Stability when Transferring the Heeling in a Ship (Determination of Δl, Δl_1) [Authors, 1]

The change of the diagram of static stability is shown below:

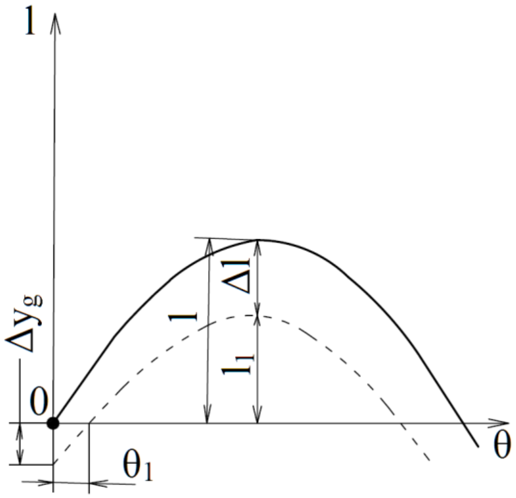

Fig. 40. Change of the Diagram of Static Stability [Authors, 1]

As can be seen in the figure, the ship gains the angle of heel θ_1 at the load transfer in the transverse direction.

Change of the Diagram of Static Stability after Receiving or Removing a Small Load

If a small load weighting P_z is received or removed from the point with coordinates y_z, z_z, it is necessary to determine a new value of the lever arm of static stability l_1; this can be done through calculating changes of lever arms of form Δl_f and weight l_g and the load transfer $\Delta l'$.

Due to the change of coordinates of the centre of buoyancy after receiving the load there happens the modification of expressions:

$$y_{b_\theta} = \int_0^\theta r_\theta . \cos\theta \, . d\theta, \tag{177}$$

$$z_{b_\theta} - z_b = \int_0^\theta r_\theta . \sin\theta . d\theta \, . \tag{178}$$

At the same time the change of the hull shape, i.e. of the acting water line, is disregarded, i.e. $I_{x1} = I_x$. The change of the metacentric radius can then be determined from familiar expressions:

$$\Delta r_0 = r_1 - r_0 = \frac{I_{x_1}}{V + \Delta V} = \frac{I_x}{V} = -\frac{P_z}{P + P_z} . r_0 \, , \tag{179}$$

$$\Delta y_{b_\theta} = \int_0^\theta r_\theta . \cos\theta \, . d\theta = \int_0^\theta \left(-\frac{P}{P+P_z} . r_0\right) . \cos\theta . d\theta = -\frac{P_z}{P+P_z} . y_{b_\theta}, \tag{180}$$

$$\Delta\left(z_{b_\theta} - z_c\right) = \int_0^\theta r_\theta . \sin\theta . d\theta = -\frac{P_z}{P+P_z} . \left(z_{b_\theta} - z_b\right). \tag{181}$$

On the grounds of the expression for the formula for the lever arm of form stability its increment will be as follows:

$$\Delta l_f = \Delta y_{b_\theta} . \cos\theta + \Delta\left(z_{b_\theta} - z_b\right) . \sin\theta = -\frac{P_z . l_{f0}}{P+P_z} . \tag{182}$$

The increment of the lever arm of weight stability will be determined using a familiar formula, too:

$$\Delta l_g = \frac{P_z}{P+P_z}\left[\left(-T-\frac{\Delta T}{2}+z_z\right)\sin\theta - l_g\right]. \tag{183}$$

Utilising the equations for static moments of volumes and weights we can find the increment of the height coordinate of the centre of buoyancy and the centre of weight using the following formulae:

$$\Delta z_b = z_{b_1} - z_b = \frac{V_{z_b}.z_b+\Delta V\left(T+\frac{\Delta T}{2}\right)}{V+\Delta V} - z_b = \frac{P_z}{P+P_z}.\left(T+\frac{\Delta T}{Z}-z_b\right) \tag{184}$$

$$\Delta z_g = z_{g_1} - z_g = \frac{\rho.V.z_g+\rho\Delta V.z_z}{V+\Delta V} - z_g = \frac{P_z}{P+P_Z}\left(z_z - z_g\right). \tag{185}$$

Then the increment of the lever arm of weight stability will be:

$$\Delta l_g = \frac{P_z}{P+P_z}\left[\left(-T-\frac{\Delta T}{2}+z_z\right)\sin\theta - l_g\right]. \tag{186}$$

Finally it is necessary to calculate the increment of the lever arm of stability at shifting the load in the distance y_z from the axis plane (i.e. by this formula for the lever arm of stability of form):

$$\Delta l' = -\frac{P_Z}{P+P_Z}.y_Z.\cos\theta. \tag{187}$$

Finally a new lever arm of static stability after the load reception will be as follows:

$$\Delta l_1 = l + \Delta l_f + \Delta l' - \Delta l_g = l + \frac{\pm P_z}{P\pm P_z}\left[\left(T\pm\frac{\Delta T}{2}+z_z\right)\sin\theta + l_g - l_f - y_z.\cos\theta\right], \tag{188}$$

$$l_1 = l + \frac{\pm P_z}{P \pm P_z}\left[\left(T \pm \frac{\Delta T}{2} + z_z\right)\sin\theta - l - y_z.\cos\theta\right], \qquad (189)$$

where: l - the lever arm of static stability until the load reception.

A new value of the righting moment will be as follows:

$$M_{V1} = (P + P_z).l_1 = M_V \pm P_Z\left[\left(T \pm \frac{\Delta T}{2} + z_z\right)\sin\theta - y_z.\cos\theta\right], \qquad (190)$$

where: M_{V1} is the moment until the load reception.

3.6.2. Construction of the Diagram of Static Stability of a Ship at any Displacement

In case of a significant change of the ship's draught the lever arms of static stability are determined using the formula $l = l_f - l_g$, using graphical dependencies of lever arms of form stability l_f by the displacement and angle of heel. Curves $l_f = f(V_1.\theta)$ are called interpolation curves and they are constructed in accordance with the following equation:

$$l_f = y_{b_\theta}.\cos\theta + (z_{b_\theta} - z_c).\sin\theta. \qquad (191)$$

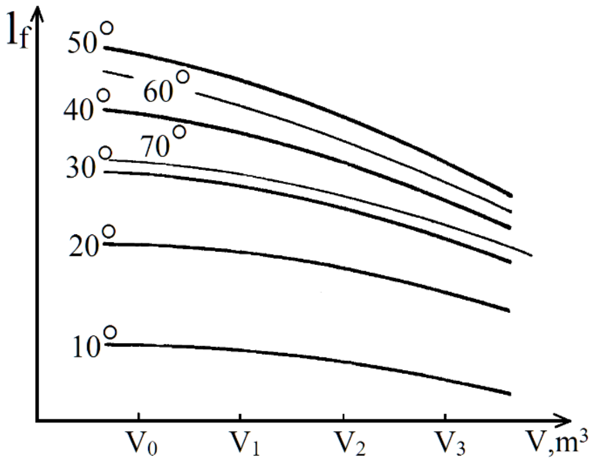

Fig. 41. Interpolation Curves (lever arms of form stability l_f by the displacement and the angle of heel) [Authors, 1]

3.6.3. Impact of a Free Liquid Cargo Surface on the Lever Arm of Static Stability

The impact of a free surface on the lever arm of static stability may be taken into account in formulae for coordinates of the centre of buoyancy

$y_{b_\theta} = \int_0^\theta r_0 . \cos\theta . d\theta$ and $\left(z_{c_\theta} - z_c\right) = \int_0^\theta r_0 . \sin\theta . d\theta$, where the metacentric radius $r_0^{'}$ must be corrected to the moment of inertia of a free liquid cargo surface:

$$r_\theta^{'} = \frac{I_{x_\theta}}{V} = \frac{\sum \rho_t . i_{x_\theta}}{\rho . V}. \qquad (192)$$

Values $y'_{b\,\theta}$ and $z'_{b\,\theta}$ will be used to calculate the lever arms of static stability l' according to the familiar formula.

Lever arms of static stability $l'$with considerations of free liquid cargo surfaces can also be determined approximately:

$$l' = l - \Delta l_{f_t} \, . \tag{193}$$

The value *l* is determined in the same way as in case of a solid load. The value Δl_{f1} - correction to the lever arm of form on a free surface in n-tanks is determined as follows:

$$\Delta l_{f_t} = \frac{1}{\rho.V} \sum_{j=1}^{n} \left(\left[y_{t_j,90} \cdot f_1(\theta) + z'_{t_j,90} \cdot f_2(\theta) + r_{t_j} \cdot f_3(\theta) \right] \rho_{t_j} .V.t_j \right) . \tag{194}$$

By analogy for approximate formulae for a ship as well as for tanks the following expressions are applied:

$$y_{t_j,90} = 0{,}5.b_{t_j} \left(1 - \frac{t_{t_j}}{h_{t_j}} \right) , \tag{195}$$

$$z'_{t_j,90} = 0{,}5.h_{t_j} \left(1 - \frac{t_{t_j}}{h_{t_j}} \right) , \tag{196}$$

$$r_{tj} = \frac{b_{tj}^2}{12_{tj}} , \tag{197}$$

where:

b_{t_j}, h_{t_j} - a middle breadth and height of the jth tank,

$f_1(\theta), f_2(\theta), f_3(\theta)$ - trigonometric functions applicable to the ship's type,

t_{t_j} - a thickness of the liquid cargo layer in the jth tank.

A high accuracy of formulae for approximate calculation is achieved in particular if the following is true:

$$h_{t_j} \geq t_t \rhd 0{,}08.b_t \tag{198}$$

3.7. DYNAMIC STABILITY AND THE DIAGRAM OF DYNAMIC STABILITY

Dynamic stability of a ship is its ability to tolerate dynamically acting heeling moments. The measure of dynamic stability is not the value of the righting moment, but the value of the work of the righting moment.

Indeed, if the heeling moment M_H acts on a ship dynamically, i.e. suddenly, or its acting on a ship changes rapidly (with a swoop) - its value from zero up to the computing value changes at a short instant which is shorter than the natural pitching period; then there in the initial period of heeling this moment exceeds the righting moment. As a result the ship heels with an angular acceleration and accumulates the kinetic energy of rotation within itself. Even after achieving the static angle of heel θ_S (point B) the ship will continue heeling unless the kinetic energy reserve is consumed.

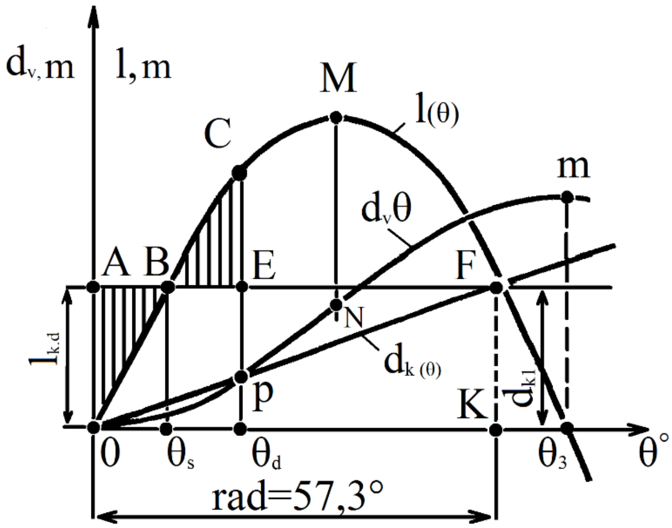

Fig. 42. Diagram of Dynamic and Static Stability (determination of the angle of a dynamic heel for a given dynamic moment) [Authors, 1]

After achieving the maximum angle of the dynamic heel θ_d the ship is under the impact of supplemental (excess) righting moment $\Delta M_R = \overline{CE}$. As a result of this fact the ship starts to compensate the acceleration and passes through the static equilibrium position again, however, to the contrary. This way the ship makes several rollings from one side to another one which are, however, damped due to resistance forces of water; finally the ship is stabilised in the position of static equilibrium determined with the angle θ_S which is significantly smaller than the angle of a dynamic heel.

To determine the angle of the dynamic heel θ_d it is necessary to compare the work of the dynamically heeling moment with the work of the righting moment (the work of resistance forces is disregarded under ideal conditions).

$$A_H = A_R \tag{199}$$

The work of the dynamically heeling moment is equal to the sum of elementary works in the heeling interval 0° to 0_d°

$$A_H = \int_0^{\Theta_d} M_{H_d}.d\emptyset. \tag{200}$$

If the dynamically acting heeling moment is constant, i.e. it is not dependent on θ, then:

$$A_H = P.d_H = M_{Hd}.\Theta_d. \tag{201}$$

In this case the specific work of heeling moment is as follows:

$$d_H = \frac{A_H}{P} = \frac{M_{Hd}\,.\,\emptyset_d}{P} \leq l_{Hd.}\,\theta_d\,, \tag{202}$$

where: θ_d - the angle of dynamic heel.

By analogy the work of righting moment is as follows:

$$A_R = \int_0^{\theta_d} M_R\,.\,d\theta. \tag{203}$$

The righting moment $M_R\ (\theta)$ as a function of the angle of heel is expressed via the diagram of static stability. Therefore the work of the righting moment is graphically represented with the area $0BC\theta_d$. Likewise, the work of the constant heeling moment is represented with the area of the rectangle $0AE\theta_d$.

Comparing these areas in accordance with the equation $A_H = A_R$ we can see they have a common part $0BE\theta_d$ and due to their identity it is necessary for the shaded areas to be equal, i.e. $0AB = BCE$.

This rule is used for a graphical solution of the equation $A_H = A_R$ using the diagram of static stability at a given constant dynamically acting moment M_{Hd} which is represented with a horizontal line AE and it is required to find a vertical CE so the shaded areas are identical. Then the intersection of the vertical CE with the horizontal axis (θ) brings the searched angle θ_d .

If angles of the dynamic heeling are not greater than $10 \div 15$, then sections of shaded areas are in an approximate straight scope of the diagram $0BC$. In this case we may assume that the angle of dynamic heel equals the double angle of static heel.

$$\theta_d = 2.\theta_S \tag{204}$$

It may be stated that in the scope of applying the metacentric formula for stability $\theta_d \leq (10° \div 15°)$ the angles of dynamic heel can be determined using the following equation.

$$\theta_d = \frac{2.M_{Hd}}{P.h_0} \tag{205}$$

Using the diagram of static stability we may find the value of the heeling moment causing overturning of the ship as well as the angle of the overturning.

For this case we need to consider such a horizontal line AE which will divide the diagram into identical (shaded) areas $0AB$ and BME (Fig. 43) showing an excess specific work of the heeling upsetting moment and allowable righting moment.

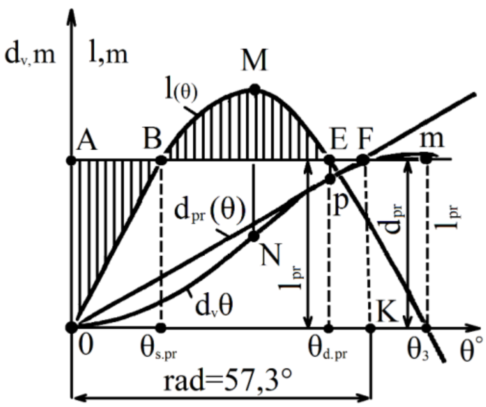

Fig. 393. Diagram of Dynamic and Static Stability (determination of the upsetting moment M_{pr} and the angle of overturning using both diagrams) [Authors, 1]

$$M_f = P.l_f \tag{206}$$

The value l_f is equal to the segment $0A$. The point E will match the angle of overturning the ship. The problems of dynamic stability verified above are better to be solved using the diagram of dynamic stability.

The diagram of dynamic stability is a curve showing the dependence of the work A_x of the righting moment M_R on the angle of heel θ.

An analytic expression of this dependency can be found in the formula $A_R = \int_0^{\theta d} M_R d\theta$, which can also be written in the following form:

$$A_R = P.\int_0^{\theta} l.d\emptyset = P.d_{v.} \qquad (207)$$

The value d_R represents the lever arm of dynamic stability, and from the physical point of view it is the change of the distance between the centre of weight and the centre of buoyancy in the vertical direction (the difference of segments $KC_\theta - GC = d_R$, see Fig. 34), i.e. the work A_v is the work of forces P and $\rho.g.V$ in their mutual distance.

The previous formula implies:

$$d_R = \int_0^{\theta} l.d\theta. \qquad (208)$$

If we substitute the value l into the expression under the integral, we will get:

$$d_R = y_{b_\theta}.\sin\theta - (z_{b_\theta} - z_b)\cos\theta - a(1 - \cos\theta). \qquad (209)$$

The diagram of dynamic stability is most frequently constructed in the coordinate system $(d_R;\theta)$ and the condition for dynamic stability in order to determine θ_R is written as follows:

$$d_R = d_H, \qquad (210)$$

where: $d_R = \frac{A_R}{P}$ and $d_H = \frac{A_H}{P}$ are measure works of the righting and heeling moment.

The diagram of dynamic stability is an integral curve in relation to the diagram of static stability. The coordinate d_R is determined as an integral with a variable upper limit according to the following scheme.

Table 2

Diagram of d_R Coordinate Determination

0°	l [m]	Even Sums $(2)\,[m]$	Sums (3) from Above (Integral) $[m]$	$d_v = \frac{1}{2}\Delta\theta(4) = \frac{0{,}175}{2}(4)$ $[m]$	$A_R = P.d_R.$ $kN.m$
1	2	3	4	5	6
0	0	0	0	0	0
10	l_{10}	$0+l_{10}$	$0+0+l_{10}$	$0{,}0875.(0+l_{10})$	$P.d_{10}$
20	l_{20}	$l_{10}+l_{20}$	$0+2.l_{10}+l_{20}$	$0{,}0875.(0+2.l_{10}+l_{20})$	$P.d_{20}$
30	l_{30}	$l_{20}+l_{30}$	$0+2.l_{10}+2.l_{20}+l_{30}$	.	$P.d_{30}$
40	.	.	.	.	.
50	.	.	.	.	.
60	.	.	.	.	.

Source: [Authors, 1]

If coordinates of the centre of buoyancy $y_{b_\emptyset}, z_{b_\emptyset}$ and the centre of weight z_g at any θ are known, then lever arms of dynamic stability can be determined using the formula mentioned above.

Properties of the diagram of dynamic stability as an integral curve:

- the coordinate axis θ is a tangent to the diagram of stability in the origin of the coordinate system,
- the maximum of the diagram of dynamic stability (point *M*) corresponds to the point of a fitting angle θ_z in the diagram of static stability,
- the diagram of dynamic stability contains an inflection point N at the angle of heel, corresponding to the maximum (point *M*) of the diagram of static stability.

If the heeling moment is of a constant value, then its work is determined using the formula $A_H = P.d_H = M_{H_d}.\theta_d$, and d_H is displayed as a line *OF*, since when $\theta = 0$ then $d_H = 0$, and when $\theta = 1\,rad$ then

$$d_{H_1} = \frac{(M_{H_d} . l)}{P} = l_{H_d} \tag{211}$$

The point P - point of intersection and thus also a point of equity $d_H\,(\theta)$ and $d_R(\theta)$, and it corresponds to the searched angle of dynamic heel θ_d in accordance with the equation $d_R = d_H$.

There in the diagram of dynamic stability it is, of course, possible to solve a reverse problem, too, i.e. to find a moment M_{H_d} with the given angle θ_d.

If the dynamic heeling moment M_{H_d} acts on a ship which has the initial angle of heel $\pm\theta$, then all solutions in the diagram of dynamic stability for solving these problems are made likewise above, however, from a new origin of the coordinate system θ'' that corresponds to the angle $\pm\theta_0$, and θ_d is calculated from the initial origin.

3.8. STANDARDISATION OF SHIPS' STABILITY

To ensure ships' stability in their operation is one of the most important tasks when ensuring the safety of their navigation.

While designing and constructing ships for inland navigation it is necessary to meet normative requirements for safety per the principles of the applicable classification society in terms of ship's stability. In accordance with these principles a ship is considered sufficiently stable in practical operation conditions (which are also set in these principles) if it fulfils the following conditions:

1. the initial metacentric height calculated with considerations for effects of free water surfaces has a positive value,
2. the basic (meteorological) measure of stability determined per the principles is satisfactory,
3. additional requirements for the stability determined per the regulations and depending on the ship's type and purpose are met.

Stability, as per the basic criterion, is considered sufficient if the heeling moment by a dynamic wind effect M_{wd} does not exceed the tolerable moment for a dynamic ship heeling M_{ALD}, i.e.:

$$M_{wd} \leq M_{ALD} \tag{212}$$

The heeling moment by a dynamic wind effect on a ship will then be:

$$M_{wd} = 0.01.P_w.A_w.z, \tag{213}$$

where:

P_w - the specific wind pressure determined in dependence on the tolerable ship area and the lever arm of the windward area z in [Pa];

A_w - the effective windward area in [m^2];

z - the lever arm of the windward area of a free-floating ship. In other words it is the distance of the centre of windward area (C_p) from the corresponding water line in a straight position of the ship in tranquil water in [m].

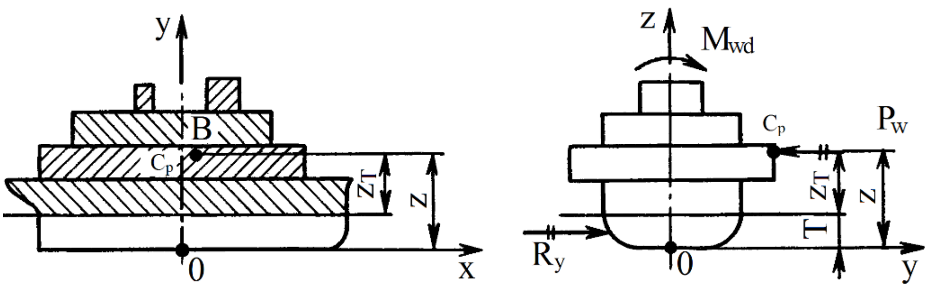

Fig. 404. Determination of Windward Area S_p and the Centre of Windward Area

[Authors, 1]

The effective windward area includes all projections of the area of the hull, superstructures, deck machines, deck cargo and other sections of the ship above the water surface onto its axis plane.

In constructions of a circular section individually placed on the deck the projection of the area is multiplied with a coefficient of flow around equal to 0.6.

In calculation of the windward area of lattice works the area defined with the construction's outline is multiplied with a coefficient of area infilling:

Table 5

Calculation of the Windward Area of Lattice Works

Type of a Lattice Work	Coefficient of Infilling [-]
Structural parts of floating cranes, machines, etc.	0.3 ÷ 0.6
Rails not filled with a net	0.2
Rails filled with a net	0.6

Source: [Authors, 1]

In constructions which overlap completely or partially in a side view the uncovered part of the rear area is included into the calculation of the windward area with its full value whereas the covered part of the area is included into the calculation only partially. The percentage ratio is set depending on the distance between the front and rear area.

Table 6

Stability Verification according to the Meteorological Measure

The distance between the front and rear area (a - height or breadth of the front area)	[%]
Less than a	0
$1a$ to $2a$	50
More than $2a$	100

Source: [Authors, 1]

The stability verification according to the meteorological measure is in fact a verification of the ship's stability by a wind effect with dynamic pressure.

The ship satisfies the meteorological measure when, at the least favourable state of the ship load, the tolerable moment for a dynamic heeling is equal to or greater than the heeling moment by dynamic wind pressure, i.e. the following condition is met:

$$M_{AL} \geq M_{WD}, \tag{214}$$

where:

M_{AL} - the tolerable moment for a dynamic heeling of the ship corresponding to the angle of flooding or the angle of overturning, if this one is smaller,

M_{WD} - the heeling moment by dynamic wind pressure which is set based on the specified conditions.

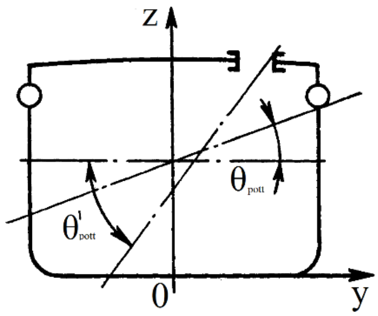

Fig. 415. Determination of the Angle of Flooding a Ship [Authors, 1]

The stability verification according to the meteorological measure is not required for the following types of ships:

1. cargo ships intended for transport of piece goods or loose bulk cargo in cargo space, tank ships and pusher tugs, in case the ratio of their dimensions is $L/_{B} < 9$ and $B/_{T} > 3$,

2. cargo ships transporting the cargo on the deck (full-scantling) provided the following conditions are met:

   - the ratio of main dimensions satisfies conditions given in point 1,

- the height of the centre of windward area of the body together with the cargo above the water line does not exceed 2 m.

The stability verification is performed according to the meteorological measure and according to additional diagrams of static and dynamic stability. Diagrams are constructed for all states of the load anticipated in classification society principles.

Table 7

State of Load for Individual Types of Ships

Number	**Type of Ship**	**State of the Load**
1	Passenger ships	1. without any passengers and cargo with 10 % of supplies 2. with 100 % passengers occupancy with luggage, 100 % of possible cargo and 10 % of supplies 3. with 10 % passengers occupancy with luggage, 100 % of possible cargo and 100 % of supplies
2	Cargo ships	1. without any cargo with 10 % of supplies 2. with 100 % of cargo and 100 % of supplies
3	Floating cranes	1. in transport with 10 % of supplies 2. at work with 100 % of cargo weight when in its least favourable position and with 10 % of supplies
4	Tugs, floating machines and other vessels	1. with 10 % of supplies, without any potential cargo 2. with 100 % of supplies and 100 % of potential cargo

Source: [Authors, 1]

Diagrams of stability are considered valid only up to the angle of heel at which the flooding of the inner ship space with water starts. In case of a heeling at a bigger angle it is supposed that the ship has completely lost its stability.

For ships with straight vertical sides the tolerable moment may be determined using the following formulae, i.e. without the application of diagrams of stability.

1. At dynamic effect of external forces:

$$M_{AL} = 0.0856.\Delta.MG.\theta_{AL}[kNm]. \tag{215}$$

2. At static effect of external forces:

$$M_{AL} = 0.1712.\Delta.MG.\theta_{AL}\,[kNm], \tag{216}$$

where:

Δ - the displacement of the ship [t],

MG - the initial metacentric height with a correction as a result of effects of free water surfaces of liquid cargoes [m],

θ_{AL} - the tolerable angle of a transverse heel determined per requirements of individual points $[^\circ]$.

The stability verification according to additional requirements is done based on the type of the ship. In case of ships which have the ratio of performance of main engines *N* and displacement ∇

$$N/_{\nabla} \geq 0{,}75, \tag{217}$$

it is necessary to verify the stability at circulation, even if the stability check is otherwise not required at circulation for the given type of the ship. The angle of static heel cannot exceed 60 % of the angle of flooding $\emptyset_{pott}$ at loading.

3.8.1. Passenger Ships

Passenger ships must be checked for the following additional requirements for stability:

1. The angle of static side heel at the most unfavourable distribution of passengers along the breadth and height of the ship must not exceed the angle at which 75 % of the freeboard immerses into water, however, this angle must not exceed 10º.

2. The angle of static side heel cannot exceed the angle of flooding $\emptyset_{pott}$, however, this angle must not exceed 12º, while:

   - simultaneous effect of heeling moments by the most unfavourable passengers' gathering on one side (M_{h1}) and by the centrifugal force at circulation of the ship (M_{h2}),

   - simultaneous effect of heeling moments by the least favourable passengers' gathering on one side (M_{h1}) and by the static wind pressure (M_{h2}).

The verification of stability of passenger ships according to additional requirements is done for prescribed states of load which correspond to the most hazardous number of passengers together with luggage and 10 % of supplies.

The heeling moment by static wind pressure M_{Vst} can be determined using the formula:

$$M_{Vst} = 0.001.P_{Wst}.A_v.\left(z + \frac{T}{2}\right)\ [kNm], \tag{218}$$

where:

P_{Wst} - the specific wind pressure at a static effect equals 50 % of wind pressure value P_v which is determined with the following table.

Table 8

Size of Lever Arm of the Windward Area of a Free-Floating Ship in a Respective Shipping Area

Shipping Area	**z** $[m]$					
	1	2	3	4	5	6
2	232	279	318	345	369	388
3	176	217	247	269	286	302

Source: [Authors, 1]

The heeling moment by centrifugal force at circulation of the ship M_{h2} can be set using the following formula:

$$M_{h2} = \frac{0{,}2.V.v^2}{L}.\left(z_g + \frac{T}{2}\right)[kNm], \qquad (219)$$

where:

v - the highest speed of the ship in tranquil water [m.s^{-1}],

z_g - the height of the centre of weight of the ship above the basic plane [m].

To determine the heeling moment by passengers' gathering we shall start from these conditions:

1. the distribution of passengers must represent the least safe gathering possible in standard operation situations on those decks which are accessible to them,
2. the number of passengers per *1 m*^{2} of free deck space is supposed to be as follows:
   - there on ship walkways with breadth up to *1 m*, lying along a bulwark or rail - 6 persons,
   - in passages between seats or tables - 4 persons,
3. the breadth of a seat for 1 person is *0.45 m*,
4. the computing weight of 1 person is *75 kg*,
5. the centre of weight of standing persons is considered at height of *1.1 m* above the deck plane, and of sitting persons at height of *0.3 m*.

3.8.2. Cargo Ships

Cargo ships intended for transport of cargo on the deck, for which the height of the centre of windward area above the water line exceeds 2 m, must satisfy the following additional requirement:

the heeling moment by static effect of wind pressure M_{Wst} cannot be greater than the tolerable moment for static heeling M_{AL}.

$$M_{AL} > M_{Wst} \tag{220}$$

where:

$$M_{Wst} = 0.001. P_{Wst}. A_v. \left(z + \frac{T}{2}\right)$$

M_{AL} - the tolerable moment for a static heeling of the ship corresponding to the angle of side heeling of the ship which equals 80 % of the angle of flooding θ_{pott}.

Tugs

Tugs are considered stable enough if their tolerable moment for a dynamic heeling M_{AL} is equal to or greater than the sum of the heeling moment by dynamic wind pressure M_{Wd} and the heeling moment by dynamic side component of tow-rope pull:

$$M_{AL} \geq M_{Wd} + M_t. \tag{221}$$

The heeling moment by dynamic side component of tow-rope pull can be determined using the following formula:

$$M_t = 1{,}1.S.(z_n - T) \ [kNm], \tag{222}$$

where:

S - the biggest pull in a tow rope measured during testing a tug on a fixed rope,

z_n - the height of the centre of pull in rope above the basic plane of the ship.

In case the pull in rope is not known the following values S [kN] are used in the calculation:

- at displacement of tugs $V \leq 30t$
  - 0.13N - tugs without a propeller nozzle
  - 0.20N - tugs with a propeller nozzle

- at displacement of tugs $V > 30\ t$
  - 0.16N - tugs without a propeller nozzle
  - 0.20N - tugs with a propeller nozzle

Where N is a total performance of main engines in [kN].

Besides additional requirements the stability of tugs must satisfy the condition that the angle of side heel by current effect of the heeling moment by dynamic wind pressure M_{Wd} and heeling moment by centrifugal force at circulation M_{h2} cannot exceed the angle of flooding θ_{pott} and this angle cannot be greater than $15°$.

Likewise, there in regulations of the classification society we may find additional requirements for floating cranes, floating machines, hydroplanes, ferries and other ships.

3.9. INFORMATION ON SHIP STABILITY

To assess the practical safety of ship navigation in different states of load there is a special document called Information on Ship Stability in every ship. This document contains the following materials:

a) Data on a ship and its stability for all typical (computing) states of load, the diagram of positioning non-closing openings, data from the heeling test, etc.

b) Instruction for the ship captain which contains guidelines for operation restrictions as well as recommendations needed to maintain the safety of the ship in some cases of operation of hazardous cargoes from the stability point of view, specific for a given ship.

This data and restrictions include: restrictions on shipping zone at the entrance to the circulation; introduction of safety measure for passengers getting on and off the board; determination of forbidden zones where passengers cannon occur; limitation of the number of passengers in individual cases of ship load and under certain hydro-meteorological circumstances of the navigation; restriction on the ship tonnage, restrictions on the height of placing and fixing the cargo on the deck; recommendations for the necessity to accept hard or liquid ballast when the ship is empty or when it runs out of fuel or other supplies; restriction on performance of tugs; no transfer of ships which have struck a shoal; restrictions on course angles during the navigation in heavy seas; data on behaviour character of a ship swinging to and fro during a storm; cases when side light windows (illuminators) need to be closed, etc.

c) Aid diagrams, tables, schemes of positioning the load and closing openings during a storm, and other data which enables to perform required calculations quickly and assess the ship stability in those states of load which are not taken into account in advance, as well as the captain's signature acknowledging that they have been familiarised with the "information" and adopted it in order to use it and abide by it.

4. UNSINKABILITY

Unsinkability means the ability of a ship to preserve a minimum required buoyancy and stability after flooding one or more compartments of the ship's hull. The unsinkability is ensured through a buoyancy reserve. A means to ensure unsinkability is the division of the hull into individual compartments using watertight bulkheads. The length of watertight compartments and their number depend on the buoyancy reserve of the ship (the depth of ship's side above water) and on the standardised number of simultaneously flooded (damaged) compartments. At the same time it is understood that segmentation of the hull into small compartments is undesirable from the operation point of view for majority of ships.

Longitudinal watertight bulkheads in ships are undesirable, too, because after flooding a compartment from one side the ship may turn over as a result of emergency heeling unless the compartment on the other side is flooded at the same time. The sheer of the hull or construction of a forebody and stern superstructure allows for maintaining the minimum required depth of the side above water on the edges of a ship at a strong trim in case of flooding the forepeak or afterpeak.

Finally, organisation-technical measures are of a big importance when ensuring the unsinkability: getting practice of the crew in a fast sealing of individual openings, preserving a good condition of the hull, watertight doors and windows (illuminators). The crew must be well familiar with consequences of flooding of any compartment or a group of compartments so it can always make the right decision - to immediately start with the evacuation of people from the ship, to intentionally strike a shoal or eliminate consequences of the disaster and continue in navigation provided the ship is navigable.

4.1. CALCULATION AND STANDARDISATION OF UNSINKABILITY

To calculate the unsinkability means to determine the state of buoyancy and stability of the ship after flooding a specific number of compartments. At the same time the hull strength must be verified with special calculations. Often there exists a reverse problem, i.e. to determine allowable dimensions of compartments at flooding of which the ship achieves specific parameters of buoyancy and stability.

The calculations of unsinkability can be performed on the basis of metacentric formulae for stability at flooding of compartments of the water volume less than 15 % of displacement, provided the heeling of the damaged ship is less than 15º. In case of flooding large compartments the calculations are given precision in the second approximation using methods which are generally given from more detailed materials of the ship theory. All calculations of unsinkability are usually made in the phase of designing a ship for which recommendations regarding required activities of the crew at a disaster are stated. The following lines will inform us on calculation methods using metacentric formulae for stability.

After flooding a compartment the volume and surface area of streamed water w_z and s_z are less than their theoretical values due to presence of machines, goods, mechanisms, etc. Their actual values are as follows:

$$w_z = \mu . w_{z0} \, , \tag{223}$$

$$s_z = \mu . s_{z0} \, , \tag{224}$$

where:

w_{z0} and s_{z0} are volume and surface area of the flooded compartment, calculated from the lines plan,

μ - a coefficient of compartment flooding which is recommended in principles of the classification society $\mu = 0.35 \div 0.98$.

In compliance with principles of classification society all ships have to be unsinkable in case of flooding of the forepeak and afterpeak; in case of passenger ships we speak about flooding of any compartment.

A ship is considered unsinkable if the water line of flooding of individual compartments on any place does not intersect the boundary curve of maximum flooding which is led there on the side of the ship 75 mm below non-closing openings of the ship. The initial metacentric height is calculated using the method of a constant displacement, where the following must be true: $h_1 \geq 0.005\ m$ and the maximum lever arm in the diagram of static stability must be $l_{max} \geq 0.10\ m$ at the diagram's fitting angle $\theta_z \geq 30°$ in case of a symmetric flooding or $\theta_z \geq 20°$ in case of an asymmetric flooding. Illuminators, hatchways and doors are considered non-closing openings in a ship.

4.1.1. Calculation of Unsinkability Using Metacentric Formulae

Flooded compartments can be divided into three categories:

Category 1 - fully flooded compartments (pressurised), e.g. compartments of the second bottom or compartments localised below the water line which can be water-tightly closed from all sides;

Category 2 - partially flooded compartments which are not in direct contact with the outer river water, e.g. compartments flooded through openings in the deck or in the hull which may be closed subsequently;

Category 3 - not fully flooded compartments which are in contact with surrounding river water through perforated sections of the hull or not closed openings. The level of water in this compartment is identical with the level of instant - actual water line of the ship.

Parameters of buoyancy and stability of a ship for categories 1 and 2 can be determined using familiar formulae for receiving and transferring a known load because the weight of water $P_z = \rho . g . \mu . w_{z0}$ and its centre can be calculated in advance. For flooding compartments of the category 2 it is also necessary to take the effect of a water level into account.

In case of flooding compartments of category 3 the calculation of unsinkability is done using the method of a constant weight displacement, or excluding the flooded compartment.

The principle of this method lies in the fact that the weight and the position of its centre remain constant and the form of the volume displacement changes as a result of excluding the volume of the flooded compartment in the scope of its entire height. The solution thus leads to the determination of geometric parameters (S_1, x_{f1}, I_{x1} and others) and the metacentric height of the hull which has changed in its part below water in accordance with known assumptions when compared to its initial form.

Let us introduce the following markings:

∇	- the volume displacement of the hull up to the original water line [m^3];
$w_z = \mu . w_{z0}$	- the volume of the flooded compartment up to the original water line (compartments of category 3) or the volume of water flooded into the compartment (for compartments of category 1 and 2) [m^3];
$s_z = \mu . s_{z0}$	- a lost water line surface area, i.e. area of a free water surface in the flooded compartment [m];
x, y, z	- coordinates of the centre of volume w_z of the flooded compartment [m];
a, b	- coordinates of the centre of a lost water line surface s_z (usually $a = x, b = y$), [m];
S	- a water line surface until compartment flooding [m];
x_f	- the coordinate of the centre of water line surface S until compartment flooding [m];
$S_1 = S - s_z$	- a water line surface until compartment flooding [m^2];

x_{f1}, y_{f1}	- coordinates of the centre of surface S_1 (in the figure point B_1) which due to a small addition of draught ΔT can be considered as coordinates of the centre of additional water layer [m];
I_x, I_f	- main central moments of inertia of water line surface S_1 after flooding a compartment [m^4];
i_{x11}, i_{y11}	- main moments of inertia of lost water line surface with regard to inherent central axes, transverse x_{11} and longitudinal y_{11} [m^4];
x_B, z_B	- coordinates of the centre of buoyancy B until compartment flooding [m];
x_{B_1}, y_{B_1}, z_{B_1}	- coordinates of the centre of buoyancy $B_1^{'}$ after compartment flooding provided the ship floats without a heeling and inclination [m];
T	- the initial middle draught [m];
T_F, T_A	- the forward draught and aft draught [m].

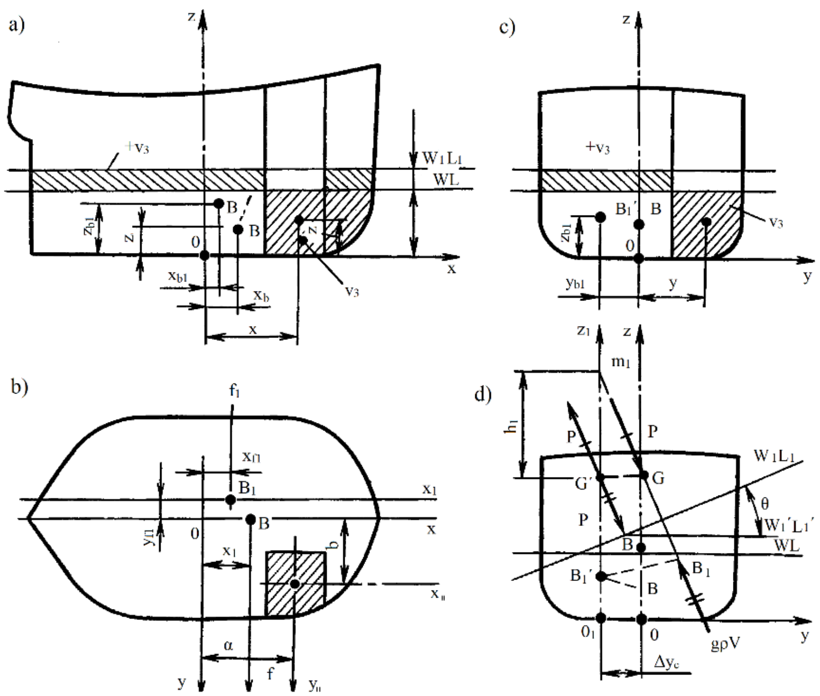

Fig. 426. Diagram to the Calculation of Unsinkability Using the Method of a Constant Weight Displacement according to GPS-Metacentric Formulae for Stability [Authors, 2]

a), b), c) a position and characteristics of a water line and ship displacement after excluding from the hull volume of the flooded compartment (in respective planes); d) a scheme of forces acting on a ship after flooding a compartment

After flooding a given compartment the ship gains an additional middle draught ΔT and turns slightly around axes passing through the centre of a new water line surface S_1 (point B_1); angles of its heel and inclination will be determined with new values of metacentric heights, i.e. new values of coordinates of the displacement, moment of a water line surface S_1 and metacentric radiuses of the ship with a changed hull form.

The addition of a middle draught ΔT can be determined on the basis of the following condition: when $P = g.\rho.V$, the lost volume of displacement w_z will be equal to a supplemental layer of displacement with the surface area equal to S_1, and then:

$$w_z = \Delta T.S_1 = \Delta T(S - s_z), \tag{225}$$

wherefrom:

$$\Delta T = \frac{w_z}{(S - s_z)}. \tag{226}$$

Coordinates of the centre and the position of central axes of the water line S_1 can be determined from equations for static moments of this surface area with regard to the *y-axis* and *x-axis:*

$$S.x_f - s_z.a = (S - s_z).x_{f_1}, \tag{227}$$

$$S.0 - s_z.b = (S - s_z).y_{f_1}. \tag{228}$$

Then

$$x_{f_1} = \frac{S.x_{f_1} - s_z.a}{S - s_z} = x_f - \left(a - x_f\right)\frac{s_z}{S - s_z}, \tag{229}$$

$$y_{f_1} = -\frac{b.s_z}{S - s_z}. \tag{230}$$

Now we will determine the addition of coordinates of the buoyancy centre after the hull immersion to ΔT. For the purpose of this calculation we will construct equations of moments with regard to planes y_{0z}, x_{0z}, x_{0y} where the lost volume of the compartment will be considered negative and equal to the volume of the supplemental layer which is positive.

$$V.x_B - w_z.x + w_z.x_{f_1} = V.x_{B_1}, \tag{231}$$
$$V.0 - w_z.y + w_z.y_{f_1} = V.y_{B_1}$$
$$V.z_B - w_z.x + w_z.\left(T + \frac{\Delta T}{2}\right) = V.z_{B_1}.$$

If we substitute values for x_{f1} and y_{f1} into these equations, then for increments of coordinates of the centre of buoyancy after algebraic modifications the following is true:

$$\Delta.x_B = x_{B_1} - x_B = -\frac{w_z}{V}\left[\left(x - x_f\right) + \left(a - x_f\right)\frac{s_z}{S - s_z}\right], \tag{232}$$

$$\Delta.y_B = y_{B_1} - 0 = -\frac{w_z}{V}\left[y + b\frac{s_z}{S - s_z}\right],$$

$$\Delta.z_B = z_{B_1} - z_B = -\frac{w_z}{V}\left[z - \frac{\Delta T}{2} - T\right].$$

The main moment of inertia of the water line surface S_1 with regard to the x_1-axis passing through the centre B_1 is determined using the formulae for transition to parallel axes. Thus from the moment of inertia I_x of the area S with regard to the x-axis we can subtract the inherent moment of inertia i_{x11} and the transfer moment of inertia of the area s_z, so we will get the moment of water line S_1 with regard to the x-axis:

$$I'_x = I_x - i_{x_{11}} - b^2 . s_z . \tag{233}$$

Then for the transition to the x_1-axis from I'_x we can subtract the transfer moment of the area S_1 which equals $y_{f1}^2 . (S - s_z)$.

This way we will get

$$I_x = I_x - i_{x_{11}} - b^2 . s_z - \ v_{f_1}{}^2 (S - s_z). \tag{233}$$

Likewise, the main moment of inertia of the water line surface S_1 with regard to central transverse axis f_1 will be:

$$I_{f_1} = I_f - i_{y_{11}} - s_z (a - x_f)^2 - (S - s_z)(x_f - x_{f_1})^2 . \tag{234}$$

After substituting values for x_{f1} and y_{f1} and after some simple algebraic modifications we will get:

$$I_{x_1} = I_x - i_{x_{11}} - s_z . b^2 [1 + \frac{s_z}{(S - s_z)}], \tag{235}$$

$$I_{f_1} = I_f - i_{y_{11}} - s_z . (a - x_f)^2 [1 + \frac{s_z}{(S - s_z)}]. \tag{236}$$

Then with a constant displacement V new values of metacentric radiuses will be:

$$r_1 = \frac{I_{x_1}}{V} = r - \frac{i_{x_{11}} - s_z . b^2 [1 + \frac{s_z}{(S - s_z)}]}{V}, \tag{237}$$

$$R_1 = \frac{I_{f_1}}{V} = R - \frac{i_{y_{11}} - s_z . (a - x_f)^2 [1 + \frac{s_z}{(S - s_z)}]}{V}. \tag{238}$$

Increments or decrements of metacentric radiuses will then be:

$$\Delta r = r_1 - r = - \frac{i_{x_{11}} - s_z . b^2 [1 + \frac{s_z}{(S - s_z)}]}{V}, \tag{239}$$

$$\Delta R = R_1 - R = - \frac{i_{y_{11}} - s_z . (a - x_f)^2 [1 + \frac{s_z}{(S - s_z)}]}{V}. \tag{240}$$

New values of metacentric heights and their changes will then be:

$h_1 = h_0 + \Delta h$
$H_1 = H_0 + \Delta H$
$\Delta h = \Delta r + \Delta z_c - \Delta z_g$
$\Delta H = \Delta R + \Delta z_c - \Delta z_g$

Or in case of a constant displacement when $\Delta z_g = 0$, the following is true:

$$\Delta h = \Delta r - \Delta z_B = -\frac{i_{x_{11}} - s_z . b^2 \left[1 + \frac{s_z}{(S - s_z)}\right]}{V} - \frac{w_z}{V}\left(z - \frac{\Delta T}{z} - T\right), \tag{241}$$

$$\Delta H = \Delta R - \Delta z_B = -\frac{i_{y_{11}} - s_z . (a - x_f)^2 \left[1 + \frac{s_z}{(S - s_z)}\right]}{V} - \frac{w_z}{V}\left(z - \frac{\Delta T}{z} - T\right). \tag{242}$$

Metacentric heights h_1 and H_1 are examined along the z_1-axis passing through a new centre of buoyancy B'_1 in a non-heeled position of the ship.

However, after a compartment flooding the ship gains a heel and inclination due to its turning around x_1 and x_{f1}-axes which pass through the centre G_1 of water line surface S_1 and the ship gains a new equilibrium position which is determined with the water line WL_1. At that time the centre of buoyancy is shifted from the point B'_1 into the point B_1 which lies on one vertical with the centre of gravity of the ship: $B_1 G \perp WL_1$.

Thanks to the projection of the point G onto the z_1-axis we will get the point G' which corresponds to the position of the centre of gravity of the ship floating without a heeling on the water line WL'_1. In the point G' we apply two forces acting against each other P parallel to $\overline{B_1 G}$. Then for the ship heeled at the angle θ it may be stated that it occurs in equilibrium with the effect of the heeling moment $P.\overline{GG'.\cos\theta}$ $P.\overline{GG'}.\cos\theta$ and the righting moment $P.h_1.\sin\theta$, i.e.:

$$P|\Delta y_B|.cos\theta = P.h_1.sin\ \theta. \tag{243}$$

After substituting the absolute value of Δy_B into this equation the angle of heel can be determined:

$$tg\theta \approx \theta = \frac{\Delta y_B}{h_1} = \frac{g.\rho.w_2[y+b\frac{s_Z}{(S-s_Z)}]}{P.h_1}. \quad (244)$$

Analogically the inclination of the ship, or its angle can be determined:

$$tg\Psi \approx \Psi = \frac{\Delta z_B}{H_1} = \frac{g.\rho.w_2[(x-x_f)+(a-x_f)\frac{s_Z}{(S-s_Z)}]}{P.H_1}. \quad (245)$$

The computing precision will not change if we take $x = a, y = b$. Then the formulae for the angle of heel and inclination can be simplified as follows:

$$\theta = \frac{g.\rho.w_2.y[1+\frac{s_Z}{(S-s_Z)}]}{P.h_1}, \quad (246)$$

$$\Psi = \frac{g.\rho.w_2(x-x_f)[\frac{s_Z}{(S-s_Z)}]}{P.H_1}. \quad (247)$$

Based on the angle of inclination the values of forward draught and aft draught will be:

$$T_{F_1} = T_F + \Delta T + \left(\frac{L}{z} - x_{f_1}\right).\Psi, \quad (248)$$

$$T_{A_1} = T_A + \Delta T + \left(\frac{L}{z} - x_{f_1}\right).\Psi. \quad (249)$$

In case of flooding a group of compartments of different categories they will be replaced with a single equivalent compartment which (as it has already been mentioned) has characteristics for one compartment.

The volume of an equivalent compartment:

$$w_z = \sum_{1,2,3} \mu_1.w_{z,0i}, \quad (250)$$

where:

$w_{z,0i}$ - for compartments of category 3 the value is taken up to the resulting water line, and for compartments of categories 1 and 2 factual volumes of flooding are calculated.

Note: Under the sum sign there are symbolically marked categories of compartments out of which the sum is performed.

Coordinates of the centre of equivalent volume:

$$x = \frac{\sum_{1,2,3} \mu_i . w_{z,0i} . x_i}{\sum_{1,2,3} \mu_i . w_{z,0i} .} , \tag{251}$$

$$y = \frac{\sum_{1,2,3} \mu_i . w_{z,0i} . y_i}{\sum_{1,2,3} \mu_i . w_{z,0i} .} , \tag{252}$$

$$z = \frac{\sum_{1,2,3} \mu_i . w_{z,0i} . z_i}{\sum_{1,2,3} \mu_i . w_{z,0i} .} . \tag{253}$$

where:

x_i, y_i, z_i are coordinates of centres of individual volumes $w_{z,0i}$.

The surface area of the equivalent compartment is determined with a sum of compartments areas of category 3 only.

$$s_z = \sum_{3} \mu_i . s_{z,0i} \tag{254}$$

Inherent moments of inertia of a lost area are summed for compartments of categories 2 and 3.

$$i_{x_{11}} = \sum_{2,3} i_{x,11i} \tag{255}$$

$$i_{y_{11}} = \sum_{2,3} i_{y,11i} \tag{256}$$

Transfer moments of inertia and coordinates of centres of lost areas are determined for compartments of category 3 only.

$$s_z.b^2 = \sum_3 .s_{zi}.b_i^2 \quad \text{or} \quad b = \sqrt{\frac{\sum_3 .s_{zi}.b_i^2}{s_z}} \tag{257}$$

$$s_z.a^2 = \sum_3 .s_{zi}.a_i^2 \quad \text{or} \quad a = \sqrt{\frac{\sum_3 .s_{zi}.a_i^3}{s_z}} \tag{258}$$

5. OSCILLATION OF A SHIP

Oscillation is a periodic movement of a ship as of a hard body during its floating in smooth water or in waves. Oscillating movement of ships is caused with external forces. A yawed body from the point of view of equilibrium stability is under effect of righting moments attempting to return the body back to its initial position. Out of six possible kinds of movement (three of them are shifting ones along x, y, z-axis and three of them are rotating ones around these axes) the following three are crucial for a ship (remaining three kinds of movement are monotonously damped after external forces stop acting):

a) vertical oscillation - dipping along the z-axis causing a decrease or increase of the ship draught,

b) transverse oscillation along the longitudinal x-axis causing a variable transverse heeling to the starboard side and port side of the ship,

c) longitudinal oscillation along the transverse y-axis causing a variable longitudinal inclination toward the forebody and stern of the ship.

All three types of ship oscillation usually happen at the same time, although their study is performed separately mainly due to simplifying the solution of problems related to ship oscillation.

Parameters (numeric characteristics) of any oscillation include the following:

- a period, i.e. a period of a full amplitude of oscillation which is characterised with smoothness or jerky movement of oscillation,
- a circular frequency, i.e. a number of the entire oscillation change in 2π.sec,
- an amplitude, i.e. a maximum deviation from the equilibrium position.

By the value of amplitude it is possible to assess the degree of oscillation from the physiological point of view. For vertical oscillation - dipping, the amplitude means the biggest change of the draught; for side oscillation it means the angle of heel, and for longitudinal oscillation it means the angle of inclination.

We distinguish free rolling and forced oscillation of a ship in case of all three types of oscillation.

Free rolling in tranquil water can be observed after the forces which caused the disequilibrium of the ship and then left it in free running, stop acting. Free rolling in tranquil water is quickly damped by forces of water resistance. Nevertheless, free rolling affects parameters of forced oscillation significantly. That is why it is required to determine parameters of free rolling, too.

Forced oscillation is evoked with periodically changing forces of water pressure which are caused with raising or lowering the water level on sides of the ship during its navigation in heavy see, acting for a relatively long time.

Oscillation evokes hard consequences which worsen the standard operation of the ship. The most fundamental consequences are: harmful effects on physiological condition of passengers and crew ("seasickness"); reduction of speed; flooding of the deck; the ability to waterlog or damage the cargo; the formation of additional inertia forces which as a result require a bigger consumption of materials in order to ensure the strength of the hull and superstructures; deterioration of operation of mechanisms and instruments resulting from the effect of dynamic forces arising at oscillation.

In some cases, deck flooding and high amplitudes and frequencies may cause disasters (or may lead to the stability loss and ship turning over mainly at side oscillation).

For self-propelled ships of inland navigation which float in big lakes and water reservoirs during a storm, the side oscillation plays a significant role with the amplitudes reaching up to $20^{\circ} \div 30^{\circ}$. Longitudinal and vertical oscillation of these ships can be felt only slightly. In practice vertical and longitudinal oscillation can be disregarded if the ratio of the length of the ship's hull to the length of a wave is greater than 1.5.

The following categories of forces act on an oscillating ship:

- weight, resultant of which acts in the centre of gravity,
- inertia forces,
- hydrostatic forces which attempt to return the oscillating ship into its equilibrium position,
- resistance forces of the environment affecting the ship oscillation, or so called damping forces which are dependent on the speed of ship oscillation,
- excitation forces - supplemental periodically changing forces of water pressure which are dependent mainly on waves parameters.

The accuracy of solving problems from the theory of ship oscillation depends on the completeness and accuracy of determining the specified force categories. The best and most accurate calculations are made within a so called "hydrodynamic theory of ship oscillation" which takes the change of water pressure (field theory) caused with a ship in an oscillating movement into account.

If values of righting moments are considered in the linear form by oscillation amplitude, e.g. based on metacentric formulae for stability, and if values of resistance are linear dependent on components of speed of the ship's oscillating movement, then the solution will be simplified significantly. Similar solving methods are called a theory of linear ship oscillation.

When considering the actual (non-linear) dependence of the above mentioned forces the differential equations of oscillation become non-linear. They are the subject of study of "theory of non-linear oscillation". Finally depending on the study of wavy motions of water causing forced oscillation of a ship we distinguish oscillation in a regular or irregular sea when calculation methods are based upon foundations of mathematical statistics and probability theory.

5.1. FREE ROLLING OF A SHIP IN CALM WATER

Let us consider the following simplifying prerequisites which form the theoretical foundations of ship oscillation in calm water:

1. We study oscillations with small angles of heel (with a small amplitude) when the righting moment may be determined based on metacentric formulae for stability.

2. Heelings of ship are considered equivolume.

3. The oscillation axis passes through the centre of gravity of the ship, i.e. the point G, which enables to assume that the moment of inertia forces equals a multiple of the angular acceleration $\theta^{''} = d^2 \cdot \theta / \, df_2$ and the moment of inertia of ship weight I_x^0 with regard to this axis. The origin of coordinates is chosen in the centre of gravity of the ship. Detailed investigations confirm that the longitudinal axis of ship heeling at the side oscillation is really located near the actual water line and it is shifted towards the centre of gravity of the ship on average in 0.2 ÷ 0.35 of the distance between the water line and the centre of gravity.

4. As a result of the fact that ship heelings are done with a variable angular speed and acceleration, water reactions will also evoke hydrodynamic forces and moments of inertia. Through effects of these forces and moments which are called virtual and concurrent, water weights are considered approximately in that way that the weight of the ship and the moment of inertia of this weight increase up to a certain value.

5. Moment of forces of water resistance against the oscillating movement of a ship with regard to the axis passing through its centre of gravity may be considered proportional to the first derivative of the angular speed $\theta_1 = \frac{d\theta}{dt}$. These resistance forces arise due to water viscosity.

When considering the above mentioned prerequisites the heeled ship ($+\theta$) will be under effect of the following moments of forces with regard to the longitudinal axis passing through the centre of gravity of the ship:

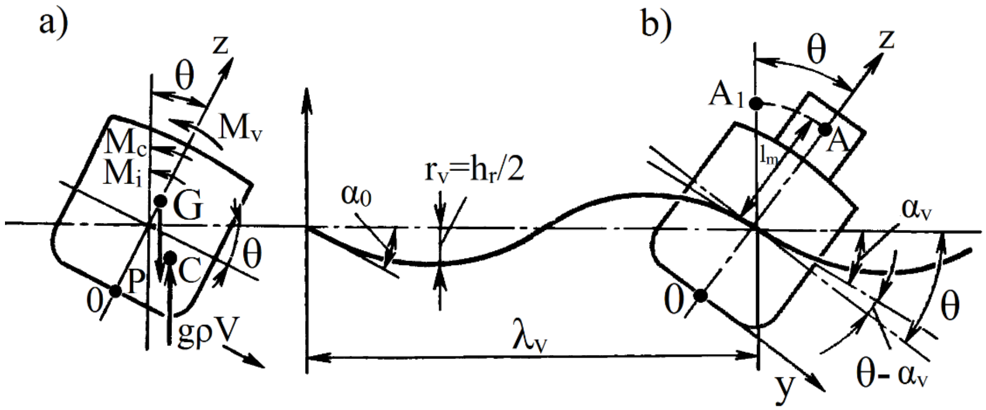

Fig. 437. Figure to the Calculation of Ship Oscillation Characteristics [Authors, 2]

a) forces and moments acting on a ship at free rolling in calm water; b) elements of waves and a position of a ship at rolling in waves

- the righting moment $M_R = P \cdot h_0 \cdot \theta$ acting counterclockwise,
- the moment of damping forces, i.e. moment of resistance $M_t = N.\theta^{\prime}$ which also acts against the angle θ (coefficient N of moment of resistance forces at oscillation is determined experimentally using approximate formulae),
- the moment of inertia forces which along with the moment of inertia of co-acting weights ΔI_x^0 is equal to $M_i = (I_x^0 + \Delta I_x^0).\theta^{\prime\prime}$ and according to D'Alembert's principle it acts in an opposite direction from the angular acceleration, i.e. counterclockwise, too.

If the sum of the mentioned moments is likened to zero in accordance with the known D'Alembert's principle we will get a differential equation of a side oscillation of a ship in calm water.

$$-(I_x^0 + \Delta I_x^0) \cdot \theta'' - N \cdot \theta' - P \cdot h_0 \cdot \theta = 0 \qquad (259)$$

The practice confirms that the moment of resistance forces affects the frequency only slightly. Thus the moment of these forces may be disregarded and the differential equation may be rewritten as follows:

$$\theta'' + w_\theta^2.\theta = 0 , \qquad (260)$$

where:

$$w_\theta = \sqrt{\frac{P.h_0}{\left(I_x^0 + \Delta I_x^0\right)}} , \qquad (261)$$

where:

w_0 - the angular frequency of free ship rolling without consideration of resistance forces [sec^{-1}],
h_0 - the metacentric height [m].

The solution of the given 2nd order linear differential equation without the right part may be of the following form:

$$\theta = C_1.\sin w_\theta.t + C_2.\cos w_\theta.t . \qquad (262)$$

In order to determine integration constants we will apply the following limit and initial conditions at time $t = 0$, when the ship was heeled at the angle θ_0 and it has been left without a supplemental impulse, i.e. without any initial velocity $\theta_0' = 0$.

Then the first condition determines $C_2 = \theta_0$ and the second condition determines $C_1 = 0$, and thus the general integral of the equation under given conditions will be as follows:

$$\theta = \theta_0.\cos w_0.t . \qquad (263)$$

This equation conveys the meaning that a ship will undergo some oscillating motions of cosine curve shape with a period 2π, i.e. $2\pi = \omega_\theta.\tau_\theta$.

Wherefrom the period of an inherent ship oscillation will be as follows:

$$\tau_{\theta} = \frac{2\pi}{w_{\theta}} = 2\pi\sqrt{\frac{\left(I_x^0 - \Delta I_x^0\right)}{P.h_0}} \quad . \tag{264}$$

The moment of forces and inertia of weight can be expressed using the radius of inertia (physical pendulum):

$$I_x^0 - \Delta I_x^0 = \frac{P.r_x^2}{g} \quad , \tag{265}$$

where:

r_x - the radius of inertia of a ship weight [m] which can also be expressed in segments of a crucial parameter of the hull breadth as $r_x = k.B$.

Then the period of one beat will be:

$$\tau_{\theta} = \frac{K_{\tau}.B}{\sqrt{h_0}} \quad . \tag{266}$$

The coefficient K_τ has quite a stable value for ships of a certain class and it changes a little after changing the ship draught. For river ships the value of the coefficient is as follows:

- cargo ships K_τ =(0.75÷ 0.85) [$s.m^{-0.5}$]
  (a smaller value for a fully loaded ship; a greater value for an empty ship),
- passenger ships K_τ =(0.83÷ 0.86) [$s.m^{-0.5}$],
- pusher tugs K_τ =(0.62÷ 0.72) [$s.m^{-0.5}$].

Formulae for the oscillation period make it clear that the higher the metacentric height of a ship is, the smaller the period of inherent side rolling of a ship is, i.e. the excess stability worsens the smoothness of ship oscillation because it causes tugging oscillation with a small period and a high frequency.

To calculate the period of a free rolling with small amplitudes it is possible to apply the following approximate formulae for determining the moment of inertia of ship weight and co-acting water weight:

$$I_x^0 = \frac{P.\left(B^2\, 4z_g^2\right)}{12g}$$ - Duaer's formula, (267)

$$I_x^0 = \frac{0{,}3.\frac{B}{T} - 0{,}75}{38{,}2}.\left[\rho.L.B.T.\left(B^2.4T^2\right)\right]$$ - prof. Pavlenko's formula. (268)

The period of a free longitudinal rolling of a ship is approximately determined using the following formula:

$$\tau_\psi = 2\pi\sqrt{\frac{0{,}98.C_B\left(1+0{,}33\frac{B}{T}\right)}{g.C_W}.T} \approx \left(2{,}7 \div 3{,}0\right).\sqrt{T}$$. (268)

The period and frequency of a free rolling at small amplitudes is determined with a test easily. To perform this test we need a calm bay where the ship is rolled as a result of a group of people running at speed $v \approx \frac{2.B}{\tau_\theta}$ from one side to another one (the number of persons is usually set so that 50 t of displacement fall on 1 person).

After the ship is rolled people are arranged at a command in the axis plane. Several full ship oscillations, i.e. the period of oscillation, are measured with a stopwatch. Oscillations of a ship may also be recorded in an inclinograph. When τ_θ is got from the test it is possible to use the given formula and calculate h_0 and finally z_g, too. Please note that values h_0 and z_g determined this way will be less accurate than those ones determined in a heeling experiment. Nevertheless, the period of a free rolling can experimentally be determined only with difficulties and therefore it is necessary to use accurate calculations which consider the entire complexity of the free ship rolling process.

5.2. ROLLING OF A SHIP IN WAVES

Steady and regular see means a process of origin and duration at the time of unchanging waves when a new oncoming wave is a true duplicate of the previous wave. Dimensions of waves on the surface of water aquatories are dependent on dimensions of the surface and depth of the aquatory as well as on the force, direction and time of the effect of the wind. Waves are characterised with length λ_v [m]; height h_v [m]; maximum angle of wave slope α_0 [rad]; period τ_v [*s*], frequency σ_v [s^{-1}] and the velocity of spreading c_v [$m.s^{-1}$] (Fig. 49).

In the theory of steady 2-dimensional trochoidal (dead) waves the following approximate relations among their characteristics are used:

$$c_v = \frac{\lambda_v}{\tau_v} \approx \sqrt{\frac{g}{2\pi}.\lambda_v} \approx 1{,}25\sqrt{\lambda_v}\,, \tag{269}$$

$$\tau_v = \frac{\lambda_v}{c_v} \approx \sqrt{\frac{2\pi}{g}.\lambda_v} \approx 0{,}8\sqrt{\lambda_v}\,, \tag{270}$$

$$\alpha_0 = 2\pi\frac{r_v}{\lambda_v} = \pi\frac{h_v}{\lambda_v}\,. \tag{271}$$

The change of the angle of a wave slope over time is determined using the formula:

$$\alpha_v = \alpha_0.\cos\sigma_v.t\,. \tag{272}$$

Further we will assume that ship dimensions (both *B* and *T*) are small when compared to the length of a wave λ_v. At the deviation of a ship at the angle θ from its straight position it may be assumed that the ship heels at the angle $\varphi = \theta - \alpha_v$ with regard to the surface area of water. In ship there will act the same moments of forces as in calm water at free rolling, however, the moment of inertia forces of the ship will depend on the absolute value of acceleration θ'', and all others will depend on the angle with regard to water, i.e. with regard to the wave slope $(\theta - \alpha_v)$.

As a result the differential equation of free rolling in regular sinusoid sea (by analogy with the equation for free rolling of a ship) will be as follows:

$$-I_x^0\theta'' - \Delta I_x^0\left(\theta'' - \alpha_v''\right) - N\left(\theta' - \alpha_v'\right) - P.h_0\left(\theta - \alpha_v\right) = 0 \quad (273)$$

If resistance forces as well as $I_x^0 . \alpha_v^{''}$ are disregarded and provided that $\alpha_v = \alpha_0 . \cos\sigma_v . t$, the differential equation may be formed as follows:

$$\theta'' + w_\emptyset^2\theta + w_\theta^2 \cdot \alpha_0 \cdot \cos\sigma_v \cdot t. \quad (274)$$

When compared to the equation for free rolling, the differential equation contains a right side. The complete integral of the equation equals the integral of the equation without the right side, i.e. for free rolling, plus any partial solution with consideration of the right side.

Free rolling due to acting of resistance forces damps quickly and thus we will mainly be interested in the partial solution of the equation with the right side which characterises undamped forced oscillation. We will be looking for the solution in a form:

$$\theta = \theta_m . \cos\sigma_v . t \quad (275)$$

After a double derivative of this expression and substitution into the differential equation we will get:

$$-\theta_m \cdot \sigma_v^2 \cdot \cos\sigma_v \cdot t + \theta_m \cdot w_\theta^2 \cdot \cos\sigma_v \cdot t = w_\theta^2 \cdot \alpha_0 \cdot \cos\sigma_v \cdot dd, \quad (276)$$

wherefrom:

$$\theta_m = \alpha_0 \frac{1}{1 - \left(\frac{\sigma_v}{w_\theta}\right)^2} = \alpha_0 \frac{1}{1 - \left(\frac{\tau_v}{\tau_\theta}\right)^2} \quad (277)$$

And thus the amplitude of an evoked ship oscillation in waves will be

$$\theta = \alpha_0 \frac{1}{1 - \left(\frac{\tau_v}{\tau_\theta}\right)^2} \cos\sigma_v . t \quad (278)$$

The following conclusions may be drawn from this equation:

- if $\tau_\theta \to 0$ resulting from $h_0 \to \infty$, or
- if $\tau_\theta \to \infty$ resulting from $\lambda_v \to \infty$,

then the maximum amplitude of the evoked oscillation equals $\theta \to \alpha_0$. This situation occurs if a ship with a high metacentric height or a ship floating in waves of a very big length is under a strong effect of oscillation and it duplicates the wave all the time (such as a small floating piece of wood in water). N. Y. Zhukovsky, father of Russian ship navigation, explained it pretty illustratively: "metacentric height is the lever through which a wave swings a ship".

If $\tau_\theta \to 0$ when $h_0 \to 0$, or $\tau_\theta \to 0$ when $\lambda_v \to 0$ (in case of a little stable ship), then the ship does not oscillate $\theta = 0$, i.e. the ship will constantly be in a vertical position like a floating bottle half filled with water.

If $\tau_\theta \to \tau_v$, i.e. in case of identical periods of the inherent oscillation and wave there arises the resonance when in theory it is true that $\theta \to \infty$. In fact, however, as a result of effects of friction resistance forces and final dimensions of waves when compared to the breadth and draughts of the ship, the maximum amplitude of the evoked oscillation in waves has a final value and is determined using a specified formula from the theory of a non-linear ship oscillation:

$$\theta_m = \chi_{T/\lambda_v} . \chi_{B/\lambda_v} . \alpha_0 \frac{\sqrt{(1-q^2)+4\mu_r^2}}{2.\mu_r} , \qquad (279)$$

where:

$\chi^{T}/_{\lambda_v}, \chi^{B}/_{\lambda_v}$ - reduction coefficients which are determined depending on radiuses and $^{T}/_{\lambda}$ and $^{B}/_{\lambda}$,

$q = \frac{\Delta I^0}{I_x^0 + \Delta I^0}$ - a dimensionless coefficient determined either with approximate formulae for moments of inertia of weight, or approximately $q \approx 0.15 \div 0.25$,

μ^r - a relative dimensionless coefficient of resistance determined with approximate formulae, or equal to $\mu_r = 0.5 \div 0.18$.

The real life practice confirms that waves are usually of an unstable random character. Yet, as observations confirm, the majority of ship oscillation in such sea happens with a period equal to the period of the inherent oscillation τ_θ and with an amplitude which is not greater than the one determined using the given approximate formula.

It is a result of these facts that τ_θ and θ_m are main and crucial characteristics of the ship oscillation in waves.

5.3. STANDARDISATION OF SHIP OSCILLATION AND METHODS FOR ITS REDUCTION

The speed of a ship v has a significant impact on parameters of oscillating a ship in waves against water and the course angle φ_K between the direction of the ship motion and direction of waves oncoming. In this case the speed of a ship against a wave is:

$$u = c_v - v.\cos\varphi_K \,. \tag{280}$$

Angles $0° = \varphi_K \leq 90°$ and $270° \leq \varphi_K \leq 360°$ indicate the motion of the ship in the direction of waves (a wave catches up with the ship), and at angles $90° \leq \varphi_K \leq 270°$ the ship floats against waves.

Then the relative wave period, i.e. a time window between any point of the ship and two subsequent crests or troughs, will be as follows:

$$\tau_v' = \frac{\lambda_v}{u} = \frac{\lambda_v}{c_v - v.\cos\varphi_K} \,. \tag{281}$$

If we compare τ_v' to the period of inherent ship oscillation τ_θ or τ_ψ, we will gain improper values of the multiple $v.\cos\varphi_K$, when the ship occurs in resonance with a side or longitudinal oscillation at motion under an oblique course to the wave

$$\left|v.\cos\varphi_K\right| = c_v - \frac{\lambda_v}{\tau_{\theta;\psi}} = 1{,}25\sqrt{\lambda_v} - \frac{\lambda_v}{\tau_{\theta;\psi}} \,. \tag{282}$$

Changing the speed of motion *v* or the course angle φ_K it is possible to avoid unsuitable zones of ship oscillation or vice-versa to fall into resonance. There exist various special diagrams of oscillation, e.g. diagram by V. G. Vlasov, J. V. Remez, V. V. Nevolin, N. F. Solarev and other authors, in order to quickly determine appropriate values *v* and φ_K at a certain wave length λ_v.

In any case the amplitude and the period of ship oscillation in waves must be of such a value so passengers and crew feel comfortable.

Thanks to statistics it is known that symptoms of seasickness are rapidly intensified if linear accelerations felt by a person are greater than $0{,}1g$ (g - acceleration of a free fall). For skilled people this norm may be increased up to *2.6 g*.

After a double derivative of the equation of a forced oscillation $\theta = \theta_m . \cos \sigma_v . t$ it is possible to find maximum values of angular accelerations of the resonance oscillation at $\sigma_v = \varpi_\theta$ and $\theta_v = \theta_m$.

$$\left|\theta''\right|_{\cos\theta_v t=1} = \theta_m . \omega_\theta^2 = \theta_m \left(\frac{2\pi}{\tau_\theta}\right)^2 \tag{283}$$

The maximum tangent acceleration in the point *A* (see Fig. 47 b)), which is located in the distance l_m from the axis of ship heeling, will be equal to $l_m . \theta''$ and at the same time it cannot be greater than $(0.1 \div 0.26)g$,

because

$$(0{,}1 \div 0{,}26).g = l_m . \theta_{m(dov)} \left(\frac{2\pi}{\tau_\theta}\right)^2 . \tag{284}$$

Then the condition for a suitable environment (from the point of view of feelings of people on the deck) is stated in the following form:

$$\theta_m \leq (0{,}1 \div 0{,}26).g \frac{1}{l_m} \left(\frac{\tau_\theta}{2\pi}\right)^2 . \tag{285}$$

The value θ_m - the maximum amplitude of resonance oscillation is determined with appropriate methods taking the speed v and course angle φ_K with the corresponding period τ_θ into account.

To decrease the amplitude of rolling various means are used in order to eliminate the ship oscillation. These means are categorised into passive and active ones by the principle of effect. Passive means used to eliminate the ship oscillation include bilge keels welded to a bilge plating so they do not stand out contour lines of the ship. Moreover there are various wings, etc. Besides the fact they cannot stand out contour lines of the ship, bilge keels must be in the direction of flow so they do not evoke a higher supplemental resistance.

Despite their simple construction bilge keels reduce the amplitude of oscillation, mostly in a resonance zone down to 30 ÷ 40 %. That is why they represent the most common means to eliminate the ship oscillation.

In some sea ships, mostly the passenger ones, there are placed complex facilities of active means to reduce the oscillation. These are stabilising side fins working on a principle of automatically controlled supporting wings, then e.g. active tanks, various gyroscopic devices, etc.

6. MANOEUVRABILITY OF A SHIP

Manoeuvrability is the ability of a ship to preserve the given direction of motion or to change it in accordance with changes of the steering gear, i.e. to move in the sea way chosen by the captain of the ship. Manoeuvrability is characterised with two opposing properties - course stability and turning quality. The contrariety lies in the fact that the more stable in course the ship is, the less manageable it is, and vice versa.

A ship must satisfy appropriately chosen complex indicators which match the hull as well as the direct determination of the ship. A ship unstable in the course misses 3 ÷ 5 % of its speed and requires constant manoeuvrability with a rudder in order to keep the course. Ships which are not manageable enough can hardly be controlled, e.g. while navigating through a sharp curvature of a river, overtaking and meeting larger ships and arrangements, getting ashore to port positions, etc. In general, river ships are characterised with a higher turning quality when compared to sea ones.

The ship is controlled with a steering gear. When the rudder is yawed from its initial position there arises a hydrodynamic force on it which can be divided into two directions $x.G_y$ in the coordinate system relative to the ship, i.e. into two components: Y_k - perpendicular to axis plane, and X_k - parallel to axis plane, which is usually a brake force against the advance motion, or a so called resistance force.

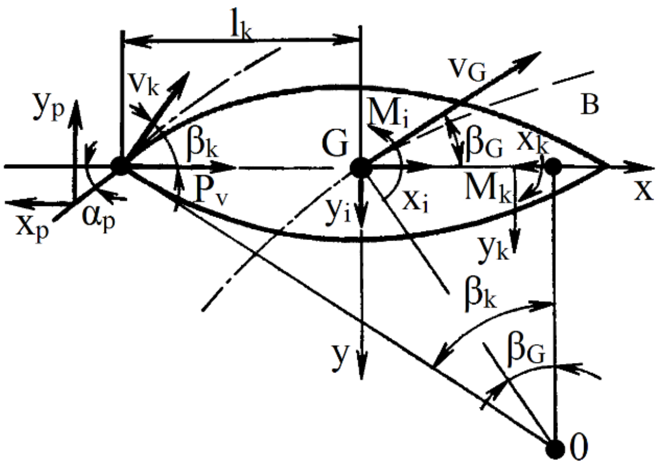

Fig. 48. Forces and Moments of Forces Acting on a Ship in a Horizontal Plane During the Evolution Phase of Circulation [Authors, 3]

In the initial moment the force: Y_k evokes circulation of the ship around its vertical axis as well as side shifting of the ship (drift) in the direction of its motion and deviation in the opposite direction (outwards).

As a result of the circulation around the vertical axis and the parallel forward motion the centre of gravity of the ship starts to move along a curvilinear trajectory with the centre of curvature 0 and radius of curvature (radius of circulation): R_g. This leads to an asymmetric flow around the hull:

- the velocity vector in the centre of gravity G is: v_G and together with the ship axis it is defined with the angle of drift β_g.

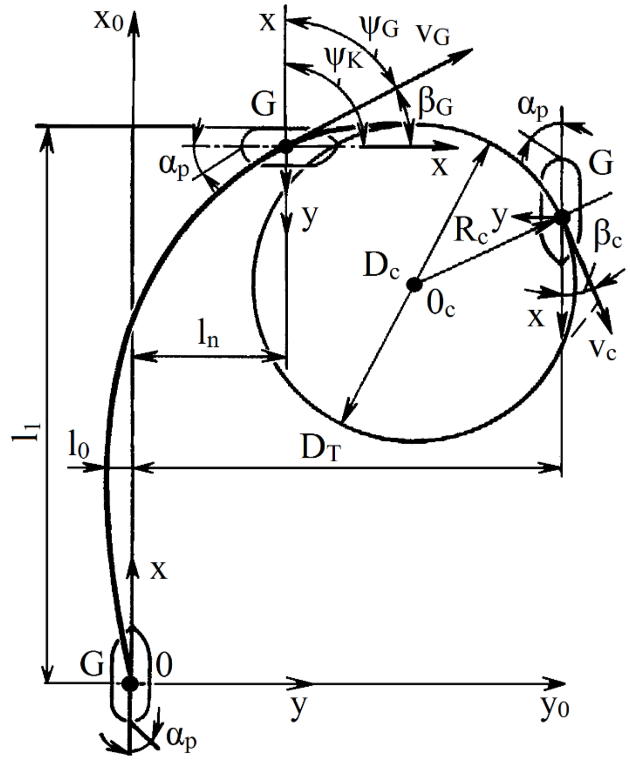

Fig. 49. Elements of Ship Circulation [Authors, 3]

From the hydrodynamic aspect, during such a motion the ship's hull can be considered a wing of breadth L and length T, i.e. a wing of a small slenderness moving along a curvilinear trajectory. There on the hull arises a hydrodynamic force of the hull Q_t which acts from the hull front at an angle towards the axis plane. This force may be split into a transverse force Y_k and resistance force X_t.

The curvilinear trajectory, on which the ship moves after yawing its rudder blade, is called a circulation.

If the yawing angle of the rudder does not change, the ship achieves a steady motion round the circle step by step, whereas the forebody of the ship will be headed inwards of the circulation and the stern outwards.

Parameters of a curvilinear motion of a ship in circulation:

R_G - the radius of circulation of the ship's centre of gravity [m] ,

v_G - the velocity of the ship's centre of gravity (along the tangent to circulation) [m.s^{-1}],

β_G - the angle of drift between the vector of advance velocity of the centre of gravity v_g and the axis plane identical with a moving axis G_x,

x_0, y_0 - coordinates of the ship's centre of gravity in a fixed coordinate system with its origin being identical with the ship's centre of gravity in the moment of yawing the rudder; the axis $0x_0$ is identical with the ship course,

φ_k - the angle of course between the straight course $0x_0$ and the axis plane in any instant of time t,

ψ_c - the angle of velocity between the straight course $0x_0$ and the vector of advance speed,

$\omega = \frac{d\varphi_K}{dt}$ - the angular velocity of ship circulation with regard to a vertical axis G_z [s^{-1}].

Last parameters $(\psi_c, \varphi_k, \omega)$ depend on R_G, v_G, β_G, and on time t.

Angles ψ_c, φ_k, β_G are mutually dependent on each other:

$$\varphi_k = \psi_c + \beta_G \quad (286)$$

Their changes depending on time are as follows:

$$\omega = \frac{d\varphi_k}{dt} = \frac{d\psi_c}{dt} + \frac{d\beta_G}{dt} \quad (287)$$

If within the time dt the angle of velocity achieves an increment $d\psi_c$, then the ship's centre of gravity passes through the segment dS along the trajectory curvature, and thus the following is true:

$$dS = R_{G.} d\psi_c \quad (288)$$

Then:

$$v_G = \frac{dS}{dt} = \frac{R_G . d\psi_C}{dt}. \tag{289}$$

Based on the equation for the velocity of the centre of gravity and the equation for the angular velocity we will get the following:

$$\omega = \frac{v_G}{R_G} + \frac{d\beta_G}{dt} \quad . \tag{290}$$

The second addend is a small value when compared to the first one and it equals zero during a steady motion.

Circulation is divided into 3 phases:

- a manoeuvre phase,
- an evolution phase,
- a steady circulation phase.

The manoeuvre phase of circulation takes short, approx. 15 - 30 seconds. It is the period from the beginning of the rudder yawing until its termination. During this period the direction of the ship and the speed of its motion practically do not change. Only acceleration and forces are changed intensively.

The evolution phase of circulation takes from the end of the manoeuvre phase until the beginning of a steady motion of the ship along the circular trajectory. This phase is characterised with constant changes of values and direction of forces acting on the ship body and rudders as well as changes of parameters of trajectory of the ship motion. During this phase the ship gains a veer in the opposite direction to rudder yawing (drift of the ship), the direction of radius of circulation curvature changes from outside towards inside; the angle of drift β_G and the angular velocity of circulation ω increase, the velocity v_G, i.e. advance velocity of the ship's centre of gravity, is reduced. During the evolution phase of circulation the ship usually yaws from the initial straight course to a course angle $\varphi_K = (90° \div 100°)$.

The phase of a steady circulation is characterised with constant values of all forces acting on the ship. At the same time the ship's centre of gravity moves with a constant motion velocity v_c along the circle of radius R_c with a constant angle of drift β_c, and thus with a constant angular velocity ω_c which equals:

$$\omega_c = \frac{v_c}{R_c}. \tag{291}$$

The angular velocity is often characterised with a dimensionless value (e.g. in normatives of manoeuvrability) and it is a parameter of ship's turning quality.

$$\overline{\omega_c} = \frac{\omega_c . L}{v_c} = \frac{L}{R_c} \tag{292}$$

The shape of circulation (trajectory of the centre of gravity) is a curve analogical to that one shown above. The main characteristics or measure of turning quality is often represented not with $\overline{\omega_c}$, but with a ratio radius $\overline{R_c}$ or diameter $\overline{D_c}$ - diameter of steady circulation:

$$\overline{R_c} = \frac{\overline{D_c}}{2} = \frac{R_c}{L}. \tag{293}$$

The magnitude $\overline{\omega_c}$ and $\overline{R_c}$ is a function of the type and characteristics of the rudder and propulsion complex of the ship and it is also a function of the form of the ship's hull, i.e. in relation to main dimensions and coefficient of fullness, but also a function of the form of a lateral and forming the stern itself. These values are determined from motion equations solved for a circular motion of the ship.

Predominantly for ships with a very good manoeuvrability $\overline{R_c} = 0.6 \div 1$; for pusher arrangements $\frac{R_c}{L_{zob}} = 0.8 \div 1.7$; for sea ships the following is usually achieved: $\overline{R_c}>2$.

Other parameters of ship's turning quality may also be determined from the equations for motion or they may be expressed through a ratio radius of circulation $\overline{R_c}$ using approximate formulae or diagrams which are equally valid for ships and arrangements of ships of all types.

For example to determine the velocity of the ship's centre of gravity in a steady circulation v_c there exists a whole series of formulae which bring satisfactory accord with the experiment.

One example of recommendations may be represented with the formula by O. I. Gordejev and V. G. Pavlenko:

$$\overline{v_c} = \frac{v_c}{v} = \frac{1}{\left(\sqrt[3]{1 + 2{,}7\left(\frac{L}{R_c}\right)^2}\right)^2}, \qquad (294)$$

where:

v - the speed of a ship of a straight direction until the rudder yawing.

If the velocity v_c is known, then it will be easy to determine the angular velocity $\omega_c = \frac{v_c}{R_c}$, too.

We are also familiar with empirical relations which qualify $\overline{R_c}$ with the angle of drift β_c. For example, prof. L. M. Ryžov recommends to determine the dependency of the angle of drift of the ship's stern from the following formula:

$$\beta_c = \frac{2{,}2 + \overline{R_c}}{1{,}4 + 1{,}6.\overline{R_c} + 0{,}9.\overline{R_c}^2}. \qquad (295)$$

As it is clear from the information above the angle of drift changes from this maximum value on the stern β_k down to zero in the point C which is called the centre of rotation or the pole of rotation. A relative distance between the centre of gravity and the pole of rotation oscillates at the value of 0.45 which gives grounds for a generally known approximate formula for determination of β_c by Munk:

$$\frac{GC}{L} = \frac{R_c}{L}.\sin\beta_c = \frac{R_c}{L}.\beta_c \approx 0{,}45, \qquad (296)$$

or

$$\beta_c = \frac{0{,}45.L}{R_c} \cong \frac{0{,}45}{\overline{R_c}} \quad . \tag{297}$$

Moreover characteristics of the trajectory of the ship's motion at circulation are as follows:

- a tactical diameter of circulation D_T $[m]$ which is determined as a distance between the position of the axis plane of the ship with a straight course until rudder yawing, and the position after changing the course to heading 180º,
- a fore shift of a ship (after altering the course to 90º) l_1 $[m]$; a distance which the ship's centre of gravity passes during the navigation on a straight course since the beginning of the ship's yawing until its inclination to 90º. In order to decrease the magnitude of the fore shift l_1 before the rudder yawing the speed of motion is decreased and afterwards it is increased depending on the rate of the ship's yawing from the straight course,
- a side shift of a ship l_2 $[m]$ - the shortest distance from the line of a straight course to the ship's centre of gravity after its rotation to $\varphi_K = 90°$,
- a veer l_0 $[m]$ - a maximum transverse shift of the centre of gravity from the straight course to the opposite side when compared to the rudder yawing (as a result of drift),
- the magnitude of these characteristics is usually expressed as a ratio R_c and it ranges in a relatively small interval for all types of ships:

$$D_T = (1{,}8 \div 2{,}4)R_c \, , \tag{298}$$

$$l_2 = (1 \div 1{,}2)R_c \, , \tag{299}$$

$$l_1 = (1{,}2 \div 2{,}2)R_c \, , \tag{300}$$

$$l_0 = (0 \div 0{,}2)R_c \, . \tag{301}$$

In case of river ships and arrangements it is more convenient to determine values l_1, l_2, and l_0 not for the centre of gravity, but for the ship's aft perpendicular, because the stern curves the circulation in a greater radius than the centre of gravity, and thus aft parameters are crucial from the point of view of safety of performed manoeuvres.

The deck crew is required to know time characteristics of circulation:

- the period of circulation - time of the ship's turn to 360º:

$$\tau_c = \frac{2.\pi.R_c}{v_c}, \tag{302}$$

- the period of fore shift - time of the ship's inclination to 90º from the straight course:

$$\tau_1 = \frac{\left(\frac{\pi}{2} + \beta_k\right)}{\omega_c}. \tag{303}$$

Let us mention main characteristics of course stability.

Course stability is labelled as automatic if without any additional handling with a steering gear the ship moves again along a straight trajectory after small random excitation forces stop their acting. The experience and experiment confirm that the majority of ships do not have an automatic course stability and they perform a circulation of radius R_{c0} with a certain minimum angle of drift β_0 which is called a critical angle of drift spontaneously with non-yawed rudders ($\alpha_k = 0$). To keep such ships on a straight course a constant manoeuvring with steering gears is required. The minimum angle α_{k0} the rudder must be yawed to so the ship conducts out of a spontaneous circulation with the angle of drift β_0 is called a critical angle of rudder yawing.

The ability of a ship to move in a given direction (course) using the steering gear is labelled as an operational stability.

Criteria of the operational stability are represented with a relative radius of critical circulation:

$$\overline{R_{c0}} = \frac{R_{c0}}{L}, \tag{304}$$

when $\alpha_k = 0$ or with the number and angle of rudder yawing needed for keeping the ship on the straight course. For current ships the operational stability of a course is considered suitable if there in deep water it is true that $\overline{R_{c0} = 5}$; or in order to keep the ship on the straight course in case of wind 1 - 3 [$m.s^{-1}$] it is necessary to yaw the rudder not more than $\frac{66.v}{L}$ -times (4-6-times) per minute to an angle not greater than 3° - 5°.

In case of a manual steering these values are impacted with a subjective quality and qualification of a helmsman. Helmsmen must sense the ship's deviation from the course in advance and they are required to eliminate the ship's drift through the rudder yawing.

The best results are gained when ships are controlled using autopilots (automatic helmsmen) which can react to ship's yawing faster than a human being and do not allow the yawing from the given course to more than 0.5° while yawing the rudder to 1° - 2° only.

In comparison to a manual control the utilisation of an autopilot eliminates the fuel consumption in 10 % as a result of reducing wavy motion of the ship, and it ensures ship navigation during reduced visibility; moreover the intensity of nautical crew's work is reduced and their work is simplified.

The essential scheme of the autopilot includes active sensors, which are specially fine-tuned to fit the alteration of the course angle and angular velocity, as well as amplifiers and multipliers of sensors impulses, which suitably control the operation of a steering gear.

6.1. CALCULATION AND STANDARDISATION OF MANOEUVRABILITY OF A SHIP AND PUSHER ARRANGEMENTS

The calculation of manoeuvrability introduces qualitative indicators of manoeuvrability and elements of a steering gear, ensuring safe control of the ship in a certain aquatory with certain gabarits of a water route (in the plane *x0y*) and in a given depth at a certain speed, direction of flow, and regime of wind and waves. With this intention it is necessary to solve the system of equations of the ship's motion; the ship behaves like a hard body with some co-acting weights and moments of these additional hydrodynamic forces which are caused with water viscosity. Motion equations are usually formed in the coordinate system which is movable and whose origin is set into the centre of gravity of the ship (or the arrangement).

Qualitative characteristics of steerability are also determined on the basis of model tests which are performed with self-propelled models, and subsequent recalculation of results for the actual size of the ship in compliance with laws on hydrodynamic similarity. Specified actual characteristics can be determined only on the basis of a natural test of the ship.

To form movable equations it is necessary to know forces and moments acting on the ship and its rudder organs. In case of more accurate calculations mostly for bigger pusher arrangements it is required to take the following into account: the change of force and moment of the propulsion device which to some extent influences the advance speed, angular velocity and subsequently the angle of drift of a convoy, too. The impact of propulsion device's parameters mentioned above is examined in detail in the area of "operating and manoeuvring properties of ships"; therefore we will get familiar only with motion equations studying forces and moments in the plane $x0y$, as it is customary in the subject Manoeuvrability of self-propelled sea and river ships. So we will study the impact of the following forces:

a) Forces of inertia of the ship's weight and co-acting weights of water whose projections on the x, y -axis will be labelled as x_i, y_i and the moment of inertia forces with regard to the z -axis will be labelled as M_i. Within the calculation of manoeuvrability the co-acting weights cannot be considered "weights attached" to the ship's hull and moving along with it without any changes. Co-acting weights are additions to the weight of the ship which take the inertness of liquid into account, and thus enable to investigate the motion of the ship like in a real life. Co-acting weights are functions of the ship's form and relations of their main dimensions like $L/_B$, but mostly $T/_B$ and they are determined with methods used in theoretical mechanics and experimental hydrodynamics.

b) Hydrodynamic forces of the hull x_t, y_t and the moment M_t acting on the immersed part of the ship are divided into two categories: advance elements, which arise at the hull's motion like wings of a small fineness with the advance speed v_G at the angle of attack β_G towards the direction of flow; and damping (rotating) elements of these forces and their moment, which arise at the hull's rotation around the vertical z - axis with the angular velocity ω. These forces are caused with the water viscosity. Values of forces are functions of the form of the immersed part of the ship, depth of a sea-way, dimensions and form of the ship's lateral, and they are functions of the following parameters: v_G, ω, β_G, $L/_T$, $B/_T$, $T/_{h_{vc}}$. These forces are usually determined on the basis of model tests' results. Advance forces or coefficients of these forces are determined in straight model troughs or aerodynamic tunnels with advance speeds of the model corresponding to v_G and at various angles of attack β_G. Damping - rotational forces and moments are determined in special rotary devices (circular troughs), where the model performs a steady circulation.

c) Hydrodynamic forces of rudders (rudder system) $\sum X_k$ and $\sum Y_k$, which are created at rudders or other steering gears yawing, and the moment of transverse force $\sum Y_k$ with regard to the ship.

$$\sum M_k = \sum Y_k . l_k \tag{305}$$

d) The thrust of the propulsion device x_p, P_P, which is determined with methods used in calculation of propellers, explained in more details in the section of operating and manoeuvring properties of ships.

e) Wind pressure forces on part of the ship above water and their moments X_v, Y_v, M_v depend on the speed and direction of wind and on the form of the ship's part above water. Values of these forces are determined from model tests of the ship's part above water in an aerodynamic tunnel at various angles of attack towards the flow of air.

If we accept (with a sufficient precision, e.g. without taking the turning moment of the propulsion device into account) that the ship's motion at circulation is only performed in the horizontal plane (a flat motion), then the following equations of motion which - in compliance with D'Alembert's principle in individual projections on the x, y - axis and moments in the direction of the z - axis - can be approximately expressed (the approximation is given as $\cos\beta_G = 1$; $\sin\beta_G = \beta_G$) using the following equations:

$$v_x = v_G.\cos\beta_c\,,$$
$$v_y = v_G.\cos\beta_c\,. \qquad (306)$$

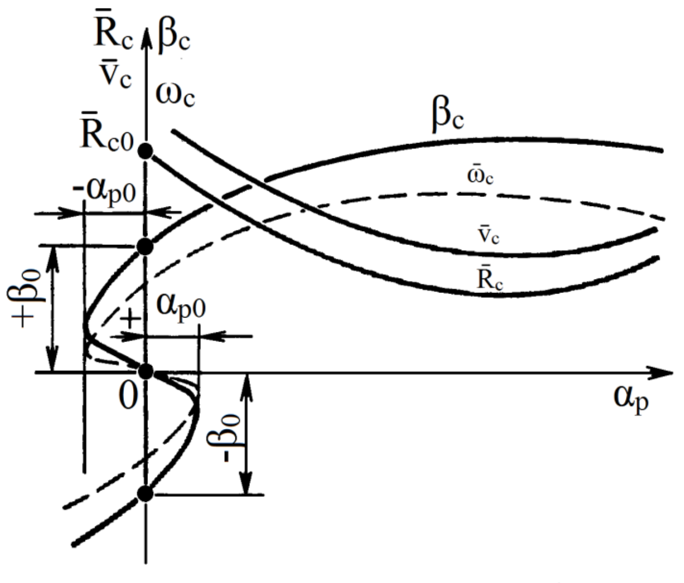

Fig. 44. Diagram of Steerability of a Ship without an Automatic Course Stability

[Authors, 3]

In equations there are k_1 and k_2 - coefficients of a co-acting weight during the ship's motion in the direction of the G_x, G_x-axis, and k_3 - a coefficient of a co-acting moment of inertia during the ship's rotation with regard to the G_z-axis.

In order to reintegrate these equations and to find parameters β_G, ω, $R_G, x_0(t), y_0(t)$ and v_G of a curvilinear motion of the ship in dependence on the initial speed v and the angle of rudder yawing α_k, firstly it is required to express forces elements $x_t, y_t, \sum X_k, \sum Y_k, X_v, Y_v$ as well as the useful thrust of propulsion x_p, P_P via parameters v_G, β_G and ω.

In case of a steady circulation the equations are transformed into a system of algebraic equations, and the solution is not difficult.

For a non-steady circulation the integration of the system of non-linear differential equations is difficult and it can be solved using automatic computers or various simplifications.

Based on the calculation results of $x_0(t), y_0(t)$ the trajectory of circulation as well as the diagram of steerability of a ship (Fig. 52)

or the diagram $\beta_c = f(\alpha_{k)}, \overline{v_c} = \frac{v_c}{v} = f(\alpha_k), \overline{\omega_c} = \frac{L}{R_c} = f(\alpha_{k)}, R_c = \frac{R_c}{L} = f(\alpha_{k)}$ for a period of a steady circulation is constructed, according to which values β_0, α_{k0}, $\overline{R_{c0}}$ for a ship unstable on its course are determined.

Due to complexity and often also due to complications with accurate determination of forces, which are entered into motion equations, the approximate methods for determining manoeuvrability characteristics are of a practical meaning.

For a steady circulation the inertia force Y_i is proportional to the ship's weight $\rho.V$ and normal acceleration $\frac{v_c^2}{R_c}$, i.e.

$$Y_i = k.\rho.V.\frac{v_c^2}{R_c}\,. \tag{307}$$

The hydrodynamic force of the hull in accordance with the theory of wing is proportional to the surface area of the lateral S_{OR}, i.e.:

$$Y_t = C_{yt}.\rho.S_{0R}.v_c^2 \frac{1}{2}, \tag{308}$$

and the lift force of rudders (rudder force):

$$\sum Y_k = \sum C_{yi}.\rho.F_i.v_{k_i}^2 \frac{1}{2}. \tag{309}$$

If we substitute these values into the second motion equation (the equation in the direction of 0_y), then we will get the following approximate formula for determining the characteristics of turning quality of a ship but also of a pusher arrangement - of a ratio radius of a steady circulation, based on which we can then determine other relations of circulation, too:

$$\overline{R_c} = K_c \left(\frac{1}{1{,}2 + \frac{0{,}21}{\alpha_k}} \right) . \frac{2.V}{L.\sum \overline{Y_k}}, \tag{310}$$

where:

$$K_c = f\left(\frac{V}{L.S_{0R}} \right) = f(\sigma_v). \tag{311}$$

The coefficient which was experimentally found during the statistical handling of tests of ships and arrangements K_c is, however, a function of the type of a steering gear (K_c - for rudders behind propellers without nozzles or for rudders behind propellers in stable nozzles; $K_c^{'}$- for rotating nozzles with stabilisers) and it is dependent on a dimensionless value $\sigma_v = \frac{V}{L.S_{CL}}$, where $S_{CL} = L.T\left(1 - \frac{S_z}{L.T}\right) \approx 0.93L.T$ and S_z - supplemental area of fullness of the lateral (run), which supplements the surface area S_{CL} into a rectangle $L.T$.

α_k	- the angle of a steering gear yawing from the axis plane [rad],
$1.2 + \frac{0.21}{\alpha_k}$	- the coefficient gained from the results of natural tests,
V	- the volume displacement of a ship or an arrangement [m^3],

$\sum \overline{Y_k}$ - the characteristics of a rudder force which is determined for individual types of a steering gear according to approximate formulae by V. A. Lesjukov.

Table 9

Coefficient Kc For Rudders and Nozzles

$\sigma_v = \dfrac{V}{L.S_{0R}}$	K_c (rudders)	K'_c (nozzles)
0.035	-	2.6
0.040	3.80	2.00
0.045	2.80	1.70
0.050	2.25	1.47
0.055	1.88	1.30
0.060	1.60	1.18
0.070	1.25	0.96
0.080	1.02	0.82
0.090	0.80	0.70
0.100	0.75	0.60
0.110	0.65	0.53
0.120	0.59	0.47
0.130	0.55	0.42
0.140	0.50	0.39
0.150	0.48	0.37

Source: [Authors, 3]

The Characteristics of a Rudder Force:

1. For ships equipped with rudders behind propellers without nozzles the characteristics of a rudder force is as follows:

$$\sum \overline{Y_k} = \sum_{i=1}^{x_k} C_{yi}.F_i\left(1+\sigma_p \frac{F_{pi}}{F_i}\right), \tag{312}$$

where:

x_k - the number of rudders, usually identical with the number of propellers x_p,
$C_{yi} = f(\lambda_{ki};\ \alpha_{ki})$ - the coefficient of a lift force of the ith rudder determined with diagrams for a rudder calculation,
α_{ki} - the angle of a rudder yawing [rad],
λ_{ki} - the fineness of a rudder (the ratio of the height to the breadth of the rudder),
F_i - the area of a rudder [m^2],
F_{pi}=$D_p.b_{ki}$ - part of the area of the ith rudder flown around with any current (D_p- the diameter of the propeller [m]) F_{pi}=0 for rudders not placed within the propeller wash,
σ_p - the coefficient of a propeller load thrust without a nozzle.

The coefficient of a propeller load σ_p can be determined with the following approximate formula:

$$\sigma_p = \frac{8.P_v}{\rho.\pi.D_v^2.v_p^2} = \frac{8.N_p}{\rho.\pi.D_v^2.v_p^2\left(1{,}03v_p + 0{,}475\sqrt[4]{\frac{n_v^2.N_p}{\rho.v_p}}\right)}, \qquad (313)$$

where:

P_v - the thrust of a propeller [kN],
n_v - the propeller speed [s^{-1}],
$\rho = 1\frac{t}{m^3}$ - the river water density,
$v_p = v.(1-\psi_p)$ - the speed of the current oncoming on the propeller slowed down with the hull [m/s],
$N_p = N_e.\eta_{prev} \approx 0.96N_e$ - the power brought to the propeller [kW].

The speed v_p is determined with considerations to the coefficient of a current slowdown, which arises as a result of the ship's hull drifting certain amount of water (frictional wake) with the average speed equal to $v_{sp} = \psi_p.v$ (v - the speed of the ship or the arrangement with regard to water on a straight course). The mean value of the coefficient of the current slowdown $\psi_p = 0.18$ - for ships without a tunnel stern if $\frac{D_U}{T} \leq 0.8$; $\psi_p = (0.28 \div 0.36)$ - for ships with a half-tunnel or tunnel stern when $\frac{D_U}{T} = (0.9 \div 1.2)$. A more accurate value ψ_p is determined either from model tests or empirical formulae, which come out of systematic model experiments.

2. For ships with rudders behind fixed nozzles the characteristics of a rudder force equals:

$$\sum \overline{Y_k} = \sum_{i=1}^{x_k} C_{yi}.F_i\left(1 + \frac{1}{2}.\frac{F_{pi}}{F_i}\left(\sqrt{1 + 1{,}8\sigma'_p} + 0{,}9\sigma'_p - 1\right)\right). \qquad (314)$$

The load coefficient of propulsion propeller - nozzle complex $\sigma_p^{'}$ can be determined with the following approximate formula:

$$\sigma'_p = \frac{8.P'_k}{\rho.\pi.D_v^2.v_p^2} = \frac{8.N_p}{\rho.\pi.D_v^2.v_p^2\left(1{,}08v_p + 0{,}38\sqrt[4]{\frac{n_v^2.N_p}{\rho.v_p}}\right)}, \qquad (315)$$

where:

$v_p = v.(1 - \psi_p^{'})$ - the speed of the current oncoming on the propeller - nozzle complex [$m.s^{-1}$].

The coefficient of the current slowdown $\psi_p^{'}$ can be adopted for the following categories of ships: engine cargo ships $\psi_p^{'} = 0.30$; tow and pusher tugs $\psi_p^{'} = 0.22$. A more accurate determination of coefficients is also achieved either from model tests or empirical formulae.

3. For ships equipped with a propeller in steering nozzles with stabilisers we may determine the characteristics of a rudder force with the following approximate formula:

$$\sum \overline{Y_k} = \sum_{i=1}^{x_k} \frac{\pi . D_v^2}{4} . \left(C_{y_{di}} + C''_{y_{si}} + C'''_{y_{si}} \right) . \tag{316}$$

To determine coefficients C_y we may use the following approximate formulae:

$$C_{y_d} = \left(\frac{1{,}05 - \alpha_d}{0{,}08 + \dfrac{0{,}61}{\sigma'_p}} + 2{,}20 + 1{,}40 \overline{l_d} \right) . \alpha_d , \tag{317}$$

$$C'_{y_S} = \frac{2\pi\lambda_c}{\lambda_c + 2} \left(0{,}88 - \frac{0{,}88 . \overline{l_d}}{1{,}10 + 0{,}03\sigma'_p} \right) . \left(0{,}5 + 0{,}45\sigma'_p + \frac{1}{2}\sqrt{1 + 1{,}8\sigma'_p} \right) . \frac{4 . F'_S}{\pi . D_v^2} \alpha_d , \tag{318}$$

$$C''_{y_S} = \frac{2\pi\lambda_c}{\lambda_c + 2} . \left(\frac{4 . F''_S}{\pi . D_v^2} \right) . \alpha_d , \tag{319}$$

where:

C_{y_d} - the coefficient of a lift force of a steering nozzle,

C'_{y_d} - the coefficient of a lift force of the stabiliser's part which is flown around with a propeller wash F'_S,

C''_{y_d} - the coefficient of a lift force of the stabiliser's part which is not flown around with a propeller wash F''_S.

The meaning of individual parameters is as follows:

$\alpha_{d_{ld}}$ - an angle of a nozzle yawing [rad],

$\overline{l_d} = \dfrac{l_d}{D_v}$ - a relative length of a nozzle (a ratio of the length to the diameter),

$\lambda_c = \dfrac{(h'_c + h''_c)^2}{(F'_c + F''_c)}$ - a relative fineness of the stabiliser,

$F'_c = l'_c.h'_c \approx 1{,}1.l'_c.D_v$ - the area of a stabiliser located in a propeller wash [m^2],

$F''_c = l''_c.h''_c$ - the area of a stabiliser out of a propeller wash [m^2].

The formula for approximation $\overline{R_c}$ is suitable for determination of parameters of circulation with a rear steering gear as well as for an advance determination of elements and types of a steering gear (Y_k) providing the required turning quality.

We need to say that so far we have been looking at a steerability of ships or arrangements only in case of a forward navigation. However, the steerability of a stern-way is important, too; almost always it is significantly worse and even it is often missing completely.

Currently all countries with a developed water transport put together standards for the evaluation of ships' and convoys' steerability in specific aquatories of their operational utilisation.

Standards of manoeuvrability determine tolerable numerical values for: turning quality $\overline{R_c}$ and course stability $\overline{R_{c0}}$, steerability in an inertia regime with propulsion aggregates turned off, steerability at a stern-way, steerability at strong side wind and in other situations depending on specific conditions of the navigation: depth of the water route h_{vc}, breadth of the water route b_{vc}, and minimum radiuses of curvature of the water route R_{vc}, , speed of the current C and maximum speed of wind.

From the point of view of the turning quality of a cargo ship the value $\overline{R_c}$ must be less than 1.0 ($\overline{R_c}< 1$) as well as less than any value $\overline{R_c}$ within the function $(\frac{h_{vc}}{T}, \frac{R_{vc}}{L}, \frac{c}{V}, \frac{B}{L})$ determined from specific conditions of the water route on site of the ship operation. The course stability is considered suitable if the critical radius of circulation with non-yawed rudders ($\alpha_k = 0$) is $\overline{R_{c0}}=\frac{R_{c0}}{L} \geq 5$.

The steerability in the inertia regime is considered sufficient if a ship moving in a circulation in deep water can be conducted out of this circulation and turned into the opposite direction only by means of a steering gear (without using any machinery) when it is yawed at the biggest tolerable angle.

When in a stern-way the ship in deep water is required to keep the straight direction only by means of a steering gear without any manoeuvers of main propulsion aggregates.

Where there is unfavourable side wind of speed v_v, which is statistically determined for each operational segment of a water route, the ship must navigate in the specified course at the rudder yawing at an angle not greater than 20°, so there exists a reserve of the angle of yawing ranging from 10° - 15° that provides steerability of the ship in case of a dynamic ship's drift.

All calculations of steerability criteria at the mentioned manoeuvers are performed on the basis of solving motion equations. Standards of individual classification societies usually present approximate methods for calculation of these criteria, too.

The turning quality may be improved with the following construction and operation methods which emerge from the analysis of ship's motion equations:

- with reduction of stern fullness of the lateral's surface S_{CL} by means of a cut-away (transom) of the stern. In given approximate formulae this measure is factored with a coefficient K_C which is decreased along with decreasing of S_{CL},

- with decrease of $L/_B$ and increase of $B/_T$ which, however, leads to worse riding qualities of a ship (greater resistance),
- with increase of the coefficient of ship's fullness C_B when the relations of main dimensions are not changed,
- with increase of characteristics of a rudder force $\overline{Y_k}$ due to the increase of the rudder surface area, number of rudders using more efficient and supplemental rudder complexes - steering nozzles, steering devices and other solutions.

The most effective methods how to increase the course stability are as follows:

- increase of fullness of the lateral's surface S_{CL} mainly in the stern, e.g. with addition of so called "trousers" for a line shafting,
- addition of non-movable stabilisers in the form of vertical blades or plates,
- increase of rudders surface area, mainly increase of their breadth,
- increase of the ship's inclination towards the stern,
- increase of the ratio $L/_B$.

The steerability of the stern-way may be increased with installation of stern-way rudders (flanking rudders) located in front of propellers. A suitable steerability is provided by steering nozzles.

Manoeuver properties of ships and pusher arrangements are significantly improved using at least 2-propeller drives when lateral propellers should be equipped with independent steering nozzles, enabling to create big rotating moments of engines working in the opposite direction.

The steerability of a ship and a pusher arrangement in shallow water is significantly changed. When the value $\frac{v}{\sqrt{g.h_{vc}}} \leq 0.5$ the relative radius of circulation $\overline{R_c}$ increases when compared to deep water. Within the speed range $0.5 < \frac{v}{\sqrt{g.h_{vc}}} \leq$

0.8 ships gain a big instability of the course (change of the angle of drift) and the relative radius of circulation decreases $\overline{R_c} = 1.5 \div 2$ -times.

Then after another speed increase the course stability of the ship increases again. That is why the pilot is required to pay special attention mainly during navigating the ship in shallow water and channels where the vessel may happen to come into a non-steerable veer which can only hardly be balanced, and it may lead to a disaster.

The steerability of fast ships, particularly ships on air cushion - air-cushion vehicles which do not have the immediate contact with water, has not been studied much. When controlling these vessels the study and evaluation of model tests, mainly natural tests, are crucial in order to get reliable operation customs.

7. STRENGTH AND CONSTRUCTION OF A SHIP

The strength of the ship's hull and its individual elements lies in the ability to keep their integrity, to deform only in tolerable bounds and to keep the stability of design elements during action of the most unfavourable strains which may arise in the operation process. It is clear that the issues of strength are ones of the most fundamental qualitative properties of a ship where the efforts are to achieve this goal with the smallest possible weight of the ship's hull.

The discipline which deals with the study of strength, stiffness and stability of ship structures is called "statics of a ship" or more accurately "structural mechanics of a ship". The methodology in calculations of the strength says that the overall strength is calculated separately from the local strength of the ship's hull.

7.1. OVERALL AND LOCAL STRENGTH OF A SHIP

In calculations of the overall strength there is determined tenacity of the hull's structure as a whole, i.e. as a box girder of a variable section, which is from one side loaded down unevenly with a spread load of dead load, weight of devices and cargo, and from the other side with forces of water displacement and in special cases, such as when tugging the ship out of water, also with various reactions. Under influence of these forces the ship's hull bends and it gains a flexure or a bending. In case of a flexure in all longitudinal girders of the bottom there will act the thrust - tensile stress, and in the deck there will act the pressure - tenacity in pressure; in case of a bending it will be vice-versa. Sides and longitudinal bulkheads represent faces of the box girder of the hull where shear stresses will mainly act.

The bending of the hull emerges from the uneven distribution of the weight of the ship and forces of displacement. If these forces were identical in each transverse section, i.e. if they were in mutual balance, then they would not evoke any bending or shear. That is why concentrated forces from heavy loads, responses from the river bed, when a ship runs aground, longitudinal variable forces during floating in waves, etc., are particularly dangerous from the point of view of a general strength.

When the ship moves slant to waves the hull is exposed not only to a bending, but also to a rolling strain. Longitudinal structure girders, sides and longitudinal bulkheads tolerate (normal - axis, and shear - tangent) stresses by overall bending of the hull, but at the same time they must also transfer a local bending by immediately acting transverse forces of water pressure or weight of load and machines. A local bending is also manifested in girders of a transverse stiffening and in transverse bulkheads which do not participate in the overall bending, but serve as shores for longitudinal girders thanks to what they can ultimately affect the overall strength of the ship in certain cases.

The study of tenacity of structural elements, such as bulkheads, girder chasses, gratings and plates of the plating under the effect of external loads immediately on these elements is the subject of a so called local strength of the hull.

The overall strength of the hull is a prerequisite for the safety of a ship. Its damage leads to serious disasters, such as a breach of the ship's hull. A local deformation of individual girders or plating plates when speaking about the local strength is not usually a risk of a direct breakdown of the ship as a compact hull. In spite of it a significant amount of the hull's repairs is caused with removals of the hull's local deformations and damages as a result of hits at loading, landing manoeuvers, under the effect of pressure when the ship runs aground, etc.

7.1.1. Methods for Determining Forces Causing the Overall Bending of the Ship's Hull

Calculations of the overall and local strength of a ship are conducted in the following order (stages) which are generally known and valid in the subject "elasticity and strength".

1. Determination of an external load, schematisation of design, calculation and formation of diagrams of bending moments and transverse forces for ships as a whole and for individual structural girders - *issue of external forces*.

2. Calculation of internal tenacities in structural elements and in the hull as a whole - *issue of internal forces.*

3. The check of strength, i.e. comparison of tenacities effect to the magnitude of tolerable tenacities - *issue of tolerable tenacities*.

First of all it is necessary to analyse the issue of external forces. The subject "elasticity and strength" informs us that the transverse force in a transverse section of a girder in the distance x from the origin of the coordinate system equals the algebraic sum of the projection of all forces (including shores reactions or displacement forces) which are taken from the right or from the left of the studied section. The bending moment is equal to the algebraic sum of moments by given forces (from the left or from the right) with regard to the centre of this section.

Based on these general assumptions bending moments and transverse forces in calm water are calculated in the following order:

a) The weight of structures and machines is calculated according to the design. Then using the formula $P_{vi} = m_i . g$ the weight of the ship and cargo is distributed into 20 theoretical compartments of the hull, i.e. a scheme of the ship's load $P_{vi} = \frac{P_{vi}}{\Delta L}$ is constructed in the form of a columnar line of load, where ΔL is the length of a theoretical frame spacing which is equal to $\frac{L}{20}$.

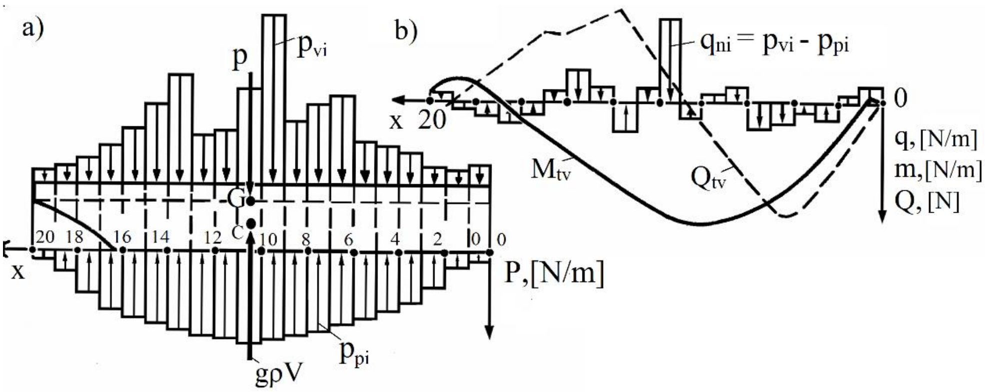

Fig. 51. Forces Causing the Overall Bending of the Ship's Hull [Authors, 4]

a) diagram of weight and displacement; b) diagram of a weight load q(x) of a transverse force Q_{tv} and a bending moment M_{tv} in calm water

The distribution of the load on the length of cargo space is taken as such as it is considered within the instruction for loading and unloading. Besides in case of cargo ships at a full loading and the calculation p_{vi} and M_{tv} it is assumed that 5 % of the total load amount from the forebody and stern part of the end cargo spaces on the length 0,25.L_{np} are transferred into the centre of cargo spaces 0,5.L_{np} or in the opposite direction whether the ship is deflected downwards or upwards at an even load (L_{np} - the total length of cargo spaces).

b) For the selected load of the ship there in the Bonjean diagram we can find a forward draught T_F and aft draught T_A, and a curve of a longitudinal displacement distribution is constructed which is in fact a diagram of forces of buoyancy q_{pi}. The coordinates of displacement forces are considered a load changing not fluently but stage by stage.

c) Coordinates p_{pi} are subtracted from the coordinates p_{vi} and then the resulting weight load of a ship $q_{\text{ni}} = p_{vi} - p_{pi}$ is determined.

d) Integrating q_{ni} along the length of the ship we will get transverse forces in sections "x " in the direction from a forebody perpendicular for a ship in calm water.

$$Q_{\text{tv}} = \int_0^x q_{ni}.dx \tag{320}$$

e) Through integration of the expression for a transverse force bending moments will be determined in the same sections for a ship in calm water.

$$M_{\text{tv}} = \int_0^x Q_{tv}.dx = \int_0^x \int_0^x q_{ni}.dx^2 \tag{321}$$

Both integrals for Q_{tv} and M_{tv} are usually calculated using approximate methods - the trapezoidal rule when the number of evenly divided theoretical frames is not less than 21.

f) In the last stage there are constructed diagrams of bending moments and transverse forces in order to determine their maximum values (see Fig. 52a).

In order to determine the most unfavourable cases when Q_{tv} and M_{tv} achieve maximum values, it is required to perform calculations for various variants of the ship load, such as: an empty ship, a fully loaded ship, a ship with various ballast loads as well as a ship in the state of disrepair, when (from the strength point of view) the least favourable compartment of the ship's hull is filled.

Almost all fully loaded ships have a flexure downwards when in calm water so the weight of the cargo in the middle part of the hull is noticeably bigger than the displacement force. On the contrary in case of an empty ship the flexure is upwards (a so called bending). In diagrams moments evoking a flexure are usually considered negative, and those evoking a bending are labelled as positive.

During the ship navigation in waves there arises a supplemental bending moment ΔM which is calculated in accordance with the principles of the given classification society.

$$\Delta M = \pm K_0 . K_1 . K_2 . C_B . B . L^2 . \square_v, \quad (322)$$

where:

K_0,K_1,K_2 - are coefficients of conditions,
$K_0 = K_0\left({}^{B}/_{L}\right)$,
$K_1 = K$ (h_v - a class of the ship),
$K_2 = K_2\left({}^{T}/_{L}\right)$,
$L$and$B$are theoretical length and breadth of the ship $[m]$,
h_v - the height of a wave $[m]$.

Then the computing moment for different ship loads is determined using the following relation:

$$M_{calc} = |M_{tv}| + \Delta M. \quad (323)$$

For ships of lower categories (from the point of view of magnitude of the computing wave) the computing moment is usually represented with the moment in calm water during an uneven loading or unloading.

In approximate calculations of strength or in initial phases of ship design the bending moment in calm water is determined using the following formula:

$$M_{tv} = K . P . L. \quad (324)$$

The coefficient K stands for the statistical value of individual ship types when taking 5 % of unevenness of the cargo distribution into account. $K = 0{,}010$ - for non-self-propelled boats, $K = 0{,}016$ - for cargo ships, $K = 0{,}030$ - for tugs, and $K = 0{,}022$ - for large passenger ships.

Approximate values of the moment at an uneven loading and unloading can be calculated using this approximate formula, if the following coefficient values are substituted into the formula:

- $K = 0{,}018 \div 0{,}013$ - for non-self-propelled boats at loading with one crane into one layer,
- $K = 0{,}012 \div 0{,}010$ - for non-self-propelled boats at loading and unloading with two cranes into one layer, when the length of the boat is L = 50 - 100m,
- $K = 0{,}020 \div 0{,}014$ - for cargo ships at concurrent loading and unloading with one or two cranes into one layer, when the length of the ship is $L \geq 80$m,
- $K = 0{,}024 \div 0{,}016$ - when the length of the ship is $L \leq 80$m.

7.1.2. Calculation of the Overall Strength and the Reasoning for the System of the Ship's Hull Stiffening

Normal stresses by the overall bending $\sigma\ [MPa]$ in transverse sections of the hull as a box girder can be determined using the formulae for a pure bending of a stiff girder:

$$\sigma = 10^{-1}\frac{M_{calc}*z}{I_{y0}}, \tag{325}$$

$$\max\sigma_{deck} = 10^{-1}\frac{M_{calc.}}{W_{deck}}, \tag{326}$$

$$\max\sigma_{bott} = 10^{-1}\frac{M_{calc.}}{W_{bott.}}, \tag{327}$$

where:

M_{calc}	- a calculating bending moment in a given section of the hull [kN.m],
I_{y_0}	- the main moment of inertia of the transverse section area of the hull with regard to a neutral axis [cm^2.m^2],
z	- a distance from the neutral axis of the transverse section to the point in which the stresses are calculated σ [m],
$W_{deck.} = \frac{I_{y0}}{z_{deck.}}$	- the moment of resistance for deck strakes [cm^2.m],
$W_{bott} = \frac{I_{y0}}{z_{bott}}$	- the moment of resistance for bottom strakes [cm^2.m],
z_{deck}, z_{bott}	- a maximum distance between outermost fibres of deck and bottom strakes and neutral axis [m].

These formulae provide a correct result only if there in the calculation of I_{y_0} all specialties of the hull are considered, i.e. the distance between the hull and stiff girders of a constant section is substantial.

The first specialty lies in the fact that girders of a longitudinal stiffening feature a frequently changing section, or are discontinuous. Therefore during I_{y_0} calculation only the area of these longitudinal girders is taken into account which are firmly tied to the hull's plating and continue in both directions (towards the stern and bow) with regard to the solved section with the length of twice the height of the ship's hull (2.H). The moment I_{y_0} is usually calculated for the most weakened ship's sections, i.e. in the middle of the ship and in locations of loading hatches.

Another more substantial specialty of the hull as a box girder lies in the fact that it is constructed from various considerably different girders, stiff ones as well as flexible ones. The group of stiff girders includes longitudinal stiffeners along with segments of plating plates with the breadth $0{,}25.a_r$ from each side of the profile (a_r- distance between the ship's frames). Flexible girders of the hull usually include central segments of plating plates between profiles of longitudinal stiffeners, or longitudinal profiles themselves, if they are not stiff enough.

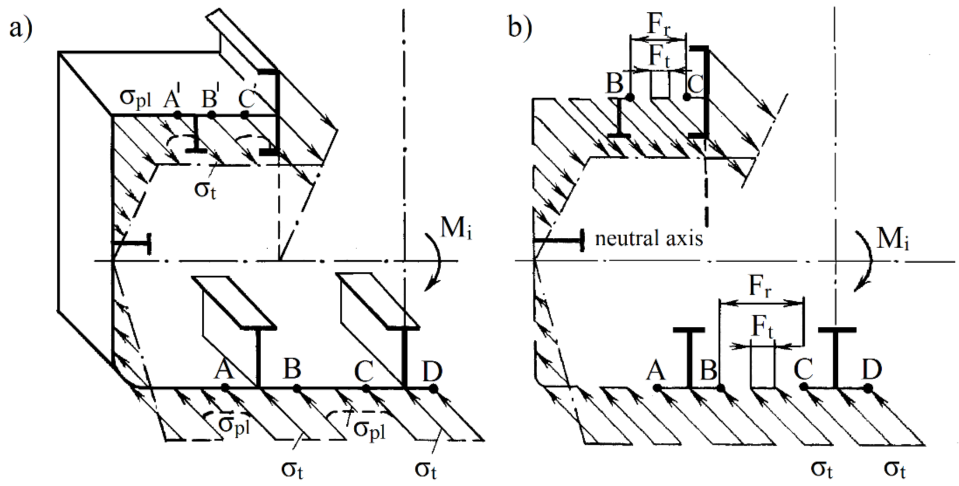

Fig. 52. Display of Stresses in a Transverse Section of the Hull during the Overall Bending Calculation [Authors, 4]

a) scheme of distribution of normal stresses by the overall bending moment in locations of stiff parts σ_t and in flexible parts of the hull σ_{pl}; b) conditional scheme of distribution of normal stresses in a hull girder

Plates are segments of the plating which are bounded with neighbouring girders of a longitudinal and transverse stiffening.

Stiff elements must not lose stability during the action of axial pressure forces caused with the overall bending as well as pressure forces by the local load with transverse forces. Stresses in stiff elements by the overall bending are almost independent on stresses caused with a local transverse load. The sum of stresses is performed algebraically through addition and subtraction of stresses by the local load to stresses by the overall bending.

Flexible elements do behave differently. Their ability to accept the axial pressure or thrust by the overall bending is substantially related to the local transverse load (e.g. by water pressure) and in particular by the initial flexure which always arises as a result of deformations at welding.

For the sake of an example we analyse the way the central flexible element of the bottom plating $\overline{BC}$ (Fig. 53) tolerates the load when the bottom is suppressed, thus there exists a bending of the ship. First of all let us assume that the studied segment of the plate does not have any initial flexure and is not water pressurised. Then pressure stresses in the bottom by the overall bending will achieve a certain critical value or, as similarly Euler's stress σ_e is called, when a flexible segment of the plate BC loses its stability, it bulges and loses its ability to resist a further compression. Further increase of pressure forces by the overall bending must be accepted only by such elements which do not lose their stability and where tensions are significantly bigger than in flexible elements.

If a plate has the initial flexure (bulge) or is under effect of a transverse load, its ability to tolerate even bigger load decreases rapidly.

Values of Euler's stresses σ_e and axial reduced stresses σ_{pl} in plating plates are substantially dependent on the system of the hull's stiffening.

The system of the hull's stiffening is called a way of spreading the hull's stiffeners and it is given with a frame spacing α (distance between adjacent transverse girders - frames) and with a distance *b* (distance between adjacent longitudinal hull's stiffeners).

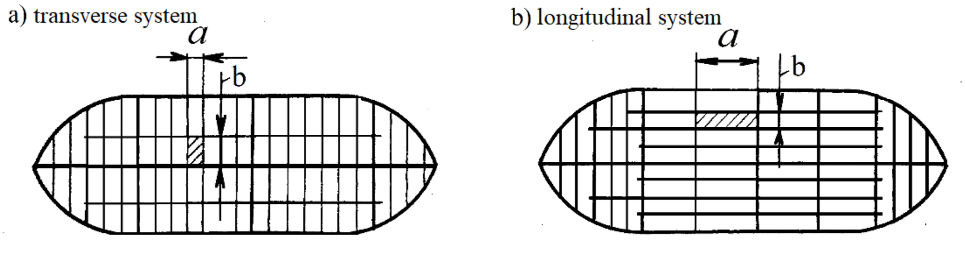

Fig. 453. Systems of Stiffening of the Ship's Hull [Authors, 4]

If the frame spacing α is less than the distance between longitudinal stiffeners b, or if these distances are equal, such a system is called a transverse system of the hull's stiffening. And vice-versa - if the distance between longitudinal stiffeners *b* is less than the frame spacing α, such a system is called a longitudinal system of stiffening.

Euler's stress [MPa] of a freely supported surface area of the plate which does not carry a transverse load at the transverse system of stiffening (α), when it starts losing its stability, will be as follows:

$$\sigma_e = 20{,}0\left(\frac{100.t}{a}\right)^2\left(1+\frac{a^2}{b^2}\right)^2, \tag{328}$$

where:

t - thickness of the plate in [cm].

For plates of a longitudinal stiffening ($a > b$) the Euler's stress is as follows:

$$\sigma_e = 80{,}0\left(\frac{100.t}{b}\right)^2, \tag{329}$$

i.e. approximately 4-times more than in case of a transverse system of stiffening.

A fully reduced stress in a plate σ_{pl} including the initial flexure and transverse load is calculated using the following formula:

$$\sigma_{pl} = \rho_p.\left(x_p - 1\right).\sigma_e, \tag{330}$$

where:

$\rho_p = f\left({}^{a}/_{b}\right)$ - the coefficient taking the degree of fixing the plate on structure girders into account,

x_p - the coefficient determined from the equation by P. F. Papkovič which depends on the initial flexure h_p, transverse load of the plate p, thickness t and ratio of sides ${}^{a}/_{b}$ as well as on the stress in adjacent stiff elements towards Euler's stresses of the plate ${}^{\sigma_t}/_{\sigma_e}$, and on the coefficient ρ_p.

If reduced stresses σ_{pl} in plates are determined using the formula $\sigma_{pl} = \rho_p\left(x_p - 1\right).\sigma_e$, then the moment of inertia of a transverse section and stress in stiff hull's elements can be found using the formula for normal stresses by a pure bending based on the following fictional method, which was firstly applied by the prof. I. G. Bubnov and is known as a reduction of flexible elements.

Let us substitute the surface area F_r of the segment $BC = (b - 0{,}5a)$ of a flexible plate, which loses its stability and in fact carries only the stress σ_{pl} (but σ_t), with the surface area F_t which is capable of transferring the same force, however, at stresses in adjacent elements:

$$\sigma_{pl}.F_r = \sigma_t.F_t, \tag{331}$$

wherefrom:

$$F_t = \frac{F_r.\sigma_{pl}}{\sigma_t} = F_r.\phi_r. \tag{332}$$

The ratio of reduced stresses of a flexible element to the stress in adjacent stiff elements of the hull, which is equal to the ratio of a fictional surface area of the plate to its full surface area, is called a reduction coefficient of a flexible element. This coefficient characterises the work capability of a plate, i.e. it determines what part of the flexible element's surface area F_t may be considered equivalent to stiff elements, and thus it is included into the surface area of the hull girder of the ship.

$$\phi_r = \frac{\sigma_{pl}}{\sigma_t} = \frac{F_t}{F_r} \tag{333}$$

A hull girder means a fictional transverse section of the ship's hull in which the surface areas of flexible elements are substituted with fictional reduced surface areas, i.e. the real transverse section of the ship's hull consisting of stiff and flexible profiles is substituted with a transverse section of a stiff girder, equivalent to that one which is able to accept the overall bending of the ship. Stresses are determined using the following relation $\sigma = 10^{-1}\frac{M_{calc.}}{I_y}$ and their calculation is often called a *calculation of a hull girder*. Reduction coefficients ϕ_r are found with the method of a sequential approximation which is part of the subject "statics of the ship (structural mechanics of a ship)". These coefficients have the following approximate values:

- for the entire breadth of pulled plates at a longitudinal system of stiffening $\phi_r = 1$,
- for segments of compressed plates of the breadth $BC = b.\left(1 - \frac{0,5.a}{b}\right)$ at a longitudinal system of stiffening $\phi_r = \frac{80,0}{\sigma_t}.\left(\frac{100.t}{b}\right)^2 \leq 1$,
- for segments of compressed plates of the breadth BC at a transverse system of stiffening $\varphi_r = 0.03$ with $t \leq 5mm$ and $\phi_r = 0,07$ with $t \geq$ 6 mm,

- for segments of spread plates at a transverse system of stiffening $\phi_r = 0{,}03 \div 0{,}53$,
- segments of the breadth $AB = CD = 0{,}5.a$ (with $0{,}25.a$from each side of a longitudinal girder for both compressed and spread plates in case of all systems of stiffening) are considered stiff and the following is true: $\phi_r = 1$ (a - is a smaller side of the adjacent plate),
- all plates of the side ϕ_r may be considered compressed regardless of the zone of the hull girder they are located in. Bilge plates in points of their round are considered stiff $\phi_r = 1$.

When the values ϕ_r of each plate and dimensional scheme of a transverse hull's section (main frame) are known, it is easy to determine values I_{y_0} and W according to principles arising from calculations for the strength and elasticity of girders with a complex section. Values I_{y_0} and $W$are calculated separately for a bending and flexure of the ship.

Based on values W for several variants of the hull's structures and $M_{calc.}$ actual stresses σ_0 in elements of the hull girder are calculated; they are then compared to tolerable stresses and an optimal hull girder is chosen.

Based on statistical methods the principles of classification societies enable to determine dimensions and parameters of the plating and longitudinal girders of the structure depending on the material and main dimensions of the hull, its class and structure type.

The analysis of ϕ_r values allows for the following statement. The longitudinal system of stiffening provides better engagement into the overall bending of flexible elements of the plating. And thus the required strength of the hull with the longitudinal system of stiffening may be secured in case of smaller thicknesses of the plating, which enables to reduce its weight in 20 % to 30 % when compared to the hull stiffened with the transverse system of stiffening, with other parameters remaining the same. In case of thin plates of the plating within the transverse system of stiffening their load capacity is considerably smaller. Moreover in case of the longitudinal system of stiffening the majority of girders is oriented in the longitudinal direction of the hull, and thus is fully included with its entire surface area of the transverse section into the hull girder. In case of the transverse system of stiffening the majority of girders is oriented transversely, and thus is not included into the hull girder.

Nevertheless, the thickness of the plating of the bottom, sides and the deck is not determined only due to securing the overall strength, but also due to securing the local strength and reliability of the hull (when local loads act while handling the cargo, e.g. while using a grab), but also from the point of view of durability against the corrosion and overall reliability. For these reasons the thickness of the plating for relatively short ships [$L \leq 40 \div 60$m] is considered greater than that one which arises from calculations of the overall strength of the hull because the overall bending moment is proportional to the ship's length. In these cases for ships of the length up to 60 m the transverse system of the hull's stiffening is chosen most frequently because it provides many other benefits.

The most important elements of the hull which provide its overall strength include: the plating of the bottom and covering of the deck, bilge segments of the plating (points of transition of sides and the bottom), upper strake of the side plating which is called a sheer strake, longitudinal girders of the bottom and the deck, longitudinal mouldings of openings of a big length.

Sides plating and longitudinal girders located near the neutral axis of the hull girder are from the point of view of normal stresses by the overall bending exposed to the strain to the least extent, however, they form the sides of the hull girder, and thus there arise tangent stresses by transverse forces in them. These stresses τ_{max} [MPa] achieve the greatest value on the neutral axis and can be calculated using the famous formula from the theory of elasticity and strength.

$$\tau \frac{Qst_{max}}{I_{y_0} . t_{max}}, \qquad (334)$$

where:

Q_{max} - the computing value of a transverse force [kN],

S_{st} - a static moment of part of the surface area of the hull girder positioned at the side of the neutral axis [cm^2.m],

I_{y_0} - the main central moment of inertia of the hull girder's surface area [cm^2.m^2],

t - a summary area of sides plating of longitudinal bulkheads at the level of the neutral axis [cm].

Sections of the hull where the greatest bending moments and the greatest normal stresses act are located near the main frame; the most exposed sections of the hull by transverse forces (from the point of view of tangent stresses) are located near edges of the ship; therefore the check of the overall strength is performed separately for normal and for tangent stresses. It is a general truth that the overall strength from the point of view of tangent stresses is secured if it is sufficient according to normal stresses.

As can be seen from fundamental formulae for the overall bending moment of the hull it is proportional to the length L, and the moment of resistance W of the hull girder is proportional to the height D. That is why the ratio of the length L to the height D of the side is the most important characteristics of the overall strength of the hull. The smaller the ratio $\frac{L}{D}$ is, the greater the overall longitudinal strength at the same thicknesses of the plating is; there is less metal used per a ton of the ship's displacement. Maximum values $\frac{L}{D}$ and $\frac{L}{B}$ are set with principles of the classification society.

For example the ratio $\frac{L}{D}$ for ships, in case of which a check calculation is required, is given with the value $\frac{L}{D} > 32$ and $\frac{L}{B} > 10$ in our midst.

7.1.3. Local and Vibrating Strength of Elements of the Hull and the Ship's Plating

In terms of a local strength the transfer of forces among individual elements of the hull and their role in providing the strength is studied.

Plating plates immediately accept the water pressure (e.g. plates of the bottom, bottom parts of sides as well as bulkheads) in case of flooding a relevant compartment or they accept the weight of the cargo (plates of the second bottom and plates of cargo decks). Some of these plates are also exposed to individual isolated forces. All these transverse loads in plates lead to bending stresses. As a result of the fact that the frame stiffener cannot change its mutual position, there in plates also emerge some tensile, so called "chain" stresses, similarly to stresses in a chain or a stiff diaphragm. The ratio of bending and chain stresses depends on the thickness of the plate and the degree of its clamping on shores (i.e. on the stiffness of plates).

The transverse stiffener of a plate is transferred to a transverse and longitudinal stiffener of the hull and to its support frame. We distinguish regular (thin, light profiles) and frame (reinforced, relatively stiff profiles) stiffeners. Regular girders are leant against frame girders which are oriented perpendicularly and are loaded with the bending by the relevant part of the plate.

The segment of the hull's plating, which is stiffened with the hull's elements and defined with a support frame individual elements are leant against, is called a grating. For example, a grating of the bottom is a set of design frame stiffeners passing along and across the hull which is usually defined with adjacent transverse bulkheads and sides of the hull (or with a side and its adjacent longitudinal bulkhead).

A grating must transfer the entire transverse load acting on the entire surface area of the given plating. Girders of the grating with a smaller spacing between them are called main girders and vice-versa; those perpendicular ones to them are called cross girders and the spacing between them is always greater. Gratings, for which girders of the main direction are parallel to the shorter side of the support frame and which are of a smaller length than the cross ones, do also have a smaller weight under otherwise same conditions.

As a result of the fact that the length of a ship's cargo space is usually greater than the ship's breadth or the breadth of the loading hatch, the weight of girders in case of the transverse system of stiffening is smaller than the one in case of the longitudinal system. Due to this reason the transverse system of stiffening is applied in case of relatively short ships for which the local strength represents a limitation for the strength of the ship as a whole.

Girders of the transverse stiffening which are placed in one transverse section of the hull and are stiffly welded in adjacent corners form a frame (a frame ring).

Since girders of the grating and frames are joined together in corners they are able to co-transfer any load, i.e. they help each other to accept the load. This is the cornerstone of the advantage of multiple statically indefinite constructions (including gratings and frames) to truss ones which comprise individual separate girders, mutually not stiffly jointed. Thus a stiff joint of girders of the transverse and longitudinal stiffening in nodes of their intersection is of an important meaning for the increase of the local strength and at the same time for the decrease of the weight of the stiffening construction.

In order to decrease the weight of the bottom and the deck grating it is important to join them mutually using vertical girders - stanchions or longitudinal and transverse bridge girders inside the ship's hull.

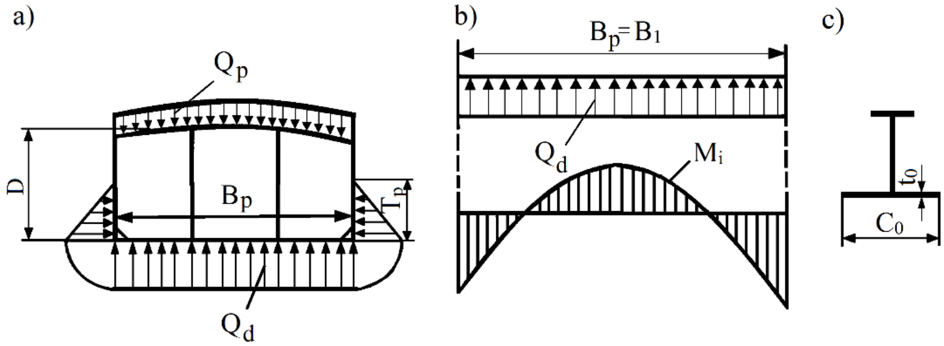

Fig. 464. For the Calculation of the Local Strength of a Frame Ring [Authors, 4]

a) diagram of a frame ring load (Q_p - a load of a beam; Q_d - a load of a bottom transverse); b) diagram of a frame transverse load without taking the responses of keelsons into account, and a diagram of bending moments on a transverse; c) transverse section of a frame transverse and a co-acting strake of the plating it is being deformed with

In this case there emerge frame structures when one grating helps the other one to transfer the load. Such structures are used in deck barges which transport the cargo only on the deck.

It is quite difficult to determine bending moments and transverse forces for individual girders of the frame and of the grating mainly due to a static indefiniteness of these structures. Solution of similar problems is the subject of the ship's static, or applied elasticity and strength in structural mechanics of ships.

Practical calculations of structures of individual hull's elements are performed in accordance with principles of the given classification society. For example the resistance moment of the transverse section surface area of the bottom frame transverse which is part of the frame ring must be at least:

$$W = k \cdot a_1 \cdot B_1^2 \cdot (H + T) + 20 \text{ [cm}^3\text{]} \qquad (335)$$

where:

a_1 - the spacing of solid floors [m],

B_1 - the spacing of solid floors between bracings, but at least 0.5 B. Bracings of a solid floor are considered the hull's sides, longitudinal bulkheads and inboard sides provided they extend to the bottom plating,

k - a coefficient equal to:

- 1.2 - in case of longitudinal system of the bottom and sides stiffening,
- 1.6 - in case of transverse/longitudinal system of the bottom stiffening and in case of transverse system of sides stiffening,
- 1.75 - in case of transverse system of the bottom stiffening and in case of longitudinal system of sides stiffening,
- 2.0 - in case of longitudinal system of the bottom and sides stiffening for ships transporting heavy goods (e.g. ore),
- 3.0 - in case of longitudinal/transverse system of the bottom stiffening and in case of transverse system of sides stiffening for ships transporting heavy goods (e.g. ore).

Each girder of the structure involves a certain segment of the plating which this girder is welded to into collaboration. The breadth of the co-acting strake C_0 is determined for individual structure elements with principles of the relevant classification society. The profile of a girder is determined so its moment of resistance together with a co-acting strake of the plating C_0 is not smaller than the one calculated using the above mentioned formula, or a similar one.

Appropriate bending strains of the entire ship's hull as a flexible girder are called an overall vibration, and variable bending strains of individual plates of the plating, girders, gratings and other elements of the hull are called a local vibration.

Each structural element and the hull as a whole is characterised with an inherent frequency which is narrowly defined and which, of course, must be calculated. The frequency is a function of the moment of inertia of the transverse section and the length of the structure, and it also depends on the load and character of its distribution along the girder.

A complex vibration of the system, i.e. such a vibration which has more degrees of looseness, splits up to simpler degrees - to a so called first, second, third, etc. tone. Each vibration tone has an inherent frequency. If the frequency of inherent oscillations of any tone is near or identical with the frequency of variable pulse forces (e.g. propeller speed, engine speed or their multiples), then there arises the resonance and afterwards also bending stresses in the structure's material. Many of these stresses particularly evoked with the local vibration may be very large. These vibrations often cause ruptures in the plating as well as in the girder's elements and lead to their destruction. Moreover the vibration worsens living conditions in a ship and sometimes prevent mechanisms and instruments from their normal operation.

That is why the calculation of vibration aims to determine frequencies of inherent oscillations of the hull and its elements so there does not happen their identification with the frequency of excitation forces (frequency of inherent oscillations must differ from the excitation ones in at least 15 % to 20 %). This condition may be fulfilled e.g. with increasing the structure's stiffness or decreasing the values - magnitudes of pulse forces, i.e. with a suitable positioning of main and auxiliary engines, or their equilibrating or positioning on locking washers, etc. The issues of vibration are studied in the branch of science called "vibration of a ship" and the requirements for the hull's vibration as well as practical methods for calculating the vibration are also presented in principles of the relevant classification society.

7.1.4. Standardisation of a Ship's Strength

The third phase of the strength calculation is in fact the evaluation of data obtained in the first two phases, i.e. based on the values of bending moments and stresses evoked with them by the overall and local load in individual parts of the ship's structure these stresses are compared to the tolerable ones.

Currently there are used two methods to check the structure's strength; by tolerable stresses and by allowable (destructing) loads.

In compliance with the first method the hull's strength is considered sufficient if the greatest summary stresses $\sigma_{c,m}$ within its elements do not exceed the tolerable stress$[\sigma]$, i.e.:

$$\sigma_{c,m} = \sigma_c + \sigma_c \leq |\sigma| \tag{336}$$

The summarisation of overall and local stresses in individual longitudinal elements of the hull is performed algebraically: in the sections on bracings and in the centre of each girder at a bending as well as flexure of the ship downwards, so their most dangerous value can be found.

Tolerable stresses $[\sigma]$ are standardised by the classification society and their values range from 0.30 ÷ 0.95 of the material's characteristic stress $[\sigma] = (0.30 \div 0.95)\sigma_\tau\ \sigma_\tau = R_{eH}$.

Smaller values of tolerable stresses are adopted for elements where the ratio of stresses by the overall hull's bending σ_c to the total stress $\sigma_{c,m}$ is equal to 1 as well as for suppressed elements of the hull girder where there exists a risk of the stability loss. Multiple destructions - breaches of ships happen as a result of the plating stability loss and often also with ordinary - light longitudinal stiffeners in a suppressed zone of the ship's hull. In case of the ship breakage the plating bulges in the form of a fold which continues in the transverse direction of the ship, and the structure girders located in this area detach from the plating in welds and are deformed, too. Damages (breakage) of longitudinal elements of the "pulled zone" of the hull girder in fact represent the result of big hull's deformations after losing the stability in the "suppressed" zone.

The critical (allowable) bending moment is such a moment when there in the most distant elements of the hull girder from the neutral axis arise stresses of the value achieving the characteristic stress. Under the condition $\sigma_t = \sigma_\tau$ the reduction coefficients of plates, the position of the neutral axis z_{max} as well as allowable moment of resistance are determined.

$$W_{cr} = \frac{I_{ycr}}{z_{max}} \tag{337}$$

Then the critical bending moment will be as follows:

$$M_{cr} = \sigma_{\tau}.W_{cr}. \tag{338}$$

W_{cr} and M_{cr} are determined separately for the flexure downwards and upwards (deflection and crippling) so it is possible to determine their minimum values.

The strength of the ship's hull is considered sufficient if the coefficient of the critical moment reserve $k^{'}$ is:

$$k' = \frac{M_{cr.min}}{M_{calc.max}} \geq (1.35 \div 1.50). \tag{339}$$

Greater values k' are for elements which also transfer the local load, and smaller values $k^{'}$ exist if elements do not transfer this load.

The analysis of "breaches" of river ships proves that approximately 35 % of all disasters happen during loading of the ship, approx. 20 % during their unloading, and approx. 20 % when they run aground. As a result of this statistic data we can conclude that a proper performance of loading and unloading operations is an important prerequisite to prevent disasters of this type.

To design the basic rule for loading a ship with cargo and its unloading there is determined a supplemental bending moment in the most dangerous central section of the ship after receiving the load in the point with the distance x from the central section of the main frame.

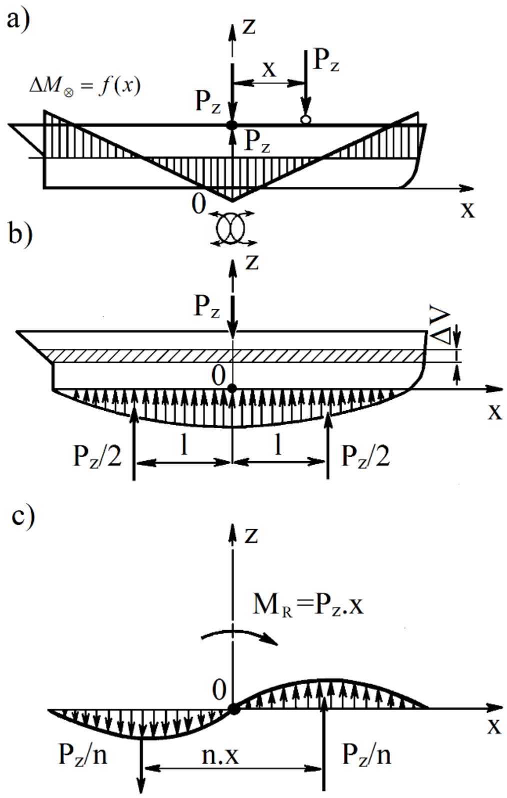

Fig. 55. Determination of a Supplemental Moment in the Plane of a Main Frame from the Concentrated Load Pz [Authors, 4]

a) coordinate of a supplemental moment on the main frame with the load by supplemental load P_z in the distance x, or the line of acting impact P_z on the main frame; b) load by P_z if its point of action is on the main frame; c) load by the righting moment $M_R = P_z.x$

If there on the main frame are applied two fictional forces (P_Z, P_Z) of the same size, acting against each other which together with the inclination moment $M_{dif} = P_Z.x$ (Fig. 55a) evoke an increase of the ship's draught ΔT, and thus an increase of displacement forces which are proportional to coordinates of the water line, the supplemental inclination moment evokes supplemental displacement forces which must be in mutual equilibrium.

The supplemental bending moment $\Delta M_{P\otimes}$ in the middle of the ship is determined as a sum of moments by displacement forces from the left side of this section and by the concentrated force P_Z according to the figured diagrams.

$$\Delta M_{P\otimes} = -\frac{P_Z}{2}.\,l + \frac{P_Z}{n}.\frac{n.x}{2} = -\frac{P_Z}{8}.\frac{L}{(2-C_W)} + \frac{P_Z.x}{2} \qquad (340)$$

where:

$l = \frac{L}{4(2-C_W)}$ - the approximate distance between the main frame and the centre of the half of the area,

n - the coefficient dependent on the water line form,

C_w - the coefficient of water line fullness.

This formula makes it clear that the adoption of the load into the point $x = 0$ evokes there in the central section on the main frame a supplemental bending moment downwards, and during the unloading a supplemental bending moment upwards, which is equal to:

$$\Delta M'_{P\otimes} = -\frac{P_Z}{8}.\frac{L}{(2-C_W)} \approx \frac{P_Z.L}{2}. \qquad (341)$$

Receiving the cargo on the edge of a loading hatch ($x \approx 0{,}4L$) there in the centre of the frame arises a supplemental moment upwards and during the unloading a supplemental moment downwards, which is equal to:

$$\Delta M''_{P\otimes} = +\frac{P_Z}{2}.\,0{,}40L - \frac{P_Z}{8}.\frac{L}{(2-C_W)} \approx \frac{P_Z.L}{13}. \qquad (342)$$

If the cargo is received into the point located in the distance:

$$x = \frac{L}{4(2-C_W)} \approx \frac{L}{4}, \qquad (343)$$

then the supplemental bending moment on the main frame will be equal to zero. The supplemental bending moment will be equal to zero or will approximate this value if the load is received into the central or outer section of the loading hatch at the same time. In order to prevent increased stresses the loading operations are required to be performed in layers simultaneously from points 1 and 2 towards the ship's forebody and from points 1′ and 2′ towards the stern. The unloading must be performed in the opposite direction, as it is represented in the following figure.

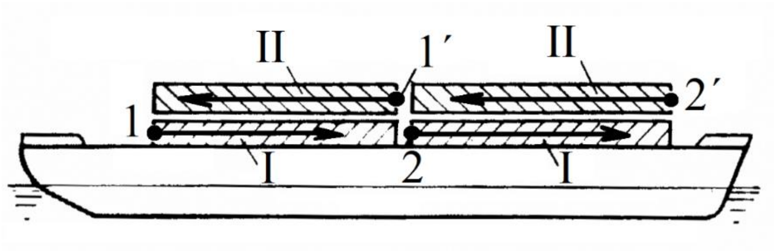

Fig. 476. Scheme of Burdening or Lightering a Ship during Concurrent Action with two Cargo Handling Equipments [Authors, 4]

7.1.5. Direction for Loading and Unloading

The sequence of loading (unloading) a ship is given with a so called "Direction for Loading and Unloading a Ship". These directions are worked out by a ship designer for individual types of ships and are approved by a relevant classification society and shipping company. Diagrams and instructions presented in the above mentioned direction are mandatory for the port staff who decide about the method of loading and unloading a ship. Often there are worked out standard collections of instructions for loading and unloading of series cargo ships and boats intended for the port staff.

Nevertheless, unloading operations are required to be performed when water surface is not wavy. Before any loading operations start it is necessary to check whether the reserve of water - under-keel clearance (depth under the ship's bottom) is sufficient, and thus whether the ship does not come into contact with the bottom of the port basin during handling with the cargo in the port.

The cargo should be distributed evenly along the length of each cargo space. The distribution of cargo in individual cargo spaces should accommodate their tolerable load capacity with respect to the entire load bearing capacity of the ship.

Wood, certain kinds of metallurgic material, etc. are loaded into cargo spaces in bundles and layers, and it is also necessary to maintain an analogical sequence.

During the loading of grain and similar loose bulk cargo with the ability to "flow" in the cargo space, certain measures must be taken to prevent this effect, e.g. through creating longitudinal bulkheads, though temporary ones.

Some bulk substrates are allowed to be loaded in the form of a conic shape. Such examples include coal, ore, etc. However, their unstableness cannot be greater than it is allowed in the "Direction for Loading and Unloading a Ship".

During the loading of stone using a grab there in the first layer a grab (polyp) must be lowered just above the deck of the loading hatch. Free fall of stone on the deck is not allowed by any means. Loading of a ship is usually performed in two to three layers, only rarely in one layer.

The "Directions for Loading and Unloading a Ship" also contain loading scales. After finishing the loading the ship with a full rate of fuel and stock cannot exceed the draught recommended in the above mentioned direction.

Diagrams and text of the direction present the sequence of loading works for the given type of the ship and for the given type of the cargo when various methods of mechanised loading and unloading are applied, i.e. how many layers a ship can be loaded in or unloaded from, as well as other necessary recommendations.

7.2. DESIGN OF METAL SHIPS

7.2.1. Choice of the Material of the Hull and Transverse Sections of the Hull's Girders

The construction of ship's hulls requires materials which are highly strong and have a high modulus of elasticity, high plastic and impact toughness, good welding property and constancy against corrosion, and if possible, their unit price is not high.

High strength of the material enables to decrease dimensions of individual hull's elements, e.g. their weight, but at the same time it enables to dispose with a high modulus of elasticity in order to provide the required stiffness and stanchion stability.

High plasticity and impact toughness ensure the formation and spreading of ruptures in the hull's structure while hits and dynamically variable loads are acting.

A good welding property is characterised with a constancy of mechanic properties in the welding spot and their stability against the basic material.

High constancy against corrosion prevents eroding of the metal itself which reduces additives for the protection against corrosion, and thus it ensures the required strength even after a certain life time expires. For current ship steels the constancy against corrosion is very low and therefore it must systematically be smeared from inside and outside constantly one-two times every four years, since the coat of paint gets rapidly eroded due to damages or impact of chemically corrosive elements. Costs of coatings significantly increase the operation costs of a ship. This is the reason why the efforts to increase anti-corrosive constancy of metal while eliminating coatings of the hull represent one of the hottest challenges in the shipbuilding industry.

Currently the majority of hulls of river ships and ships intended for a mixed river-sea navigation technology is constructed from (killed and semikilled) silicon steel, cat. A. ČSN 11378.1-LL and 1425.1 according to ON 420180 with the characteristic stress $\sigma_t = 235\ MPa$, and relative extension $\delta_{10} = 20$ %.

Since bending moments and stresses by the overall bending increase proportionally to the displacement and length of the ship, for ships longer than 80 - 100 m it is more useful to use steel with a higher strength or so called Cor-Ten steel with the characteristic stress $\sigma_t \geq 300\ MPa$. These steels are more expensive due to the fact they contain alloys of manganese, silicon, nickel, chrome and other substances of volume 0.5 % - 2 % and their application is justifiable from the economic point of view, provided the reached economics of the hull's weight enables to keep the price of the ship with regard to the effective loading capacity on that level as if the ship were made of standard carbon steel.

Statistic data inform us that if the whole ship's hull is made of Cor-Ten steel with the characteristic stress $\sigma_t = 343\ MPa$ when compared to the common stress $\sigma_t = 235\ MPa$, the weight of the cargo ship of the length $L = 120m$ decreases to 12 % and the initial price is cut to 4 %. For ships of the length $L = 80m$ these indicators are only on the level of 7 % and 1.5 % and for ships of the length $L = 40m$ the weight of the ship almost does not change as a result of the fact that dimensions of the plating's thickness and dimensions of profiles girders do not have to be greater than it is necessary to ensure the strength, due to their life time.

The bigger the characteristic stress of steel is, the smaller dimensions of sections of individual structure elements are, however, their stability decreases, too. That is why it makes no sense to use steel with the characteristic stress $\sigma_t \geq 400\ MPa$ for the shipbuilding, even not for ships longer than 150 m.

Application of Cor-Ten steel is mostly profitable when used in combination with standard carbon steel and when it is used to manufacture only the most strained hull's parts - longitudinal framing of loading hatches and/or longitudinal stiffenings of the deck and bottom, or other special profiles and stiffenings. Difficulties related to the welding property of steel which arise during such technological procedures, can be overcome without using loading equipment.

Although aluminium-dural alloys would satisfy the requirements mentioned above, unfortunately, they are not used in the construction of standard ships. Although these materials are light and resistant against corrosion and they are strong and plastic, too, they are, however, very expensive and feature a small modulus of elasticity E (3-times smaller than in case of steel).

Light alloys are used for the construction of small, fast passenger ships only.

For the construction of ship's hulls there are used steel sheets, profiles and strips.

Sheets with the thickness of less than 4 mm are labelled as thin sheets, and sheets with the thickness of more than 4 mm are labelled as thick sheets.

For girders of a standard - light structure there are used sectional bars, such as bulb angles and bulb flats (a special ship profile).

Frame girders are usually profiles shaped in "T" form and most often they are produced directly in a shipyard - thick strips (often made of rolled strip steel) are usually welded to thin web plates.

Recently there have widely been applied compress bent profiles of the same thickness, but different breadth, or height of the web plate. The usage of such a bending under the press enables to decrease the operoseness in 20 % despite the construction quality of the profile is decreased in 30 % ÷ 35 % as a result of the same thickness of the web plate and framing.

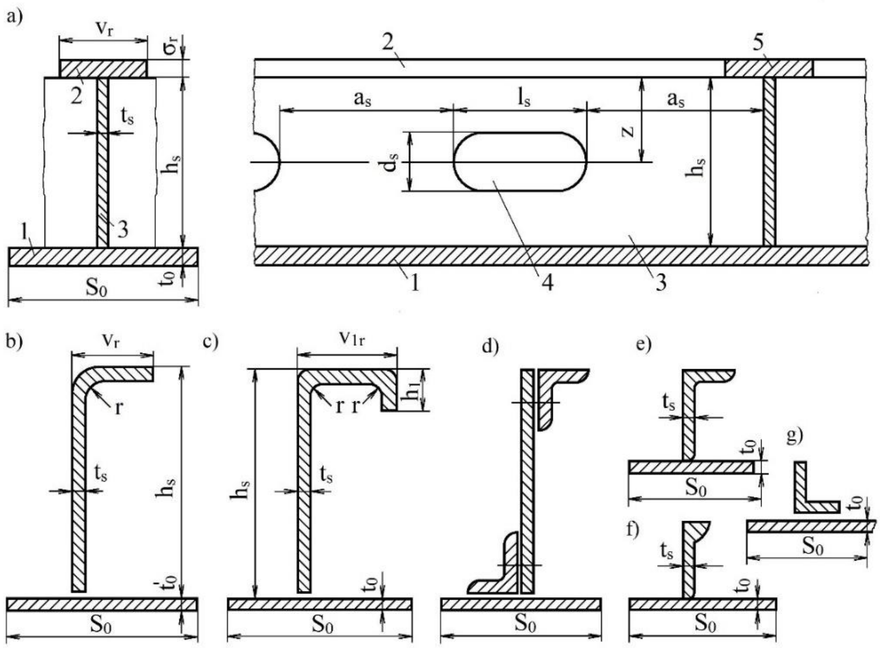

Fig. 487. Transverse Sections of Frame and Standard Girders (Stiffenings) [Authors, 4]

a) welded T-profile (a transverse and side view); 1 - co-acting strake of the plating, 2 - moulding, 3 - wall, 4 - slot, 5 - frame girder of a perpendicular direction; b) flanged profile; c) flanged profile with a supplementary flanged moulding; d) riveted frame girder; e) standard girder with a bulb-angle steel; f) standard girder with a bulb-flat steel; g) riveted standard girder

Construction quality of profiles is characterised with a ratio of a sectional modulus (moment of resistance) to its surface area:

$$k_k = \frac{W}{F}. \qquad (344)$$

Asymmetry of the profile with respect to its wall significantly decreases its strength in the bending, stiffness and construction quality of the profile which is evaluated with a reduction coefficient ϕ_r, which is then used as a multiple of the area of the girder's flange (a so called opposite free strip) when calculating its area as well as moments of inertia and the section modulus.

Construction quality of different profiles represented in Fig. 59, when the surface F is roughly the same, is given in the following table.

Table 30

Construction Quality of Profiles

Type of the Profile	**Surface Area of the Transverse Section without a Co-Acting Strake of the Plating F [cm^2]**	**Section Modulus of the Profile with a Strake of the Plating W [cm^3]**	**Construction Quality of the Profile** $k_k = \frac{W}{T}\left[\frac{cm^3}{cm^2}\right]$
Welded T-Profile	11.4	113.0	9.9
Bent Profile	11.5	70.0	6.1
Unequal Bulb Angle	11.4	56.0	4.9
Bulb Flat	11.2	59.5	5.3

Source: [Authors, 4]

For the sake of comparison the section modulus is calculated with the one co-acting strake of the plating it is welded on, and for the bent profile the reduction coefficient ϕ_r is taken into account.

The table above makes it clear that each unit of the welded girder's surface area has approximately 1.6 to 2-times greater section modulus than in case of a bent profile and sectional bar. It is obvious that the weight at the same strength will range in the same ratio, regardless of a greater operoseness during the production of T-profiles which are, however, considerably more effective than other profiles.

7.2.2. Methods for Joining Elements of the Ship's Hull and Construction of Girders

Individual parts of girders and plating are mutually joined using welding and nowadays only very rarely using riveting alloys which are hardly to be welded.

Electrical welding in shipbuilding is performed by means of automated and semi-automated devices as well as manually using electrodes covered with a thick layer of added packaging materials. In case of a mechanised - automated welding there is achieved a much higher quality of weld as well as saving of electrodes; but mainly the labour productivity is increased 5 ÷ 10-times and proportionally to it the price of welding works is cut.

Dimensions and the size of welding elements as well as technological welding procedures are defined with principles of the classification society.

To join sheets of the plating of walls and strakes among them butt welds are used, either without any edge shrinkage, or with a skew edge shrinkage. Welds without edge shrinkage can be used for automated welding of rather thick sheets with the thickness up to 15 mm.

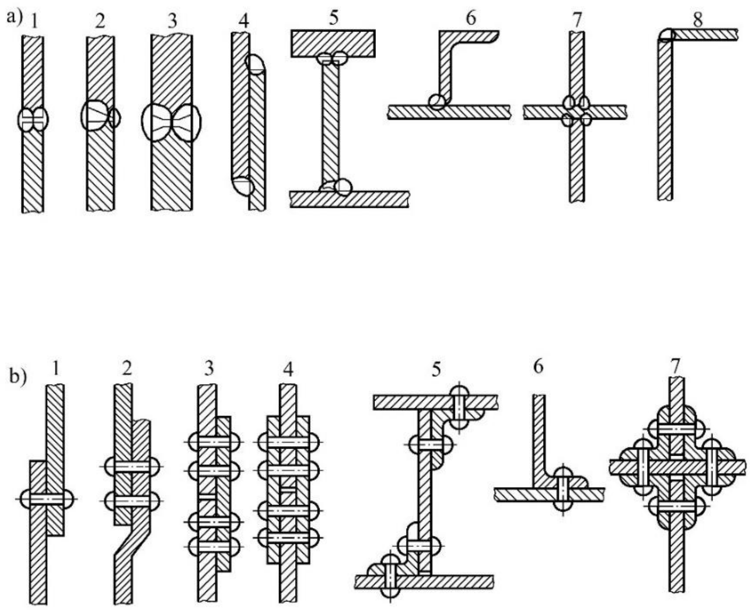

Fig. 58. Joints of Elements of Ship's Structures [Authors, 4]

a) types of welded structures: 1 - butt weld without any edge skewing, 2 - butt weld with a one-side edge skewing of V type, 3 - butt weld with a two-side edge skewing of X type, 4 - lap-jointed weld, 5, 6 - perpendicular weld of a sheet without any edge skewing, 7, 8 - corner welds; b) types of riveted structures: 1 - lap joint without a crippled flange, 2 - lap joint with a crippled flange, 3 - one-side joint with a heel strap, 4 - two-side joint with heel straps, 5 - riveted T-profile, 6 - riveted bulb angle, 7 - corner joint

A weld with lap-jointed sheets is not recommended and/or allowed because such a weld is not strong enough, and it even requires a greater material consumption for lap-jointing; moreover surface areas between lap-joints erode intensively as a result of residential humidity not being ventilated.

Welding of walls of girders and strakes or profiles of girders to the plating is done with corner welds.

Riveted structures are performed using various methods. Mostly they are very difficult from the production point of view, and thus the ship's price is increased in 30 % on average when compared to welding; moreover they are ecologically inappropriate due to the production noisiness.

Shortcomings of welded structures include their decreased operational ability in the hull girder of the ship due to residential stresses along the weld at its cooling and due to deformations mostly during welding thin plating sheets. When the technological procedure of welding is incorrect, stresses may be so high that they cause ruptures and a more serious damage of the ship's hull may occur.

Stresses by welding and deformations of the ship's hull may be reduced to a safe level if technological procedures and recommendations within principles of the classification society are adhered to. For example it is not possible to allow for exceeding the weld and concentrating welds in one point. Often it is necessary to exclude welds mainly from slots and other areas where stresses concentrate.

Dimensions of a transverse section of girder's profiles are defined based on the condition of strength, i.e. they must feature a required section modulus at a high quality of the profile. Mutual relations of the profile's dimensions are chosen so the wall neither the co-acting strake loses its stability unless pressure stresses by the bending in them exceed the characteristic stress σ_t.

The ratio of thicknesses of welded elements should not exceed double values so it would be possible to ensure a weld of good quality. In case of a higher difference between thicknesses it may happen that the thin element is overburnt and the thicker element does not have time to melt due to an intensive conduction of warmth with the material mass.

Therefore an analytical approach to the design of frame carrying profiles is recommended; a basic value is represented with the required section modulus of the girder W, which is set with principles of the classification society. Usually the thickness of the plating t_0, which the girder must be welded to, is a known value, too.

The thickness of the girder's wall $\boldsymbol{t_s}$ [mm] is set by the rule as a thickness of the plating $\boldsymbol{t_0}$, or in case of frame girders of a T-profile the thickness is 1 - 2 mm smaller if $t_0 > 5\ mm$, and in case of flanged profiles it is 1 - 2 mm greater if $t_0 > 6\ mm$, i.e.:

$$t_s = t_0 \pm (1 \div 2). \tag{345}$$

The height of a wall h_s out of the condition for a minimum of the surface area of a transverse section with the given value of W is determined as follows:

$$h_S = k.\sqrt{\frac{W}{t_S}},\ W = \frac{b.h^2}{6},\ I = \frac{b.\square^3}{12}. \tag{346}$$

For welded T-girders it is the value $K = 1{,}12$; for flanged T-girders it is the value $K = 1{,}4 \div 1{,}55$.

Afterwards it is necessary to specify h_S and t_S out of the condition for stability of a wall:

$$\frac{\square_S}{t_S} \leq m_S. \tag{347}$$

For steels of a standard quality: $\sigma_t = 235\ MPa,\ m_s = 80$.

For steels of a higher strength: $\sigma_t \cong 300\ MPa,\ m_s = 65,$

$\sigma_t \cong 400\ MPa,\ m_s = 50.$

For profiles made of aluminium alloys $m_s = 60$.

After rounding the value h_s into the nearest standard dimension the surface area of the wall $f_S = \square_S.t_S$ is calculated.

The breadth of the co-acting strake of the plating is determined with principles of the classification society or approximately $c_0 \leq \frac{1}{6}.l$ (where l - length of the girder.

Wherefrom the surface area of the co-acting strake of the plating is

$$f_0 \approx 60.t_0^2 \tag{348}$$

or

$$f_0 \approx C_0 \cdot t_0 \approx 0{,}5 \cdot a_0 \cdot t_0, \text{ however, } f_0 \leq \frac{l \cdot t_0}{6}. \tag{349}$$

The section modulus of a transverse section of frame girders of T-profile and flanged Γ-profile will be determined using the approximate formula by I. G. Bubnov as follows:

$$Ws\left[\frac{f_0}{6}\frac{(4.f_0+f_S-2.f_p)}{(2.f_0-4.f_S)}+f_p\right]_{min}. \tag{350}$$

This formula, when the section modulus W is known, will lead to determination of the surface area of the profile flange using the following formula:

$$f_p = \frac{4.f_0(3.W-\square_S.f_S)+f_S(6.W-\square_S.f_S)}{4.\square_S(3.f_0+f_S)}. \tag{351}$$

The thickness of the flange t_p is determined with conditions for welding property and profile quality as 2 ÷ 4 mm greater than the thickness of the wall, but not more than twice the thickness of the wall.

$$t_p \leq 2.t_S \tag{352}$$

The breadth of the flange will then be:

$$b_p = \frac{f_p}{t_p}. \tag{353}$$

From the point of view of the flange's stability it is true:

$$\frac{b_p}{t_p} \leq m_p. \tag{354}$$

For T-profiles when $\sigma_\tau = 300\ MPa$, $m_p = 19$, and for $\sigma_\tau = 300\ MPa$, $m_p = 16$.

For flanged Γ-profiles $m_p \leq 15$.

For bulb-flat profiles, so called bulb flats $m_p \leq 25$.

For T-profiles made of light alloys $m_p = 14$.

In all cases, however, the following must be true: $m_p \geq 8 \div 10$.

After specifying dimensions of the girder it is recommended to repeatedly verify the section modulus using the formula by I. G. Bubnov.

The height of flow openings (water flows) in bottom stiffeners should not exceed $\frac{1}{5}\square_S$, and it should not exceed 90 mm either. The length of flow openings must not exceed 15-times the thickness of the plating adjacent to the stiffener, or 150 mm. If dimensions of flow openings are required to be enlarged for the sake of a greater flow (e.g. of diesel oil or water in tanks), weak areas of stiffeners must be reinforced.

There in walls of frame girders are often made some round or oval openings in order to decrease the weight. These oval openings are subject to relevant principles of the classification society.

All slots for openings in longitudinal stiffeners are oriented in the direction of the ship's axis as a matter of principle.

All right-angled slots in longitudinal stiffeners must have their corners rounded with a radius less than 0.1-times of the slot's breadth.

If there are more slots in the longitudinal direction of the ship, it is necessary to locate them within one line, if possible. In one transverse section of the ship there should not be located more slots, or lightening holes. Longitudinal discontinuous stiffeners (e.g. in decks) must be terminated in special transverse joints they must also be welded to.

There in web plates of girders strained mostly at bending it is possible to create lightening holes with rounded corners, provided the following conditions are met:

- slots must be of such a height that their edges are not less than 0.25 of the girders' height distant from the web plates' edges,
- the length of slots must not be greater than the twice of their breadth, and the distance between edges of two adjacent slots must not be less than the length of the smaller one.

Girders strained mostly at shear or torsion must not be equipped with lightening holes.

Slots in web plates of girders must not lie in the area under joining brackets and, of course, near shores. The end of a slot must be in the distance of at least ½ of the girder's height from the end of the bracket.

The distance between edges of any slots in transverse frame stiffeners and edges of slots, longitudinal stiffeners pass through, must be at least equal to the height of these stiffeners.

Since there in principles of classification societies no hatchways in girders higher than $h_s \geq 500\ mm$ are mentioned, we will adhere to general principles that the length to the breadth of the hatchway is in the relation $d \cdot l \leq 0.5 \cdot h_s \cdot 0.75 \cdot h_s$ (from the practical point of view standard hatchways share the dimensions $d \cdot l \leq 320 \cdot 450\ mm$ or $d \cdot l \leq 400 \cdot 600\ mm$). If slots for hatchways are greater than the given values or if the ratio is $\frac{h_s}{t_s} \geq 80$, even if the girder lacks any slots, it is necessary to toughen the walls using so called horizontal frames of stiffness.

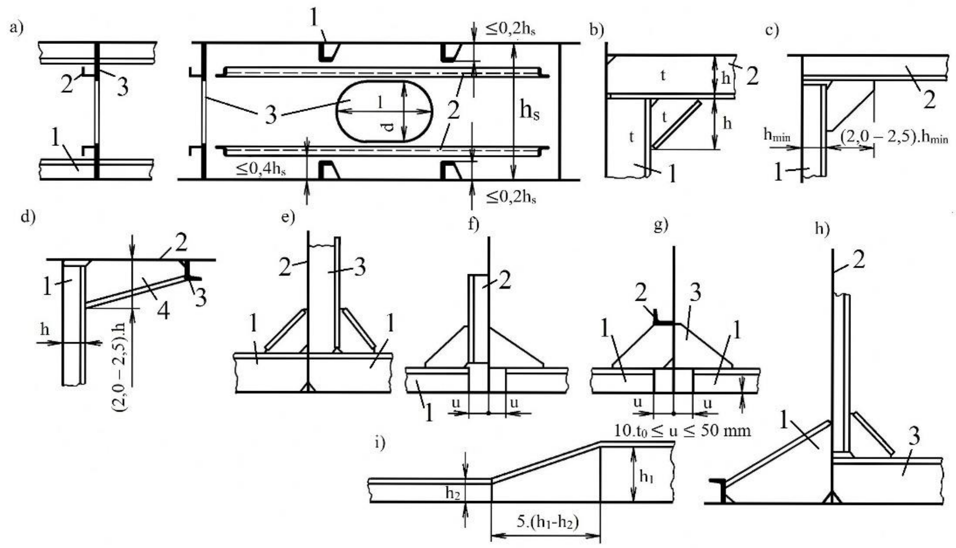

Fig. 59. Construction of Connect Nodes [Authors, 4]

a) standard girders: 1 - frame girder, 2 - horizontal stiffening frame of a web plate, transverses located at openings and hatchways, 3 - transverse in the double bottom (a hatchway in a transverse's wall); b) joint of the deck and side with frame girders of the side (1) and a beam (2); c) joint of the deck and side with standard stiffeners of the side (1) and a beam (2); d) clamping of the upper part of a standard side frame (1) with the deck (2) at a mixed system of stiffening; e) clamping of frame girders (1) to a frame web plate (3) of a watertight bulkhead (2); f) clamping of standard longitudinal girders (1) to a standard web plate (2) of a watertight bulkhead; g) clamping of standard girders (1) to a bulkhead without any standard web plates (2 - special horizontal frame to lean the brackets against - 3); h) clamping of a frame girder (3) on a bulkhead (2), 1 - bracket on the opposite side of the bulkhead; i) fluent change of the girder's profile

All standard stiffeners, mostly made of ship T-bulb, are produced in compliance with principles of the classification society.

In crossing nodes of standard continual girders with frame stiffeners perpendicular to them there in walls are cut trapezoidal slots with the overall height not greater than $0.4 \cdot h_s$. The vertical wall of a standard girder is required to be welded to the wall of the frame girder or to be stiffened with a bracket.

When two frame girders cross, one of them is cut and welded to a perpendicular girder with a double-sided continual weld.

Girders which join in a perpendicular direction are stiffened with a bracket.

The thickness of the bracket must be at least twice the thickness of the thinner of joined stiffeners, or if the bracket is extended, then it must be 2.5 % of the carrying length of the bracket, or 2 % of this length, provided the bracket has a flange on its free side, depending on which value is greater.

Dimensions of bracket sides at the joint of frame girders must be greater than the height of the frame girder wall, and if they are not equal, then they must be at least such as they are in the smaller girder. At the same time the bracket is welded to girders with a continual weld.

To connect the end of a standard girder with segments of a not-stiffened plating or stanchion of a bulkhead, which leans against the plating, a bracket brought to the nearest stiffener perpendicular to the bracket must be used there or such a stiffener must be placed specifically.

Bevels are constructed the way as it has already been mentioned. Standard longitudinal girders are usually not brought up to bulkheads or transverse frames rings of stiffeners due to a flow opening formation. The distance $u \leq 10 \cdot t_0$ is important, however, it cannot be more than 50 mm (where t_0 - is the thickness of the plating).

The design and building of the ship's hull require to adhere to the principle of continuum, or cohesion and confinedness of stiffeners in one plane according to the pattern of construction nodes, which have already been mentioned. The passage where the height of a longitudinal girder is changed, must be formed fluently in that compartment whose length is not smaller than 5-times the difference of girders' height.

Longitudinal girders, which are discontinued on transverse bulkheads or frame transverses, must be terminated there on the other side of the bulkhead or frame transverse with a progressively decreasing bracket brought to the nearest standard transverse.

Girders of ships intended for navigation in drift ice must be stiffened in accordance with special requirements of the classification society.

7.2.3. Outer Plating of the Hull and Deck and Hatch Moulding

Outer plating of the hull and deck is welded from individual sheets with longer sides oriented in the longitudinal direction of the ship. These sheets form so called plating strakes. Welds of sheets which are oriented transversely are called transverse joints and those in a longitudinal direction are called longitudinal joints.

The distance between longitudinal joints and girders walls must be 1/3 or 1/4 of the distance between these girders.

To determine the count and dimensions of plating sheets needed for the ship building a special plan is drawn; it is called the **expanded plating of the hull and deck** and it contains all transverse and longitudinal joints indications.

Plating sheets often have a double curvature and cannot be expanded into the plane. That is why the development of plating is drawn only across the ship along frames onto one side from the axis plane, whereas dimensions in the longitudinal direction of the ship (along the water line) are kept on the plan in a deformed state.

On the expansion plan of the plating and the deck there are indicated main stiffening girders which provide the overall and local strength of the ship's hull. The role of individual plating strakes is, however, different and it is given mainly with the distance from the neutral axis of the hull girder.

The biggest load lies mainly on deck sheets, hatch side coaming as well as upper strakes of the side plating, a so called fender, but also a rounded bilge strake in the area of transition of the bottom and the side. It is these strakes which are important for determination of their thickness.

Besides the relevant classification society sets minimum tolerable thicknesses of individual plating strakes independently from σ_τ of steel, under the condition of the ship's operational life, whereas the corrosion and mechanical abrasion as well as issues of operational reliability are taken into account. Minimum thicknesses of the hull's sheets are given dependent on the length and class of the ship.

Thickness of the outer plating of the bottom t_0 is the fundamental dimension which serves as the ground. Thicknesses of other strakes are usually expressed as a ratio to its thickness.

Sheets of outer plating of the bottom and sides in the forebody of the ship t_c are thicker in 0.5 to 2 mm as a result of the fact that these sheets must tolerate hits of waves intensively.

Along the axis plane on the bottom there passes a strake of the plating called a garboard plate. The breadth of this plate must be at least 600 mm and the thickness $t_{kp} = t_0 + 1$.

The plating of outer sides (from the upper edge of the bilge plate) may be reduced in 1 mm, i.e. $t_{bp} = t_0 - (0 \div 1)$.

On the other side, the plating of inboard sides of motor cargo ships and push boats must be 1 mm thicker - $t_{vbp} = t_0 + 1$ due to hits of the grab during cargo handling.

Ships navigating in drift ice have a so called ice strake along their entire hull; the thickness of the strake is 15 % ÷ 25 % greater than the thickness of the plating of the bottom t_0. The upper edge of the ice strake must be located at least 500 mm higher above the cargo water line and the lower edge must be located 500 mm below the water line of an empty ship, taking its potential inclination into account.

The fender's height must not be less than $0{,}2H$ (depth of the side) and its thickness must be $t_{op} \geq 1.25 \cdot t_0$ (when *L* = *20 mm*) and $t_{op} \geq 1.4 \cdot t_0$ (when *L* = *40 mm*). The fender is the most exposed part of the hull girder mostly during mooring, i.e. standing at a perpendicular edge of a quay wall.

The fundamental strake of the deck plating is the awl which runs along the side from the bow up to the stern. In case of full-deck vessels the awl breadth is usually 0.6 m and the thickness $t_{okp} \geq t_{op}$ (the thickness of the fender). There in the awl it is forbidden to create hatchways and slots with a diameter greater than $d = 20 \cdot t_{okp}$ without their appropriate toughening. The thickness of other sheets of the deck plating is a bit smaller than the thickness of the bottom $t_{pp} = t_o - (0 \div 2)$ mm.

The deck of vessels and boats intended for a deck cargo as well as the second bottom of cargo ships and boats are protected against a potential damage during loading operations, and thus they are also 3 ÷ 4 mm thicker: $t_{dd} \geq t_0 + (3 \div 4)\ mm$. In the corners of individual slots of loading hatches, feeding hoppers and hatchways there arise big concentrations of stresses. As a result it is necessary to round corners of individual openings with a radius not less than 0.1 of the opening breadth. The toughening can also be performed using thicker plates when compared to deck plates or awls. To satisfy the condition for a continual stiffening, slots in the deck and plating are required to be located onto one level along the ship.

Big slots in decks for openings of cargo spaces are stiffened with a continual hatch moulding. Their vertical walls are embedded into the cargo space up to the level of a stringer of a frame web plate and upwards they are conducted out above the deck level; they use to be equipped with a flange, but more often with a welded coaming stringer. The coaming stringer delivers stiffness and strength to the hatch moulding. For this purpose continual horizontal stringers of a moulding and vertical stiffeners of a coaming in the plane of frame rings are often used, too.

The thickness of moulding's walls is minimum $t_{ob} \geq t_0 + (2 \div 4)\, mm$ and all dimensions are, of course, toughened based on the calculation of the overall strength of the ship.

In compliance with principles of the relevant classification society the height of the moulding above the deck must be min. 300 mm due to flooding of the deck with water. Inner mouldings of hatches are usually $t_{ob} \geq 75\, mm$.

In current cargo ships and boats with a big cargo spaces throat the moulding must also provide the angular strength of the ship's hull. As a result of this the moulding runs high up to 0.8 ÷ 1.0 m and the breadth of the upper strake up to 0.6 m which at the same time enables to locate a device for travelling and sealing of loading hatches covers there on the moulding.

Nowadays, mostly during repairs of ships, e.g. of their second bottom, as part of rationalisation there are used composite structures when the space between two thin platings (with the distance of 30 ÷ 50 mm between them) is filled with various fillings, such as asphalt-concrete mixtures or other lightweight materials depending on insulating properties of the materials used. Such structures share positive properties, such as a lower overall weight, greater stability, greater strength and good sound proofing and thermal insulation.

7.2.4. Construction of Ships with a Transverse System of Stiffening

As it has already been mentioned in the section containing methods for calculating the overall and local strength, the main characteristic sign of a transverse system of stiffening is a lower distance between girders of the transverse direction when compared to longitudinal girders.

This distance is called a frame spacing and in case of river ships it is usually $a_{\gamma} \geq 500 \div 650\, mm$ besides an afterpeak and forepeak where this distance is 50 mm smaller than in the central part of the ship.

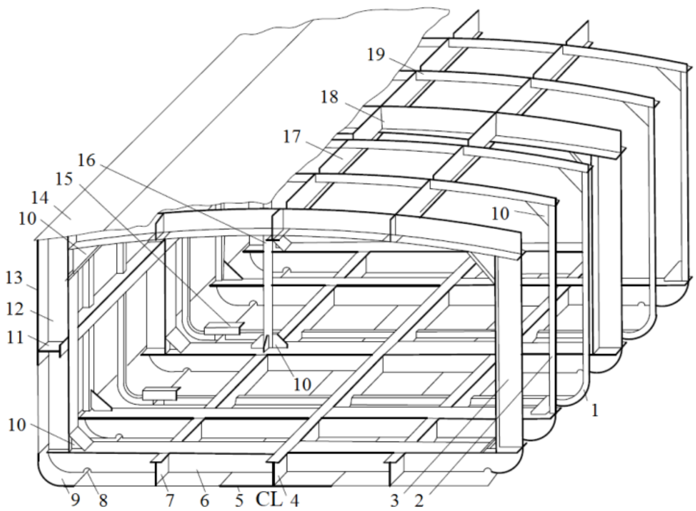

Fig. 60. Transverse System of the Ship's Framing Stiffening [Authors, 4]

1 - standard frame, 2 - reinforced standard frame (with a frame bottom transverse), 3 - frame ring, 4 - centre keelson, 5 - horizontal keel (a reinforced garboard plate), 6 - frame bottom transverse, 7 - side bottom keelson, 8 - flow opening, 9 - bilge strake of the plating (bilge), 10 - bracket, 11 - side stringer, 12 - side frame ring, 13 - sheer strake, 14 - awl, 15 - bridge (an interconnecting bulb angle), 16 - stanchion, 17 - frame underdeck girder, 18 - frame deck transverse, 19 - standard (light) deck transverses

A set of girders in one transverse plane is called a **frame**. Depending on the profile of girders entering the frame structure the following 4 types of frames are distinguished:

- A frame ring comprising welded reinforced T-profiles or flanged profiles on the bottom, side as well as below the deck (Fig. 61 b).
- A standard frame which has light profiles below the deck and on the side and on the bottom there are reinforced transverses made of a T-profile or flanged profile. Transverses of standard frames and frame rings together with longitudinal stiffeners form a bottom grating (Fig. 61 a).
- Frames with a lightweight open bottom transverse which comprises light profiles below the deck, on the side and in the bottom. The bottom transverse of a lightweight type can also be found in the level of the upper strake of frame stiffeners which are mutually tied with stiffening strakes, so called brackets. A transverse as a whole forms a barrier structure without so called wind diagonal girders.

  Open bottom transverses are under water pressure and from the opposite side they are exposed to counterpressure of the cargo stored on the brickwork. The force action of the cargo is transferred as a reaction to longitudinal keelsons.
- A light frame when all profiles on the bottom, sides and below the deck are of a lightweight type, e.g. bulb angles or bulb flats. The function of the light frame is to accept the pressure of water, and pass it through the plating sheets as a reaction to shores which are represented with longitudinal frame girders.

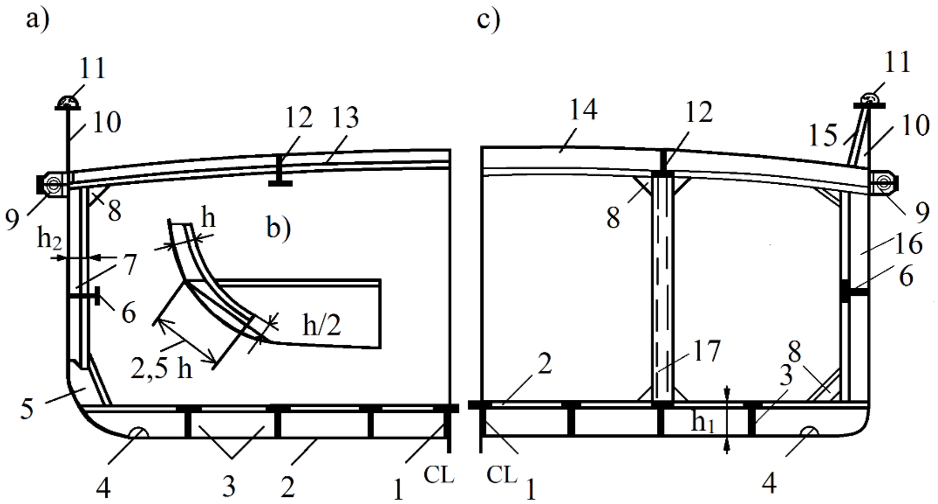

Fig. 61. Structural Manufacture of Frames with a Transverse System of Stiffening

[Authors, 4]

a) reinforced standard frame; b) frame ring; c) connection node of a standard side frame with a frame transverse of the bottom at its dead rise in end parts of the hull; 1 - centre keelson, 2 - frame transverse of the bottom, 3 - side keelsons, 4 - flow opening, 5 - bracket (a cleat) with a flanged moulding, 6 - side stringer, 7 - standard profile of a side frame, 8 - bracket, 9 - fender, 10 - bulwark, 11 - handle (wooden), 12 - underdeck girder, 13 - standard deck transverse, 14 - frame deck transverse, 15 - frame web plate of a bulwark, 16 - side frame ring, 17 - stanchion

In individual compartments of the ship's hull there can be located frames of one type (most frequently a standard frame) or of various types with alternations in a certain order. The construction of ships with the same frames is technologically simpler. Such a construction is used only in case of smaller ships, usually with a keel bottom which rises askew from the axis plane towards the side. Constructions with the same frames are used most frequently as a result of the fact that the overall weight of the construction is a bit smaller in this case. Its disadvantage is that a bit higher height of sparsely built frame girders reduces the useful volume of cargo spaces, and the constant diversity of girders makes the technology of shipbuilding a more challenging. In other words each reduction of the ship's weight is achieved with a higher price and a certain decrease of cargo spaces cubage.

That is why there exists an optimal construction for each ship depending on its type, dimensions and class from the point of view of the weight, price and operational parameters (indicators); the optimal construction usually features an alternation of various types of frames and can be determined through a comparison of different alternatives.

In ships with a transverse system of stiffening there exist efforts to construct all frame and lightweight transverse girders as continual ones. Such a solution enables to utilise automated devices for welding structure elements of frames to the plating to a large extent.

Continuous light girders pass through walls of frame longitudinal girders through special trapezoid slots; they are then mutually tied using brackets, or the vertical wall of a light profile must be welded with the wall of a longitudinal girder using a double-sided continual weld. In this case, however, a more precise adjustment and splicing of girders to each other is required.

Dimensions of sides of a bilge bracket are chosen so the ends of the bottom and the side profile do not have to be bent and so the welding of these ends can be ensured on the length 2 ÷ 2.5 of the profile's height. The following manufacture is worth a consideration, too. It is less technological in its essence because it requires a bending of the ends of the side profile by the radius of the bilge.

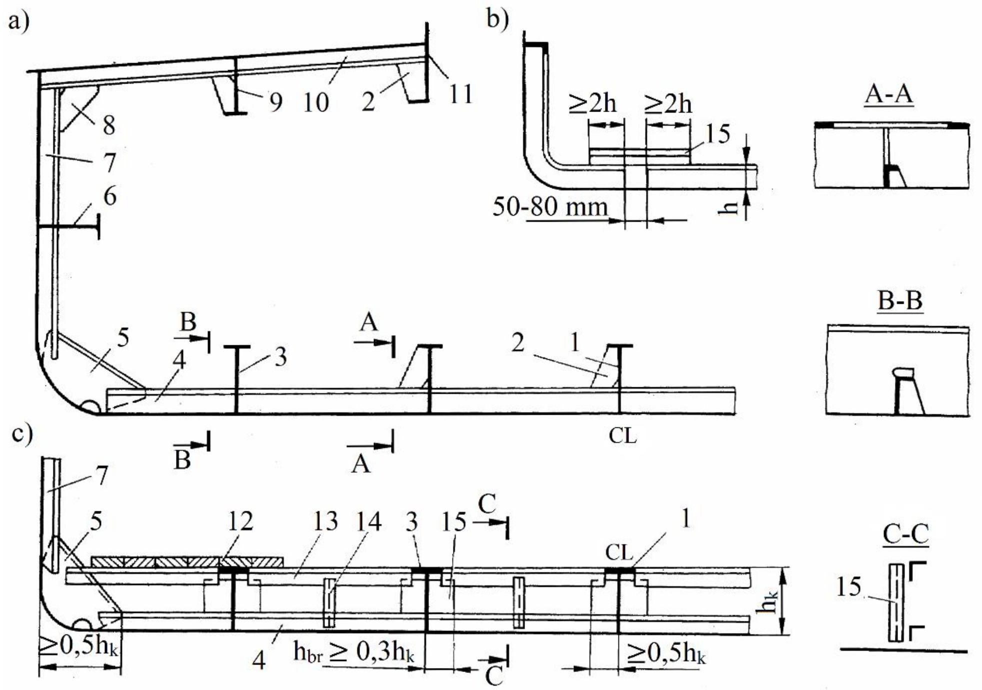

Fig. 62. Construction of Standard Frames with a Transverse System of Stiffening [Authors, 4]

a) standard frame with a bilge bracket (a cleat); b) with an open transverse with brackets (cleats); c) variant of a bilge node with a trying bulb angle - a bridge; 1 - centre keelson, 2 - trapezoid bracket (a bevel), 3 - side keelsons, 4 - standard bottom transverse, 5 - bilge bracket, 6 - side stringer (frame), 7 - standard frame of the side, 8 - bevel without a flange, 9 - deck stringer (frame), 10 - standard deck transverse, 11 - longitudinal moulding of a cargo space, 12 - wooden bottom floor, 13 - underfloor bulb angle, 14 - bridge (an interconnecting bulb angle), 15 - rectangular brackets (cleats)

A section modulus of individual girders of a transverse construction stiffening is determined based on principles of the relevant classification society in accordance with the character of load, length between shores, and type of the shore. In case of standard girders the length is given with the distance between longitudinal frame shores, and in case of frame transverses and frame beams it is either the distance between sides or between a side and a longitudinal bulkhead, or for side frame rings the shores are in the distance of the depth of the ship's side.

Dimensions of transverse sections of girders are determined in concordance with principles for girders design which have already been presented. Bottom transverses, as the most loaded girders, have the biggest sectional area equal to for example a side frame and a beam.

7.2.5. Longitudinal Girders of the Hull with a Transverse System of Stiffening

Bottom longitudinal girders are called **keelsons**. Keelsons often have the same dimensions of a transverse profile as transverses. Keelsons are those girders of the hull which directly contribute to the overall strength of the hull. Girders of such a character play a big role in the local strength, because they represent crossing girders of the bottom grating and at the same time they serve as shores for standard bottom frames.

In all ships there must be located a centre keelson attached to a stem and a sternpost. The number of side keelsons is determined based on the overall breadth of the ship. In the central part of the hull the distance between keelsons must not exceed 2.5 m. In the forebody and stern even a smaller distance is required and it depends on the class of the ship. Side keelsons are required to pass along the entire hull of the ship, from the forward up to the stern bulkhead, and they are to be straight along their entire length. Bases under main engines or at least one of them must be an underlay of the keelson.

Keelsons are formed into a T-profile from individual girders, length of which equals the distance of frame transverses.

Walls and flanges of keelsons are welded to walls and flanges of transverses using a double-sided continual weld. The process of forming girders is required to be performed especially carefully in order to observe the alignment of individual components of the keelson at their splicing points. A tolerable deviation from the alignment is one half of the thickness of the girder's wall. Otherwise the continuity of the girder as a whole is disrupted. From the technological point of view there in big ships the central keelson is manufactured as a continuous one between watertight transverse bulkheads, and transverses are cut and welded to the keel using a double-sided continual weld in this node.

Underdeck longitudinal girders are called underdeck girders and they also provide the overall and local strength of the ship. The location of underdeck girders must be in compliance with the location of loading hatches on the deck so hatch side coamings are a continuation of respective underdeck girders. Dimensions of a transverse section of an underdeck girder are equal to those of a beam.

Side longitudinal girders are called side stringers. Side stringers represent a support to standard frames which are loaded with water pressure. They bring concentrated forces (arising from hits at getting the ship ashore) to adjacent frame rings. In compliance with principles of the respective classification society there must be at least one stringer in a ship when the depth of the side is $2{,}5 \leq H \leq 4m$, and there must be at least two stringers when $H \rhd 4m$. Dimensions of a transverse section of side frames are usually equal and their construction is analogical to the construction of keelsons and underdeck girders.

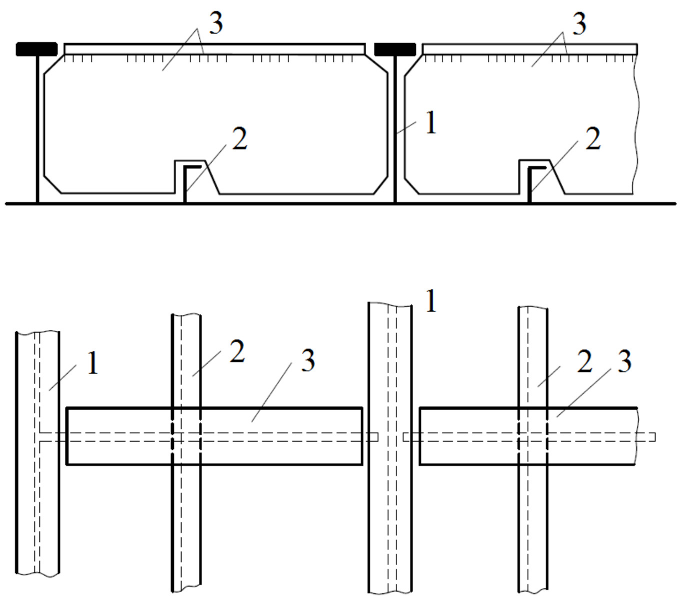

Fig. 63. Joint of Keelsons with Transverses [Authors, 4]

1 - frame transverse (continuous), 2 - standard transverse, 3 - frame keelson (discontinuous)

All girders of the longitudinal stiffening of the keelson, stringers and underdeck girders are there on transverse bulkheads mounted with brackets from both sides. If one of them ends on a transverse bulkhead, there on the opposite side of the bulkhead a skew stiffener - bracket of the length not less than two frame spacings (frame distance - spacing) must be positioned. For the sake of eliminating the development of a dangerous concentration of stresses in the area of a girder gap it is possible to discontinue two longitudinal frame stiffeners at most.

There are efforts to align underdeck girders in one vertical plane with keelsons; in this case a principle of confinedness of longitudinal girders is adhered to. It is optimal to bind bottom and deck gratings using vertical stanchions which are usually round and from a tubular profile. Thanks to stanchions both gratings co-act regardless which of them is being loaded down at that moment. The strain of stanchions is normally caused with pressure, thus they are checked for a strut stability.

A thicker position of stanchions enables to decrease the size of transverse beams and bottom transverses, keelsons and underdeck girders, however, this could lead to a worse mechanisation of loading and unloading operations in cargo spaces. On the contrary, there in these spaces exist efforts to exclude stanchions, thus it is necessary to widen the transverse section of the bottom frame stiffening. In the area of the double bottom stanchions may be rationally positioned since openness of this area is not important.

7.2.6. Pros and Cons of the Transverse System of Stiffening of a Ship

The transverse system of stiffening of the hull provides a high local strength against the acting of various types of concentrated forces which is particularly required in case of ships navigating in drift ice and often getting ashore to port facilities (port tugs, pusher tugs, small passenger and transit ships, etc.).

Due to a rather small height of girders with the transverse system of stiffening a relatively small part of useful volume of the under deck gets lost which is important mainly for ships intended for transport of light and big-size cargoes.

The transverse system of stiffening is particularly effective from the technological point of view in case of curvilinear circumferences of the hull. There is the disadvantage that outer plates of the hull plating are less stable, and thus the transverse system is used only rarely in case of ships of a greater length ($L > 60m$).

In ships of a small length (e.g. $40 \leq L < 60m$) as well as in icebreaking ships and ships of the radius $^{L}/_{H} \leq 15$ this shortcoming is not manifested because the thickness of the plating required from the point of view of constancy against corrosion and fender resistance, or from the point of view of the local strength is sufficient and also it is often overlarge from the aspect of the overall longitudinal strength of the ship's hull. That is why all small (short) ships are normally built exclusively according to the system of transverse stiffening of the ship.

7.2.7. Construction of Ships with a Longitudinal System of Stiffening

In case of the longitudinal system of stiffening the transverse girders are positioned in a significantly greater distance than the longitudinal girders. That is why there are only frame construction rings positioned in this system.

The smallest weight of the ship's hull is achieved with the frame spacing from 2.4 m to 3.6 m, however, the height of girders is significantly greater. Big brackets in such a system of stiffening also restrict the cargo space to a large extent. For these reasons the longitudinal system of stiffening can only be used for tank ships where such a spatial restriction of frame rings is not of a higher importance.

In cargo and passenger ships the frame spacing is reduced to values from $1{,}2m$ to $1{,}6m$ which leads to an increased weight of the hull's stiffening, however, on the other side a greater spatial usefulness of the inner volume is obtained.

Sparsely positioned, and thus very loaded girders of the transverse stiffening, are usually constructed as continual ones. Inside of them there are cut some trapezoid openings through which continual standard girders of the longitudinal stiffening pass; they are attached to frame rings with a bracket or with a welding of a vertical wall of a light girder to a frame's wall, whereas girders of both directions in the area of the slot must be welded to the plating at least in the distance of 200 mm using continual welds.

At the same time brackets serve as a means to increase the stability of a high wall of the transverse stiffening. The distance between brackets in case the wall is not stiffened with horizontal profiles cannot be greater than 1.5 m.

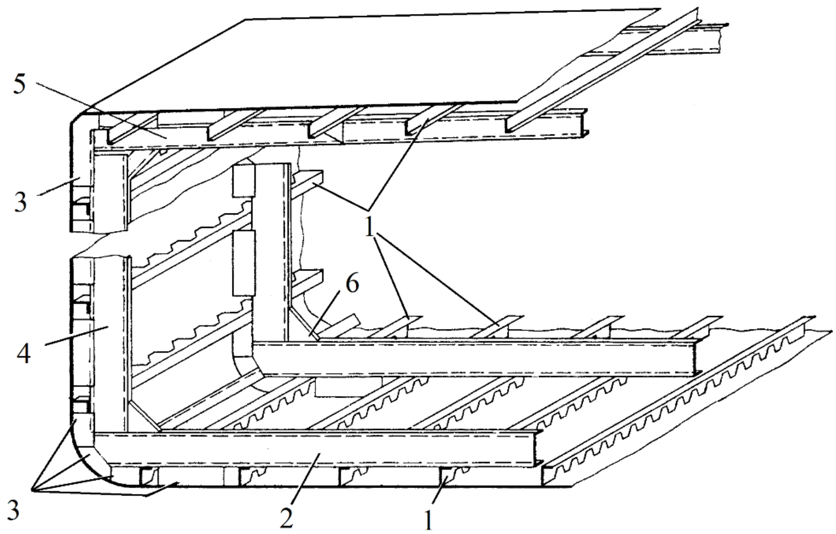

Fig. 64. Construction of a Cistern Vessel (Tank Ship) with the Longitudinal System of Stiffening and Inner Frames from U-Profiles [Authors, 4]

1 - longitudinal standard (light) girders, 2 - inner frame transverse, 3 - brackets connecting inner frame girders with the plating, 4 - frame ring of the side, 5 - frame deck transverse, 6 - bevel

There in tank boats the construction is sometimes designed so that a frame ring leans on a longitudinal system of stiffening from the inner side. In such a construction the transverse, side frame and the beam made from U-profile do not come into contact with the plating, but they touch longitudinal girders. Individual U-profiles are equipped with brackets in the corners. To stiffen and join frames with the plating brackets are welded in gaps between longitudinal stiffeners. Such a construction requires somewhat greater requirements for positioning of a rolled material, however, it brings the following advantages:

- a decrease of operoseness at the manufacture and welding of the hull as a result of using rolled profiles,
- a better cleaning of cargo space in tank ships. Liquid cargo flows better towards collecting containers of pumping devices which are usually positioned near transverse bulkheads.

The longitudinal stiffener of ships stiffened according to the longitudinal system comprises: keelsons, stringers and underdeck girders, and the hull also comprises longitudinal lightweight girders (profiles), i.e. stiffening frames.

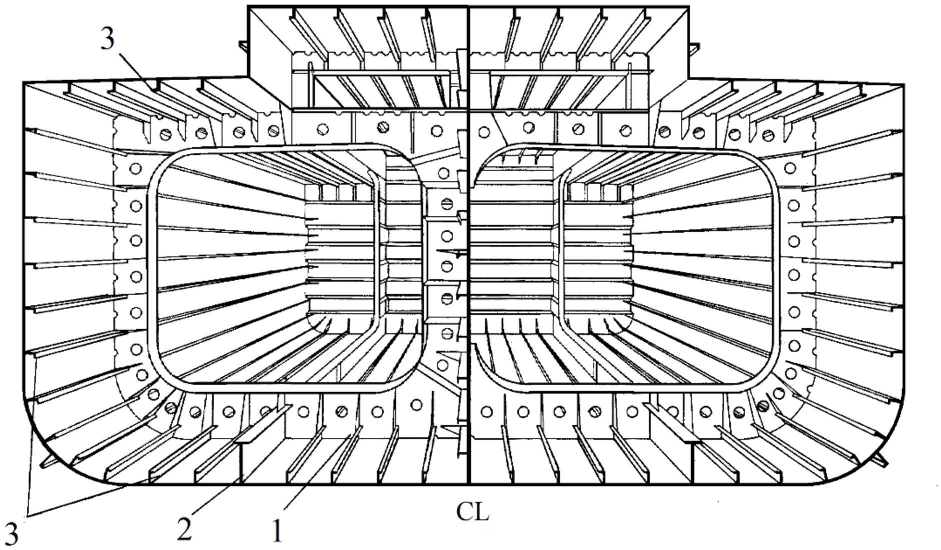

Fig. 495. Construction of a Cistern Vessel (Tank Ship) with the Longitudinal System of Stiffening [Authors, 4]

1 - frame ring, 2 - frame keelson, 3 - longitudinal standard girders on the bottom (lower stringers), on the side (holes) and in the deck (deck stringers)

The distance between girders of the longitudinal direction ranges from 500 mm to 600 mm and underdeck girders and keelsons are placed in the distance of 1.5 m to 2.5 m so it is possible to position 3 to 4 longitudinal stiffening frames between them. Underdeck girders and keelsons are also usually positioned into one vertical plane and they are mutually interconnected with stanchions or bulkhead girders especially in tanks transporting liquid cargo. Mostly they are constructed as discontinued girders which are welded to transverse bulkheads and beams along the entire circumference of their front section. Longitudinal girders are equipped with brackets on both sides of transverse bulkheads so the principle of confinedness of the stiffening structure is adhered to.

Longitudinal stiffening frames do not lean up on transverse bulkheads, they are brought in the distance of 50 mm so water or liquid cargo could freely flow through a suction basket of a draining or loading system. They are mounted to bulkheads using brackets. At the same time, the tolerable abaxiality of longitudinal stiffeners must not be greater than one half of the thickness of the profile's wall.

Standard longitudinal stiffeners often accept water pressure from the plating and they transfer it in the form of a reaction onto frames. At that moment they contribute with their entire surface area to the work of the hull girder by the overall bending and provide the plating with an increased stiffness. Section moduli of individual girders are also determined based on principles of the relevant classification society, and the optimum construction is manufactured on the basis of the method mentioned above.

7.2.8. Pros and Cons of the Longitudinal System of Stiffening

The main and significant advantage of the longitudinal system is the provision of a high stability of the plating which enables ships of the length $L \geq 60$m to decrease the thickness of the plating and thus significantly reduce the overall weight of the framing down to 15 % to 20 %.

Shortcomings of the longitudinal system include:

- a more difficult splicing of sections during the manufacture of the hull as a result of the need for a so called "alignment" of axes of all longitudinal stiffeners;
- the difficulty, or operoseness during bending and welding of girders of the longitudinal direction in the areas where the hull features a curvilinear outline. Due to the given reason as well as due to the local strength the end

(forebody and stern) parts of the ship are manufactured according to the transverse system of stiffening;

- blocking of cargo spaces.

7.2.9. Construction of Ships with a Mixed (Transverse-Longitudinal) System of Stiffening

In plates of the side plating which are located near the neutral axis of the hull girder the stresses by the overall bending of the hull are rather small and the stability of these plates can thus be ensured using the transverse system of stiffening. In ships of a mixed navigation technology and in ships with the length $L \geq 80$m the fender is used to be toughened with discontinuous, light, longitudinal stringers with profiles identical with side frames.

With this association side gratings (manufactured according to the transverse system of stiffening) are much lighter than those ones manufactured according to the longitudinal system of stiffening because the length of cargo spaces significantly exceeds the ship's depth.

As a result of these circumstances it is useful to construct the majority of ships of inland navigation on the basis of a mixed system of stiffening so the sides are stiffened according to the transverse system, and the bottom and mainly the deck according to the longitudinal system of stiffening. There in such a symbiosis arises a mixed system of stiffening where advantages of both transverse and longitudinal systems are appropriately aligned and the majority of shortcomings from the original classic manufactures are removed.

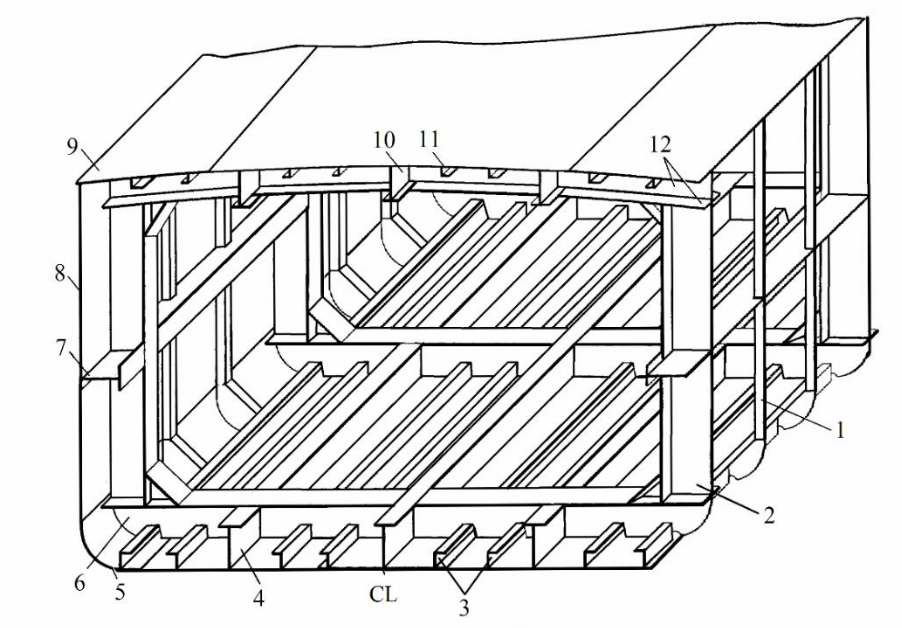

Fig. 506. Construction of the Ship's Hull according to the Mixed System of Stiffening [Authors, 4]

1 - standard frames of the side (only on sides), 2 - frame rings of the side, 3 - bottom standard longitudinal girders (frames), 4 - side frame keelsons, 5 - bilge strake of the plating, 6 - bottom frame transverse, 7 - side frame stringer, 8 - sheer strake, 9 - walkway, 10 - axial frame underdeck girder, 11 - longitudinal standard deck stringers, 12 - frame deck transverse

Such a system was firstly scientifically worked out by an academician J. A. Šimanský and thus this system is more frequently called Šimanský's system.

In the present era the mixed system of stiffening is widely spread, e.g. in motor cargo ships and boats, tank ships, cistern and deck boats, passenger and other ships with the length of $L > 40$m. An exemplary frame of a ship with the mixed system of stiffening can be seen in the following figure.

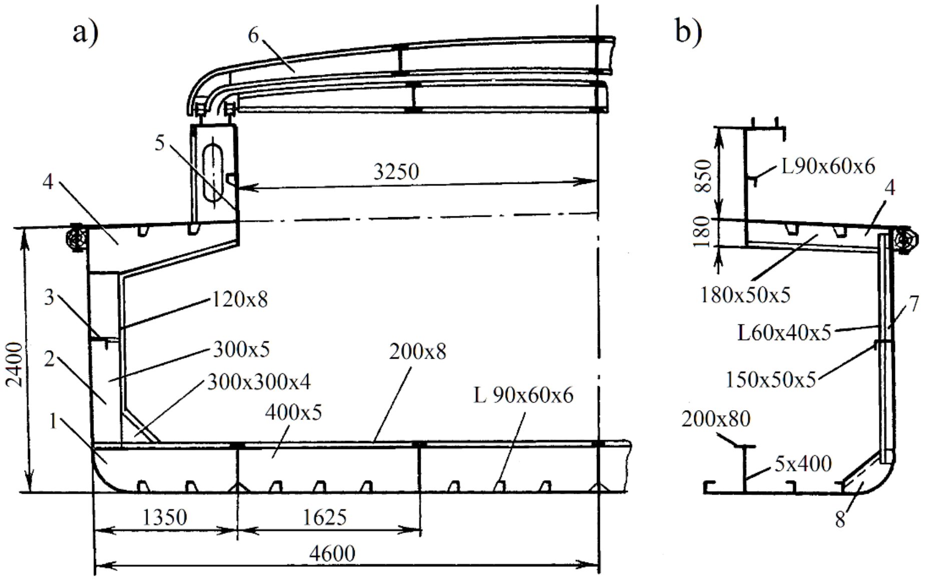

Fig. 517. Main Frame of a Cargo Ship with a Mixed System of the Hull's Stiffening with Loading Covers of a Telescopic Type [Authors, 4]

a) frame ring sections which are set through 2 frame spacings (1200 mm); b) sections through a standard side frame and frame half-transverse of the deck (between frame rings); 1 - frame bottom ring, 2 - frame ring of the side, 3 - side stringer, 4 - frame half-transverse of the deck, 5 - trim of the cargo space, 6 - suspensory cover, 7 - standard frame of the side, 8 - bilge bracket

Longitudinal elements of the bottom and deck grating are constructed in the same way as in case of the longitudinal system of stiffening. There happens an alternation of frame and standard girders, e.g. L-profiles and bulb flats which are in the distance of 500 mm to 600 mm. Longitudinal elements continue up to transverse bulkheads. Frame keelsons and underdeck girders are in the distance of 1500 mm to 2400 mm.

Frame rings are positioned on every second or third frame, i.e. their distance is approx. 1200 mm to 1800 mm. There on transverse bulkheads and beams are cut slots for positioning some standard longitudinal girders.

On the sides there are positioned standard side frames from bulb flats or bulb angles between frame rings in the frame spacing (500 mm to 600 mm). In the bilge and in the weir the side frames are connected with the nearest longitudinal stiffener using brackets.

There exists one more kind of a mixed system of stiffening which is, however, used only rarely. Only the deck is manufactured in the longitudinal system of stiffening; the bottom and sides are constructed according to the transverse system. The examples include various technical vessels (dredgers, elevators, etc.).

7.2.10. Construction of Longitudinal and Transverse Bulkheads

Watertight transverse bulkheads serve to partition the hull off into individual spaces - compartments. From the point of view of the strength, however, these bulkheads represent stiff shores for elements of the hull's framing as a whole, but they do also play an important role in that they contribute to the local as well as to the overall strength and they do also increase the stiffness and resistance of the hull against twisting forces.

The surface area of the plating and bulkheads is welded from sheets with long sides oriented in the longitudinal direction of the ship. Lower sheets are usually 0.5 mm to 1 mm thicker when compared to the upper ones. The reason is that after filling a given compartment with water these elements must tolerate the pressure of water and besides they are exposed to a more intensive corrosion than elements placed in the upper part.

Light vertical girders (frames from bulb flats or L-profiles) are usually welded to sheets of the plating. In the plane of keelsons and underdeck girders there are positioned frame vertical girders - so called bulkhead stiffeners, and at the level of side stringers there are positioned frame horizontal bulkhead stiffeners with a transverse profile which is identical with the stringer in order to adhere to principle of confinedness of frames. The distance between vertical girders is also 500 mm to 600 mm, sometimes even 700 mm.

Rarely we can come across constructions of transverse bulkheads which are stiffened mostly with horizontal stiffeners. For example, transverse stiffeners of ice breakers which must accept the pressure of ice acting on the side of the ship.

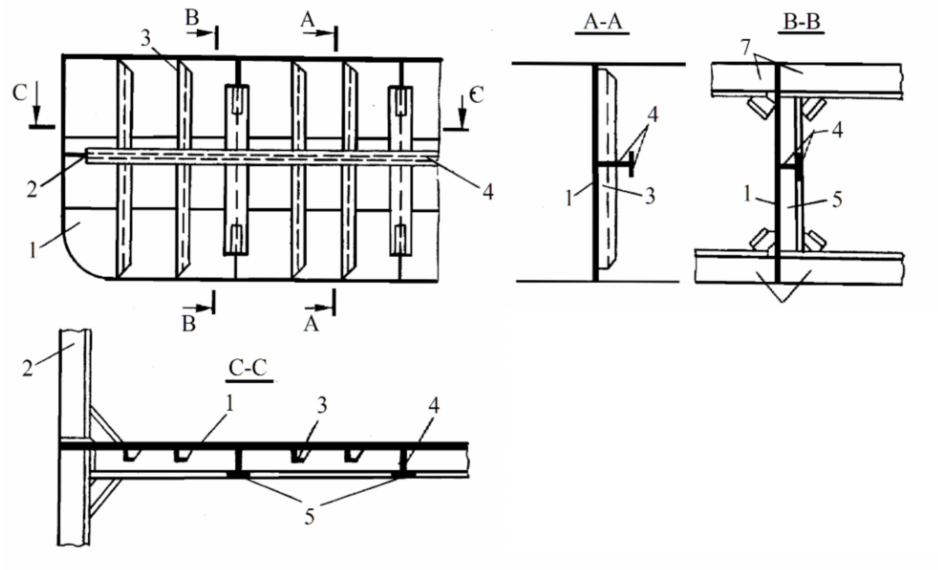

Fig. 68. Transverse Bulkhead with Welded Vertical Frames [Authors, 4]

1 - sheet of a bulkhead, 2 - side stringer, 3 - standard web plates, 4 - frame horizontal bulkhead stiffener, 5 - frame web plates in the plane of the keelson and underdeck girders, 6 - keelson, 7 - underdeck girder

Currently stamping is commonly used for the construction.

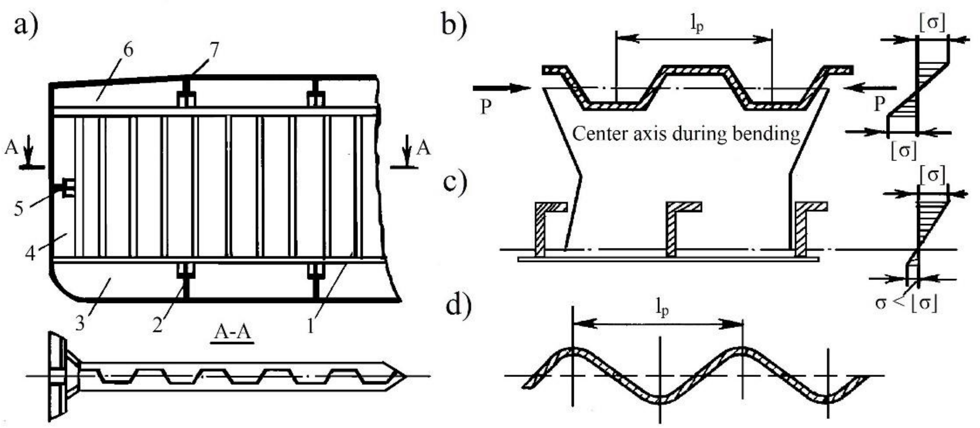

Fig. 69. Bulkheads from Stamping [Authors, 4]

a) overpressed transverse bulkhead welded into a stiff frame ring; b) section of elements of overpressed bulkheads and a diagram of the stress by the bending; c) section of elements of a bulkhead with welded frames and a diagram of the stress by the bending; d) overpresses of sinusoid bulkheads; 1 - sheet of a bulkhead, 2 - keelson, 3 - transverse, 4 - frame ring of the side, 5 - frame stringer, 6 - frame deck transverse, 7 - underdeck girder

The overall weight of overpressed bulkheads is 15 % to 20 % smaller than in case of welded bulkheads of the same strength; it is the result of the fact that the "flange" in the section is symmetrical with regard to the neutral axis of the bending, and thus stresses from both sides of the bulkhead are equal, too. For bulkheads with welded stiffeners, metal (or, in other words, the mass of the section) is not burdened from the bulkhead's side, and it is overburdened from the flange's side. The reduction of the overpressed bulkheads' weight is also achieved with not increasing the thickness of the sheet (which is inevitable in case of thin sheets so they are not deformed during the welding).

The operoseness of manufacturing overpressed bulkheads is significantly decreased in 35 % due to elimination of welding works, but mainly due to elimination of a so called "straightening" after welding. Moreover painting works and cleaning are made simpler during the change from one substrate into another, which is especially important in case of tank ships.

Overpresses are usually positioned vertically. Most frequently they feature a trapezoid or wavy profile. In case of stampings of a transverse bulkhead they are mostly mounted to a flange of a reinforced frame ring which significantly simplifies the technology of splicing and welding.

Both longitudinal and transverse bulkheads are manufactured either with a welded stiffener, or from stampings. On longitudinal bulkheads the stiffeners or overpresses are mostly horizontal ones so such a bulkhead can contribute to the overall strength of the ship.

The number of transverse and longitudinal bulkheads is recommended in principles of the respective classification society. There is, however, still a prerequisite that all ships must be equipped with a forebody and stern collision bulkhead. Positioning of doors or slots there on these bulkheads is generally not allowed in any case.

Motor cargo ships must have at least three transverse watertight bulkheads at disposal including the forebody and stern collision bulkhead. This condition is valid for ships of the length $L = 20\ m - 60\ m$. In ships of the length $L \geq 100\ m$ there must be at least six transverse watertight bulkheads. Tank ships of the length $L \leq 80m$ must also have one longitudinal bulkhead; when the length is $L > 80m$ there must be two such bulkheads. The rule is that there in tank ships transverse bulkheads are positioned in every 24th frame spacing when the side's depth is $H \leq 2{,}5m$, and in every 36th frame spacing when the side's depth is $H > 2{,}5m$.

7.3. REINFORCEMENT OF THE HULL IN THE FOREBODY AND STERN IN THE ENGINE ROOM - BASES OF MAIN ENGINES

Forebody and stern part of the ship's hull up to the distance $0{,}15L$from the forebody as well as stern perpendicular needs to be reinforced. The stern part of the engine cargo ship means a compartment between a stern perpendicular up to the bulkhead of the engine room, or up to the distance $0{,}15L$from the stern perpendicular, depending on which distance is shorter. The forebody part is exposed to hits of waves and there is also the highest probability of striking a shoal; moreover it is the most strained part during the navigation in drift ice. The stern part of the ship is loaded with periodically changing water pressure by the propeller which may evoke vibration of the structure and subsequently the noise, too.

In accordance with the character of acting forces as well as from the point of view of technology the end parts of the ship are constructed according to the transverse system independently from the way the central part of the ship is constructed.

In the forepeak and afterpeak the frame transverses are located on each frame. Frame spacing in these edge sections should not be greater than 550 mm. If a ship is intended for navigation in drift ice, then the maximum frame spacing will be 400 mm. If, however, the frame spacing remained the same as in the central part of the ship, then it would be necessary to install so called "auxiliary frames". Walls of side frames are required to be welded so they are perpendicular to the plating, if possible.

Side stringers are mounted to a "stem" using horizontal brackets.

Keelsons and underdeck girders should interfere into the rear and front collision (peak) space as much as possible, whereas there in the axis plane they should be mandatorily fixed to both the sternpost and the stem.

The stem is mostly manufactured from strip steel and its lower part runs into a garboard plate step by step. In case of ships with a transom form of the bow the stem is formed as an inner skew girder representing a continuation of the bottom keelson which is attached to a central axial underdeck girder using a bracket near the deck.

The construction of the sternpost is determined with the form of the stern. In one-screw ships this is usually a frame intended for the protection of the propeller, for the support of the end of the shafting and lift of the rudder. For ships with a spoon and transom stern the role of the stern is to form a continuation of the centre keelson which connects to a central underdeck girder there in the upper part in the area of the deck. A so called shaft tube serves for the transit of shafts through the plating of the ship's hull. The frontal part of the tube is fixed using screws to a reinforced afterpeak bulkhead, whereas the rear part passes through the plating and is fixed using a special flange (bracket) and a reinforced sheet of the plating.

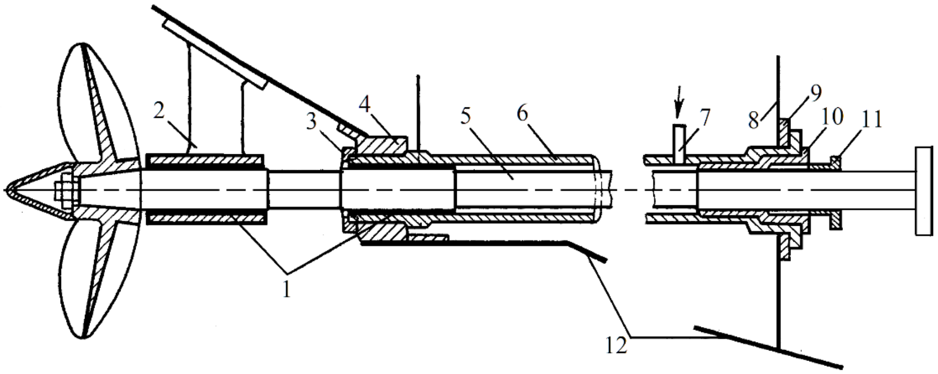

Fig. 70. Construction of a Shaft Tube [Authors, 4]

1 - rubber spring shackle in a metal sleeve, 2 - propeller shaft bracket, 3 - nut, 4 - transitional spacer, 5 - propeller shaft, 6 - shaft tube, 7 - clean water intake, 8 - transverse bulkhead, 9 - reinforced cleat, 10 - sealing caulking, 11 - pressure hub of a caulking, 12 - plating

Inside the shaft tube and the bracket there are positioned bearings usually with rubber spacers which are more resistant against fender and may stand even multiple sailing seasons without replacement. In the frontal part of the shaft tube there is a caulking to prevent seeping water intrusion into the ship's hull.

Main and auxiliary engines as well as other operational machines create very strong static and dynamic forces which must be transferred to the framing and plating of the hull via their foundations. These forces may evoke a strong vibration of individual girders of the construction, plating sheets and even the ship's hull as a whole which leads to fatigue cracks in individual elements and creates inconvenient conditions for the navigation as well as the environment in the ship itself.

A higher temperature in the engine room, a constant presence of water and aggressive liquids (residues of fuel, oil, ash, etc.) below the floor cause an intensive corrosion of elements of the hull and the plating. That is why girders as well as the hull's plating in the area of the engine room must feature a greater thickness.

In compliance with principles of classification societies transverses in the area of the engine room should be positioned on each frame, whereby there in the area of foundations below main engines the thickness of transverses walls should be 1 mm greater and the sections of flanges up to $1\frac{1}{2}$ - times of calculation values greater.

Furthermore the count and magnitude of keelsons increases; their positions must be aligned with the position of machine foundations.

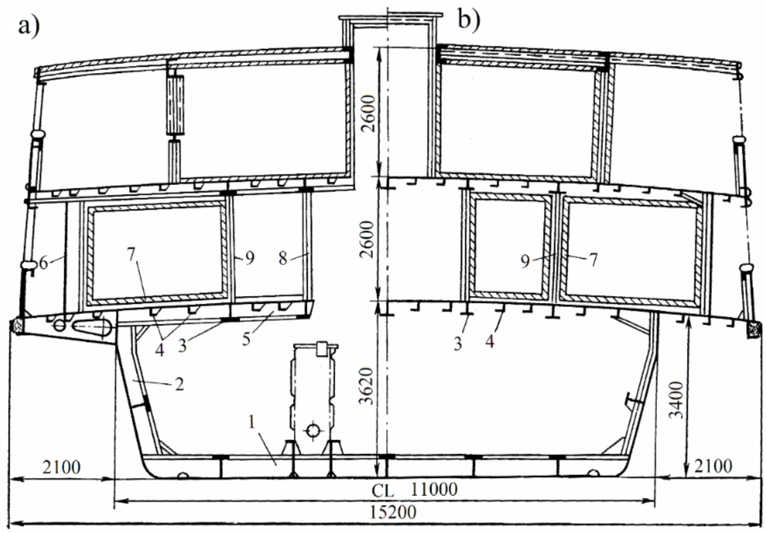

Fig. 71. Transverse Sections of the Hull and Superstructures in a Passenger Ship [Authors, 4]

a) - section through a frame ring within the area of the engine room; b) section through a standard frame in central part of the ship; 1 - frame transverse, 2 - frame ring of the side, 3 - frame underdeck girder, 4 - standard deck stringer, 5 - frame deck transverse, 6 - frame web plate, 7 - inner lining, 8 - standard (light) web plate, 9 - stanchion

In order to eliminate vibrations a reinforcement of a mutual binding of girders is manufactured by means of positioning some supplemental longitudinal frames between transverses. At the same time for the same sake it is necessary to position a propeller into a greater distance from the hull and unbalanced mechanisms onto so called "shock-absorbers".

Foundations are manufactured from continuous asymmetric T-profiles with an upper horizontal flange with the thickness of 8 mm to 20 mm. The distance between foundation girders, their length, height and other dimensions are determined based on the dimensions of supporting designs of engines or other mechanisms which are also positioned on these foundations.

Reinforced keelsons are a continuation of foundations and are brought into the transverse bulkhead. They are welded to it with brackets from both sides of the bulkhead. In the plane of the bulkhead's brackets they are strengthened with frame web plates.

Flanges between foundations of main engines are manufactured from girders cut into pieces which are welded to walls of the foundation and strengthened with special brackets from both sides.

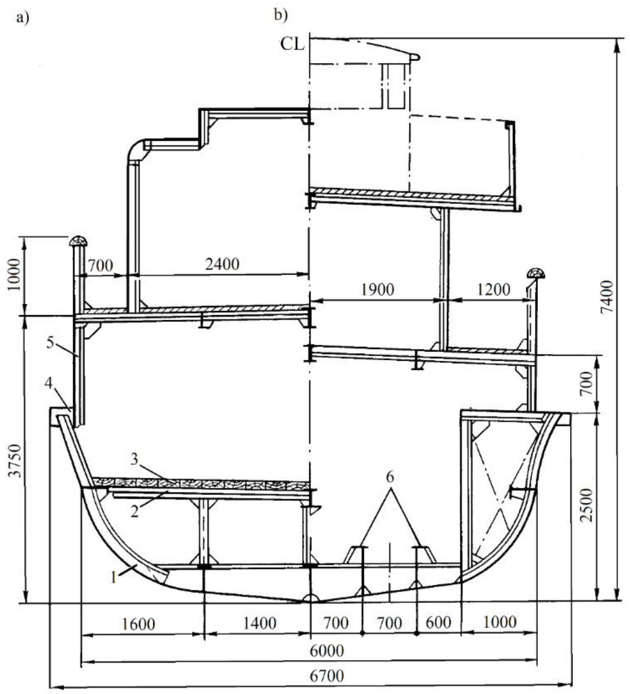

Fig. 72. Transverse Sections of the Hull and Superstructure in a Passenger Ship for Local Lines [Authors, 4]

a) section in the area of a superstructure; b) section in the area of main engines; 1 - frame ring bound to a bottom transverse, 2 - platform, 3 - wooden floor, 4 - sideway, 5 - light web plate, 6 - foundations of main engines

7.4. FIXED FITTING, CONSTRUCTION OF SUPERSTRUCTURES, AND OTHER BUILT-IN SPACES OF A SHIP

The term fixed fitting means side and deck windows, metal doors, covers of skylights and entrance openings, structure of hatchways, metal ladders and their handles, gratings of rails, etc.

Windows, whether round or angular, mostly serve to light up rooms inside the ship. They can be openable or fixed.

Openings, such as hatchways, are closed with hermetic covers, sealed with a rubber seal. The sealing can be completed with some screws or hinges with butterfly tightening nuts.

To prevent people from falling down of the deck into water all open parts of decks are fenced with bulwarks and rails. Bulwarks are stiffened with vertical frames and for the drain of rain water there are formed some passages of storm water, where the greater ones are equipped with a grid or removable shields on hinges. The surface area of passages of storm water must not be less than 10 % of the surface area of the bulwark.

Rails are usually produced from tubular stanchions with the height of 750 mm to 1200 mm fixed to the deck in the spacing of 1200 mm to 1800 mm either using the welding or special flanges with welded screws. There between stanchions in the distance of 200 mm to 400 mm are placed stanchion rods, steel ropes, chains and other elements. In passenger ships the rail is manufactured in a more aesthetic way with elements of wood or other suitable materials.

Superstructures can be divided into fixed (contributing to the overall strength of a ship) and light ones by their design manufacture. A superstructure is considered fixed or strength if it leans against at least 3 transverse bulkheads of the hull and its length is at least as long as 6-times of its height. So called light superstructures do not contribute to the overall strength or bending of the ship, therefore it is required to weaken their bind with the ship's hull as much as possible. With regard to their length it is necessary to separate its individual parts with a so called dilatation which excludes a common deformation of superstructures and the ship's hull.

By the used materials superstructures can be divided into metal, laminated, composite and seldom wooden ones.

The structure of walls and decks of fixed metal superstructures is analogical to the structure of the hull and decks. Outer and inner walls of a superstructure are usually stiffened according to the transverse system of stiffening and the framing of the superstructure comprises frame and standard vertical frames welded to the plating.

It is useful to manufacture the walls of superstructures from stampings with a horizontal overpress.

The framing of the deck of fixed superstructures must match the system of stiffening of the deck and bulkheads of the hull where the superstructure in the given section is positioned.

Superstructures manufactured from light aluminium alloys and laminates feature the following advantages:

- reducing the overall weight of the ship,
- lowering the position of the ship's centre of gravity and thus increasing the stability,
- reducing operational costs of anti-corrosion protection.

The superstructure from aluminium alloys must, however, be thoroughly insulated from the steel hull using some special flanges or claddings so there is formed an insulation bridge against the intensive electro-mechanical corrosion which happens upon the contact of these two materials.

Built-in spaces and facing of spaces comprise the insulation of the surface area, reveal (frames below cladding), lining of walls and deck covering.

Thermal insulation and sound proofing is usually attached to clean walls of the superstructure which are protected with a prime coat.

The insulation is either panelled or in the form of mineral wool; in the past crushed cork was mostly used. Nowadays a composite insulation with a thin aluminium foil on its surface is applied. Insulation materials must protect spaces from temperature fluctuation, moisture as well as sound waves or noise from the surroundings. They must be light, firm, non-absorbing and wholesome.

Reveal is in fact a system of small vertical and horizontal beams, often oriented diagonally, which serves for the clamping of lining panels. Interroom and corridor walls have an upper and lower beam in the reveal which serve to fix other elements, such as other panels of walls, ceilings and floors. All wooden beams must be impregnated against fire. Beams are attached to the hull's framing or to the plating using welded nails and screws or using other structure methods. Today there are rather some light aluminium profiles used for the reveal. Wooden reveals do not satisfy stricter fire requirements any more, and thus they are used only partially.

Lining of walls and ceilings is manufactured in specialised sheds in the form of removable panels which are later installed in a ship. These panels are attached with the insulation to, or they are directly manufactured as layered composite structures (sandwich structures).

Coatings of walls and ceilings are used only rarely nowadays and mostly their surface is covered with a wallpaper or melamine furniture layer, etc. All surface interior materials must be constant from the point of view of changing temperature and humidity, they must be well and easily washable and they must have an aesthetic colour finish.

Lining of restaurants, dining rooms, corridors, halls, rooms for passengers and the crew must create an aesthetically harmonised and convenient environment. Choice of colours, especially the outward ones, is often given with principles of the respective sailing company, but also with exerted traditions.

After a stainless coating there are put various cement materials on the floor. Linoleum or other floor coverings which used to be put on a wooden surface are used only rarely nowadays. In social (sanitary-hygienic) spaces the floors and walls are covered with ceramic tiling. Floor coverings must be wear resistant and non-slippy.

In compliance with safety and sanitary regulations of the respective classification society all interior lining materials must be fireproof. Self-extinguishing material, or material which prevents the spread of fire from an emerged fire bed can be used only exceptionally. It is not allowed to use any coatings of a flammable basis, and during burning and smouldering no explosive gases in dangerous concentrations can be released.

In saloons, restaurants and other social rooms there is positioned light furniture harmonised in terms of colour; its structural skeleton is formed with various aluminium alloys, plastics, laminates, etc.

In cargo spaces of engine cargo ships there are still used wooden removable panels with the thickness of 50 mm to 70 mm; they cover the floors and often the sides of cargo spaces, too. This is the case e.g. in ships transferring grain, etc.

The most challenging problem is the protection of metal against corrosion in ships transferring some especially aggressive goods, such as salt, chemicals, etc. To protect such surfaces some lining coverings on the basis of epoxy resin, elastic synthetic coverings, and other materials including those on the basis of teflon, are used more and more frequently.

7.5. SPECIALTIES OF CONSTRUCTING STEEL SHIPS OF NEW PROGRESSIVE TYPES

7.5.1. Passenger Ships

The passenger ships construction is characterised with the mixed system of stiffening. Frame rings are usually positioned in every second frame spacing. The main deck and the deck of a strength, often first superstructure are stiffened according to the longitudinal system. The bottom, sides and walls of a superstructure are stiffened transversely. For bigger ships the bottom features the longitudinal system of stiffening. The main deck has a so called "camber" only in the area of side walkways; in its central part it is flat in order to achieve a useful height of living spaces.

7.5.2. Cargo Ships

The following requirements are put on universal cargo ships and boats:

1. In order to ensure a mechanised manipulation with cargo the cargo spaces must be completely open without any transverse or longitudinal bulkheads. This requirement results in a structure of double sides and bottom. As it

has already been mentioned, such a structure enables to provide unsinkability of a ship, to balance heelings and inclinations, and to gain optimum trim setting in order to achieve the highest possible speed and steerability of a ship. There is the disadvantage of reducing the useful capacity as a result of the recommended frame spacing which should be less than 800 mm.

2. Cargo spaces cannot have any projecting parts, such as brackets, so it is possible to clean the cargo space with mechanisms to the nine.
3. The overall strength of the hull must be sufficient so it is possible to load and unload the cargo in one layer with a maximum unevenness of distributing the cargo longwise the space. The reinforcement of the second bottom and sides enables to use large-capacity grabs.
4. Covers of cargo spaces must feature a mechanised closing and opening while there is a gap of 50 % to 70 % of the cargo space area.

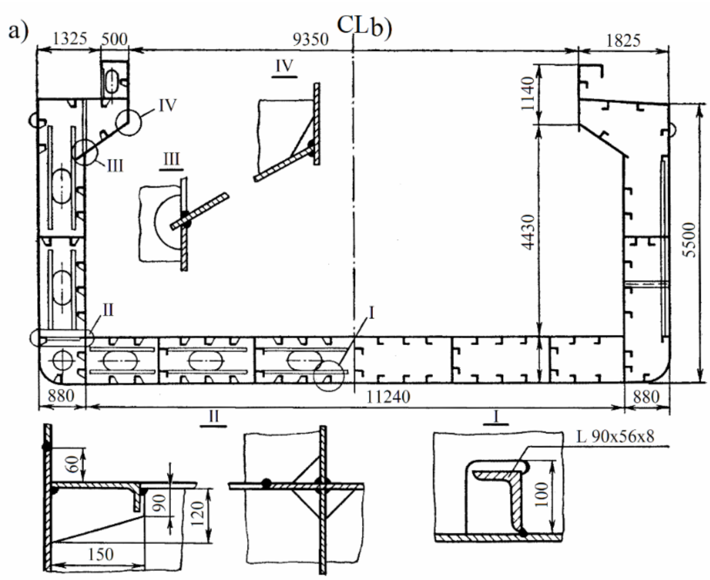

Fig. 73. Scheme of a Main Frame of a Cargo Ship of the Class "M-sp" with a Double Bottom and Double Sides [Authors, 4]

a) section through a frame ring; b) section through a standard frame

Cargo bulkheadless ships have an inner double bottom constructed according to the transverse system of stiffening while frame rings are in every third frame spacing and there are placed frames with a lightweight structure between them.

In side rooms of the bottom there are also frame transverses in every third frame spacing as if they were a continuation of interbottom frame transverses. To reduce the concentration of stresses the transverses are mounted to the plating of inboard sides using horizontal brackets.

Outer and inboard sides are favourable to be constructed according the transverse system of stiffening, too.

Since the deck and the bottom of side compartments provide the overall strength, they are constructed according to the longitudinal system of stiffening. The upper stringer, coaming, awl and fender of these ships are made of Cor-Ten steel.

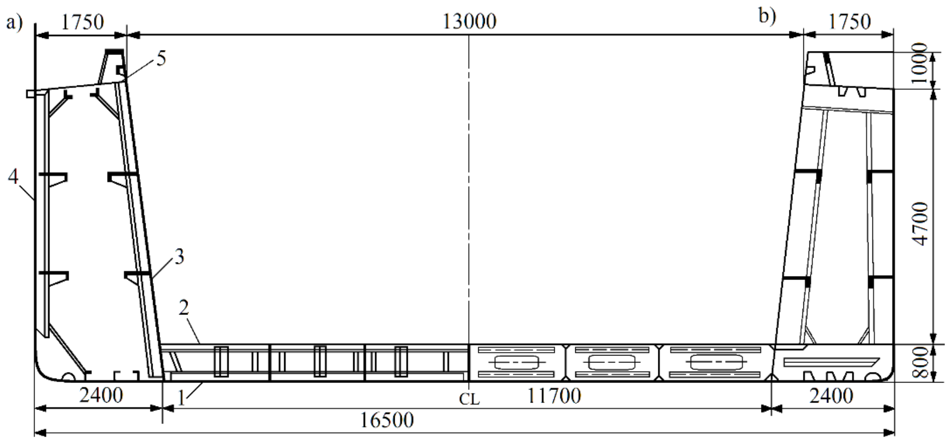

Fig. 74. Scheme of a Main Frame of a Cargo Ship of the Hopper Type with a Double Bottom Stiffened according to the Transverse System [Authors, 4]

a) section through a standard frame; b) section through a frame ring; 1 - plating of the bottom, 2 - floor of the second bottom, 3 - plating of the inboard side, 4 - outer side, 5 - moulding of cargo space

7.5.3. Tank and Cistern Ships

The most suitable constructions for current tank ships seem to be the constructions which feature a box system with a double bottom and double sides, and a longitudinal bulkhead in the axis plane of the ship.

The inner bottom is inclined to the axis plane and there is also positioned a longitudinal trough, enabling to unload almost entire cargo.

Heavy oil products (mazut) are required to be preheated before the unloading starts in order for them to enter a fluid state. The presence of the double bottom and sides enables to eliminate cooling of the transferred cargo, and thus to eliminate costs for the subsequent preheat during the unloading. A greater effect may be achieved with the additional insulation.

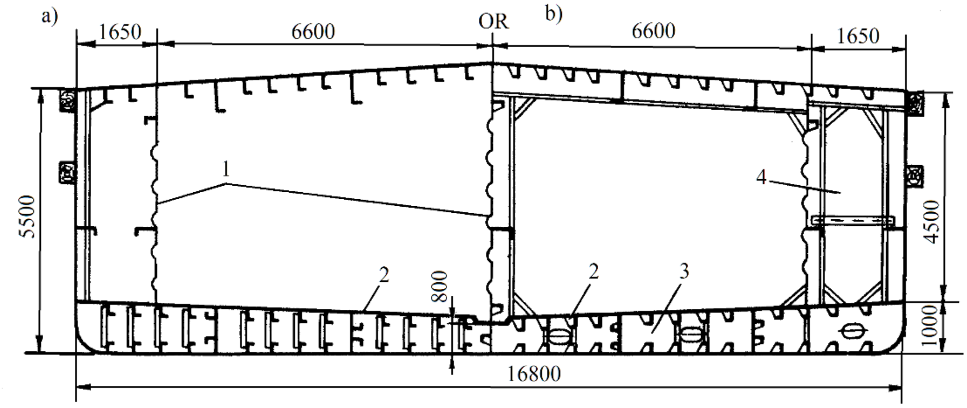

Fig. 75. Transverse Section of a Current Tank Ship of Universal Determination with the Hull Stiffened according to the Mixed System [Authors, 4]

a) section through a standard frame; b) section through a frame ring; 1 - overpressed bulkhead, 2 - floor of the second bottom, 3 - interbottom space of the bottom, 4 - interside space

The interbottom space of tank ships is also very useful for the purpose of ballast of so called "empty" voyages. To increase the fire safety the space of the double bottom and sides is filled with exhaust gases or other inert gases.

In constructions of tank ships some overpressed structures are successfully used there, mainly for bulkheads. The structure of tank ships usually features the deck and bottom manufactured according to the longitudinal system of stiffening, and the sides according to the transverse system.

From the point of view of the strength pusher tank boats are not necessarily to be constructed with a double bottom and sides. Mostly they are constructed according to the longitudinal system of stiffening with a so called "imbedded" frame ring (see the longitudinal system of stiffening).

7.5.4. Ships of Special Determination

To transport light oil products, fluid and powder chemical products there are used specialised engine cargo ships and boats with embedded vertical and horizontal steel tanks. It could be promising to use built-in spaces made of synthetic materials or embedded containers for this purpose.

In certain lines it is suitable to use combined ships: to transport liquid cargo in one direction and dry cargo in the opposite direction. The examples may include a "tank ship - metal-ferry", "tank ship - container-ferry", etc.

Ships of a special determination include catamarans. The presence of two narrow hulls and a joining element leads to a greater overall bending moment of the ship which is 2.5 to 3.5-times greater when compared to 1-hull ships (the bending moment acts so the hull is bent in its central part upwards - bending).

The bending moment per the approximate formula ($M_{\mathrm{tv}} = K.P.L$) has a coefficient $K = 0.056$; a ship of a catamaran type must be checked for the transverse bending and connection bridge - joining element, and the torsion, too. Both hulls are constructed according to the mixed system of stiffening. Since any of the hull is utilised as cargo space, they are often divided with transverse bulkheads and there in the axis plane it is even possible to use stanchions.

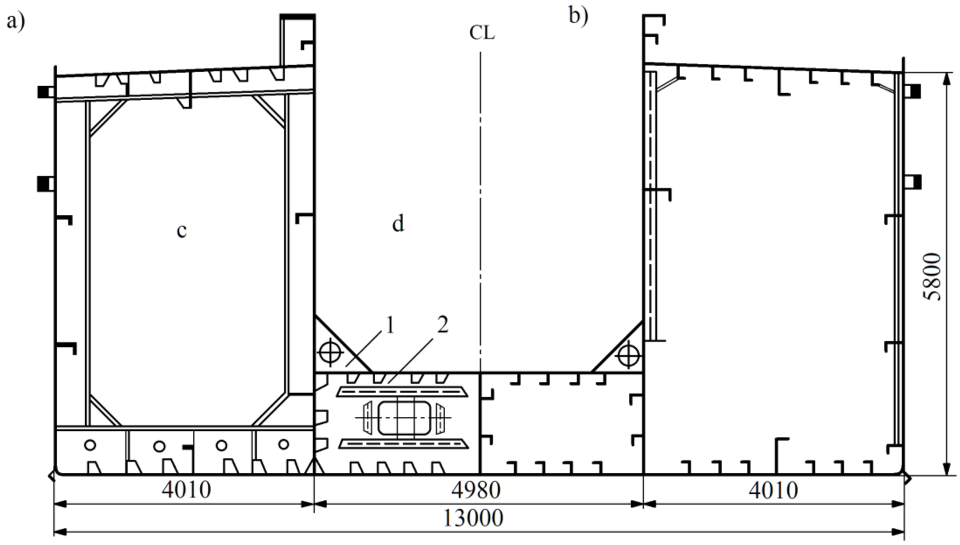

Fig. 76. Scheme of a Main Frame of a Tank Ship - Metal-Ferry [Authors, 4]

a) section through a frame ring; b) section through a standard frame; c) tank ships for liquid cargo; d) space for dry cargo; 1 - brackets in the space for dry cargo, 2 - ballast spaces in the second bottom

7.5.5. Pusher tugs

The hull of pusher tugs is constructed according to the transverse system. There are continuous transverses on the bottom and alternating standard frames and frame rings on the side and the deck. The forebody part of the hull is reinforced with longitudinal bulkheads positioned in the plane of forebody shores. These half-bulkheads in the area behind the forepeak change into longitudinal frame girders of the deck and the bottom.

7.5.6. Push boats

Push boats with cargo space inside of the hull are either of a box classic manufacture, or a trough hull with double sides and double bottom. The interbottom and interside space is divided with watertight transverse half-bulkheads and watertight transverses, and sometimes with an impermeable centre keelson.

Recently there has widely been used another type of non-self-propelled boats, so called deck barges. The hull of such boats has a greater number of longitudinal and transverse bulkhead girders as well as stanchions in order to provide the cooperation of the deck and bottom grating. Despite many construction advantages the deck barges do also have the disadvantage during the manipulation with cargo, and there also exists the disadvantage of a reduced visibility from a pusher tug.

7.6. CONSTRUCTION OF SHIPS FROM ALUMINIUM ALLOYS

Despite many adverse properties, such as a high price of the material, reduced fatigue strength and a 3-times smaller modulus of strength when compared to steel, a more complex technology of welding and treatment, etc., the aluminium alloys are irreplaceable for hulls of the ships which must be light in order to provide reliable, navigational and operation-technical properties.

The category of such ships includes ships with wings, so called hydroflights, air-cushion boats and so called hydroplanes with an extremely small draught. The hull of the ship with underwater wings is similar to a girder which is supported with two shores. The hull is exposed to the acting of significant bending moments which increase due to inertial forces while navigating in waves.

The hull of the ship with wings and their superstructures are constructed according to the mixed system of stiffening with frequent longitudinal standard stiffeners with the spacing of 200 mm to 300 mm, either of T-profile in case of a welded construction, or I-profile in case of a riveted construction. On the surface from the inner side of the hull these longitudinal girders lean against frames (with the spacing of 500 mm to 600 mm in case of a T-profile) as well as against longitudinal frame girders.

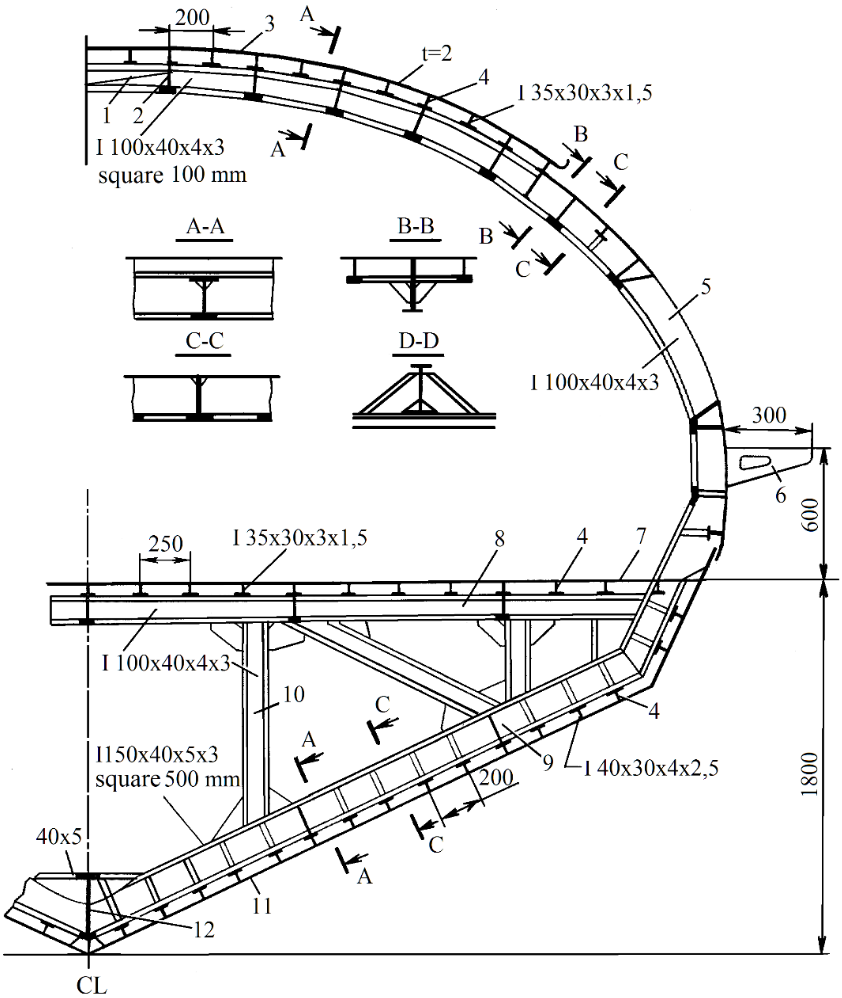

Fig. 527. Main Frame of a Hydroflight Ship of a Welded Construction of Aluminium-Magnesium Alloy (Longitudinal System with Imbedded Frames) [Authors, 4]

1 - underdeck transverse, 2 - underdeck girder, 3 - plating, 4 - underdeck stiffener (frame), 5 - window, 6 - walkway (pathway), 7 - deck covering, 8 - transverse of the deck, 9 - bottom transverse, 10 - stanchion, 11 - plating of the bottom, 12 - central keel

Air-cushion boats and other fast ships such as hydroplanes, etc., are constructed similarly.

The construction of ships of aluminium alloys is not conditioned with principles of the classification society, but in countries where specialised companies aimed at production of such a ship type are located, the ships are constructed according to recommendations of respective classification societies. Depending on the hull of the ship and its determination either the longitudinal, or mixed system of stiffening is recommended.

Besides the construction of fast ships it is possible to successfully use aluminium constructions even for small tank ships where besides the dead weight of the construction itself also the property of aluminium alloy is favourable - it is more resistant against corrosion, particularly the chemical one (when compared to steel hulls) which damages inner surfaces of steel hulls.

7.7. MEANS AND WAYS TO FIGHT AGAINST NOISE IN SHIP STRUCTURES

Noise includes various sounds which adversely impact on the human health. Noise reduces the productivity of work in approx. 20 %, it weakens the memory and attention. From the physical point of view noise is a mixture of various waves with different frequencies and amplitudes which spread in a substantial continuum - air (air noise), in structures of big dimensions (structural noise) and in structures of relatively small dimensions (sound vibration).

Sound wave is characterised with the intensity of sound L $[{}^{W}/_{m^2}]$, sound pressure p_2 $[Pa]$ and sound pressure level L_p $[db]$, where $1[db] = 2.10^{-5}$ $[Pa]$.

Main and auxiliary engines, propellers, fans, equipment of ship systems and other devices represent the most frequent sources of noise in a ship. The greatest noisiness is generated with powerful high-speed engines, e.g. the engine $N = 220$ $[kW]$, $n = 1500$ [rpm] has a medium sound level of $110db$, the engine $N = 650$ $[kW]$, $n = 1600$ [rpm] has the sound level of *118 dB*, and low-speed engines $n \lhd 300$ [rpm] have the sound level of $L_p = (90 \div 95)$ $[db]$.

Rooms where the sources of noise are located (engine room, afterpeak) are called "noisy", those in their immediate vicinity are called "contact" and those which are not directly adjacent to a noisy room are called "remote".

The noise can pervade contact and remote rooms directly from noisy rooms through open doors, windows, hatchways, passages and partially through bulkheads and deck compartments and it can evoke sound vibration and structural noise there. The main source of structural noise, which spreads along the structure from its source, are the foundations, piping, bearings which carry the noise into bulkheads, decks, platforms delimiting contact and remote rooms where it evokes sound vibration; it is then the source of air noise in given rooms.

Tolerable levels of sound pressure in living and service spaces as well as in engine rooms are set in health normatives. For example there in passenger ships the tolerable sound levels in cabins are $50\ [db]$ and in engine cargo ships and tugs they range $(60 \div 65)\ [db]$. In engine rooms where operators are constantly present the average value of noise is set to the level $85\ [db]$ and in engine rooms with a periodic operation the level comes up to $90\ [db]$.

Main methods to fight against noise:

- To reduce the noisiness of sources (engines).
- To separate and isolate living and service spaces from engine rooms, skylights of engine rooms, exhaust piping and propellers (cofferdam, etc.). An appropriate design of rooms significantly simplifies the fight against noise.
- To use special sound-absorbing and sound-insulating structures as well as structures which increase acoustic resistance and spreading of structural noise to a large extent.

Examples of Noise Barrier Structures:

Sound-absorbing structures serve to absorb air noise. They act based on the transformation of sound waves into thermal energy. Usually they comprise one or two layers of porous material (plates of rockwool, a mattress of PVC fibres or fibre-glass approx. 50 mm thick). For this purpose the noise barrier insulation in engine rooms is manufactured using a so called loggerhead method; from the outer side the porous material is protected with a perforated aluminium, steel or another material. In cabins and less noisy rooms a perforated plywood, fibreboard, etc., is used.

To fight against noise there are also used one- and two-side sound-insulating structures. The best sound-insulating properties are characteristic for heavy materials (lead, thick steel or aluminium plates, tempered plastics, glass, etc.) which have a high acoustic resistance when compared to air.

Two-sided structures comprise either two steel walls (often the other side is made of another material than steel) with an air gap between them (a cofferdam) with the breadth of 150 mm to 500 mm. It is useful to fill the gap between walls with a sound-absorbing material or material absorbing vibration (however, always from the side of a less noisy wall). As a result of such a structure there arises a combined sound-insulating and sound-absorbing structure. Steel walls must be mutually joined, or split with sound-insulating flexible bridges (joints) which prevent the transition of structural noise from one wall to another one.

Sound-insulating structural elements are used to manufacture so called "bonnets" - rendering of noisy mechanisms, rooms - niches for positioning some auxiliary noisy engines, sound-insulating cabins - places in the engine room, sound-insulating walls, e.g. to delimit the space intended for mechanism control, etc.

For one-wall structures the sound-insulating capability achieves *30 dB* to *50 dB* and for two-wall structures it achieves *35 dB* to *75 dB* (at correspondingly low and high frequencies).

In order to eliminate spreading of the structural noise the biggest effect is achieved when the engine and gearbox are mounted flexibly on silentblocks which are usually made of composite rubber-metal materials, or on the basis of springs. The first type of silentblocks eliminates structural noise in the foundations to *10 dB* to *20 dB* and the spring silentblocks to *20 dB* to *30 dB*; the spring ones are more suitable for low frequencies. The linkage of the drive unit and shafting is desirable to be done using a flexible clutch. Spreading of structural noise from the engine room can be restricted to a certain extent using corner strips, echoing the vibration. Part of the plating of the bottom, side and deck, which is adjacent to a bulkhead, is usually concreted or filled with another suitable material. Such a strip eliminates the level of structural noise out of the engine room in *4 dB* to *8 dB.*

There are also used vibration-absorbing coverings "antivibrators" which reduce sound vibrations. Vibration-absorbing materials are applied on segments of the grating which vibrates strongly, particularly near the oscillation generator (a segment of the bottom above propellers, compartments of the deck and in the vicinity of the engine room). This coating transforms part of the waving into thermal energy and scatters it partially. Vibration-absorbing coating is often adhered to the steel plating, which comprises stiff plastics, or it is directly applied on the plating, e.g. bitumen fibres and other materials which are then protected e.g. with linoleum, boards, etc., from the upper side, or a flexible coating in the form of latex rubber coatings (suntex, etc.) is applied.

Floating structures are usually walls of the floor or all walls of the room ("floating lining of the room") which are attached to the supporting structure only by means of flexible shores. Such "floating cabins" are mostly located directly above noisy rooms.

The information above implies that in small ships equipped with high-speed engines the biggest noisiness is manifested in rooms located near the engine room. A particularly important task is the fight against noise in fast ships with hydro-wings, vessels on air cushion, etc.

To ensure a so called noise barrier complex of ships there exist special normative recommendations and instructions. The weight of a noise barrier complex in small ships represents approx. 2.5 % to 10 % of the weight of an empty ship. A logical implication is that such a ship will be of a higher price.

A standard noise barrier complex of a river ship comprises parts represented in Fig. 78.

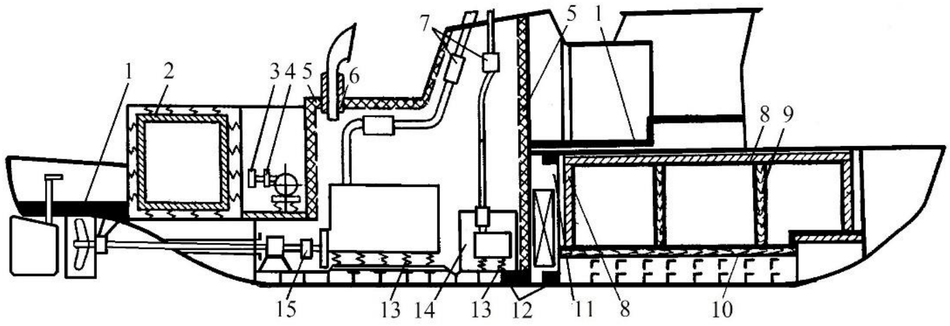

Fig. 78. Complex of Noise Barrier Structures of a Tug [Authors, 4]

1 - vibration-absorbing coating, 2 - "floating" structure of a stern cabin, 3 - noise absorber of a ventilator, 4 - elastic cuff, 5 - sound-insulating structure with a perforated tiling, 6 - sound-absorbing structure of a ventilation piping, 7 - exhaust silencer of the engine, 8 - sound-absorbing monolithic structure, 9 - wooden two-layer bulkhead of cabins, 10 - two-layer grating of the floor coating, 11 - sound-insulating cofferdam, 12 - concrete corner vibration-absorbing silentblocks, 14 - sound-insulating bonnet of noisy auxiliary engines, 15 - flexible clutch of the shafting

7.8. TECHNOLOGY OF METAL SHIP BUILDING

Building of metal ships is a complex process comprising several stages, which also other partnering enterprises take part in besides shipyards. Main tasks of the technology lie in improving the quality, shortening the due dates of the building and reducing the price of the ship. In order to achieve these objectives the current technology and organisation must come out of the following basic principles:

- the specialisation of the manufacture, i.e. a minimum number of ship types,
- the serial production, i.e. to build the highest possible number of ships according to one design using the same appliances and technological equipment,
- the unification of ship products, i.e. the utilisation of standard products and facilities in other enterprises for different types of ships,

- a wide cooperation of different enterprises when the production of various devices and products in specialised enterprises is taken into account,
- the usage of a progressive section and block ship production method in a manufacturing position - slipway, instead of a partial method, in order to shorten the production process on the slipway to the maximum and thus to shorten the delivery date, too.

Sections are big compartments of the ship's hull which are composed and welded from individual building elements of the plating and stiffening.

Examples include the bottom section with the breadth from one side to the other one, and lengths of several frame spacings, the section of the side, the section of the deck, the section of a bulkhead, the section of superstructure walls, etc.

We distinguish flat and volume sections (e.g. volume sections of the double bottom, double sides and those which feature a significant curvature of the hull).

Side sections - sides are compartments of the hull of a certain length which are composed of flat and volume sections and which are equipped with devices (mechanisms, equipment and lines) in the place of their manufacture. The area where the hull is manufactured from elements, nodes, sections or blocks, is called a slipway. The slipway is equipped with a lifting mechanism, cranes and/or special devices to let a ship down onto water.

For letting the ship down ship lifts and sometimes even tilted wooden or metal "slipways" are used. River ships are let down onto water mostly from side - a so called side launching. It is a consequence of their structure since they are not able to tolerate the overall bending which would arise at the end launching of the ship.

The section and block method of shipbuilding assumes a preliminary production of elements and nodes in special work positions and subsequently their mutual splicing in a slipway position using a specialised work group. Such a method enables to perform the majority of works in covered sheds independently on current weather using mechanised and automated devices. A specialised work is applied to a greater extent, and an operose and difficult process of splicing the individual ship parts and sections, as well as the overall production process and thus the date of delivering the ship to the customer is shortened.

The production is preceded with a design, construction, technological, material, appliances and organisational preparation.

The development component also processes a technological design of the ship production. Firstly a scheme of the hull's division into sections and blocks is drawn up, coming out of the condition that every section or block must have their dimensions and weight aligned with capacities of individual sheds, their lifting and transporting mechanisms. After the composition of sections and blocks they are required to manifest a sufficient strength and stiffness. It happens that a section is finished close to future frame stiffeners or bulkheads. Sometimes it is even necessary to place some temporary stiffenings there.

Afterwards plans of technological appliances and aids are worked out in order to splice and weld details of nodes, sections of blocks and to check the quality of their manufacturing. Bills of materials as well as specifications of devices are prepared to be provided for purchase departments. In the period of appliances production some auxiliary appliances, beds, fixators, etc., are manufactured.

From the point of view of organisational preparations there is worked out an operational plan and the schedule of production management, too. For these tasks methods of mathematical scheduling (network diagrams, etc.) may successfully be utilised.

7.9. BASIC WORK RELATED TO SHIP PRODUCTION

Basic work related to ship production may be divided into the following sequence:

Marking off and Marking Works

In the past there used to exist a rule for each shipyard to have a special shed, a so called template drawing office, where a lines plan of a ship in 1:1 scale was drawn. Joints of sheets and all types of frames were marked on this plan. The plan served as a basis for creating templates, volume models and templates for sheet cutting which were then submitted into the shed in order to manufacture parts.

Today in modern shipyards marking off the parts is done directly in a shed (in order to reduce operoseness) where lines for cutting the parts out of sheet are marked either using a light ray or a programming tape. If it is an optical method it is necessary to prepare plans, of course, in a certain scale on special transparent films.

Production of the Hull's Parts from Sheets and Profiles

Sheets must firstly be flattened on so called flattening rolls, and profiles on a special rolling mill. Then sheets and profiles are cleaned on blast cleaning devices, often with a surface heater to remove forge scales. According to marked lines the sheets are then cut either using scissors or they are kilned in automatic flame cutting machines, e.g. of SICOMAT type. Then edges are amended for welds (this process may also be performed with flame automatic machines). Then on bending machines or under the press there are manufactured flanged structures and a required curvature - either of ships or bent profiles - is created. Almost all bendings and flanges are cold-manufactured nowadays. A hot bending is used only very rarely, e.g. in the manufacturing of such forms which are needed at the construction of buckets, chain-bucket dredgers, etc. Due to a high operoseness and energy intensity there exist efforts to avoid hot flattening.

Welding of Frame Girders, Splicing and Welding of Sections and Blocks

For the production of sections and formed blocks it is necessary to manufacture special "beds", conductors, magnetic walls and other appliances which enable an accurate determination of details and their clamping in accordance with the drawing documentation. In order to decrease the operoseness it is useful to use universal devices which can simply be adjusted to the production and forming of blocks of sections and blocks of various types, dimensions and forms.

Production and Installation Works on a Slipway

Sections or blocks of the hull are placed on the slipway onto special metal or wooden cages and often carts, too. To enable the performance of works even below the ship's hull the "brickwork" should be at least 1 m to 1.3 m high. In the past the axis plane, location of practical frames, as well as of transverse and longitudinal bulkheads was often marked there on the concrete floor of the slipway; they were used to check the precision of the process of splicing individual parts of the ship's hull. On the slipway there are performed works of installation character, too, such as building of devices and equipment in the ship's hull which could not have been installed directly during the production of individual sections and blocks.

The effort of current methods of a progressive technological process is to achieve the maximum completeness of all performed operations on the slipway so the ship is launched in the "most saturated" state possible and is prepared for the testing performed after launching the ship.

Tightness Tests, Launching and Finishing Works

Tests of sections, blocks of the hull and ultimately tests of the hull as a whole with regard to watertightness and airtightness are performed directly in the production plants in compliance with applicable standards. The check of the weld tightness is most frequently performed either with compressed air and subsequent application of a soap water solution onto the other side of the weld, or with a paraffin oil test. Areas wet with the paraffin oil are checked from the other side on a chalk smear. These methods are less operose than e.g. pressuring of the spaces with water which is for example impossible in winter period during freezing.

Finishing Works on Water

These works are performed in specialised quays of a shipyard which are usually equipped with relevant lifting mechanisms. The volume of finishing works performed on the slipway depends on the saturation of the space with mechanisms, on the line systems installation, on wiring, insulation, painting and other works which can be performed on the slipway. All works to finish the ship building are performed per a special schedule and as a result of their high resistance they are also appropriately expensive.

Delivery and Acceptance Works

In principle, the tests are divided into slipway and operational ones. The program of tests is usually set by a designer or a design organisation which is required to submit it for approval to a respective classification society and future customer. Members of the acceptance committee are agents of the customer and organisations (classification society, Traffic Department, etc.), and, of course, they are representatives of the producer and the test crew.

As part of examining tests, mechanisms which ensure the safety of navigation (fire system, generators and other devices and systems) and the main drive unit are checked. The results of tests are recorded in special test protocols. Operational as well as examining tests can be divided into so called business and official ones.

Business tests are performed in order to find out and fix installation failures. After eliminating possible shortcomings there are performed official tests, i.e. acceptance of a ship by the customer. For the acceptance act a special acceptance committee is established.

When official acceptance works are completed an inspection of main and some auxiliary mechanisms is performed (dismantling and remounting of devices after relevant check works are completed). At the end there is performed a so called inspection navigation; as part of this inspection the ship is handed in to the crew, which accepts it.

7.10. NON-METALLIC SHIPS

7.10.1. Ferroconcrete Ships

For a ferroconcrete ship building a standard concrete mixture with addition of cement of a higher strength is used. Structure profiles - fittings are manufactured from structural steel. The structure made of ferroconcrete is monolithic and functions as one whole. A common work of the concrete and steel is possible thanks to their stiff connection. Coefficients of a linear extension of steel and concrete are very similar, almost identical, which indicates their common functional capability without disturbing their firm binding.

Merits of ferroconcrete structures:

- a small consumption of steel per a ton of displacement (20 % to 40 %) when compared to a steel hull,
- a possibility for the production to be made even in less specialised enterprises with simple equipment,
- the reduction of operation costs down to 30 % to 40 % as a consequence of minimum claims to repair, maintenance and long operational life of the structure.

In spite of pros ferroconcrete ships also feature some significant cons:

- a higher weight of the hull per a tone of the loading capacity (1,5 to 2-times greater than in case of steel ships),

- a smaller local strength (a fragile structure, mostly with regard to a hit).

It is these shortcomings which narrow the possibility to utilise such a type of structure for the production of ship's hulls.

The production of concrete mixture requires water, Portland cement with a minimum strength with the pressure of 50 MPa, then sand and gravel and/or another aggregate (ceramsite, etc.). There where exists the tradition of such ships' production there also exist principles of the respective classification society.

The composition of concrete is given with a weight or volume relation of cement, sand and gravel, whereas the amount of sand and gravel particles is given in a ratio to one part of cement plus the amount of water. The relation is called a "water-cement mixture" (e.g. W : S : G = 1 : 0.75 : 2, W/C = 0.5). It is important to check the composition and quality of concrete.

Ship concrete must feature an increased strength, impermeability, frost resistance and plasticity, appropriate formability and a small dimension of gravel particles up to 20 mm.

To ensure a corresponding strength all girders of the structure and the plating of the hull are reinforced using steel rods. The ratio of the surface area of the fitting section to the surface area of the overall section is called a coefficient of the reinforcement which amounts to 1.5 % to 8 % in ship structures and cannot be less than 0.3 % to 0.5 %.

The working fitting of girders comprises longitudinal elements - rods with the diameter of not less than 10 mm which should be it the greatest distance possible from the neutral axis of the transverse section of the girder. Part of working rods passing through multiple shores in their place, where the bending moment changes its meaning, is shifted from one zone to another one so the rods follow the direction of main tension stresses. The number and the surface area of fitting rods as well as their position in the transverse section of the girder is determined on the basis of a calculation.

Longitudinal (working) fitting girders with their intended spatial positioning in the section are fixed using contact welds or electrowelds to transverse yokes in order to create stiff structural elements of the fitting. Moreover yokes increase the stability of longitudinal rods and accept shear and main stresses in girders.

Plates of the bottom and the deck are reinforced with a net which is made of rods mutually perpendicularly tied in so called truss points. In case of a double-tier arrangement the nets are placed in a checkered pattern above each other.

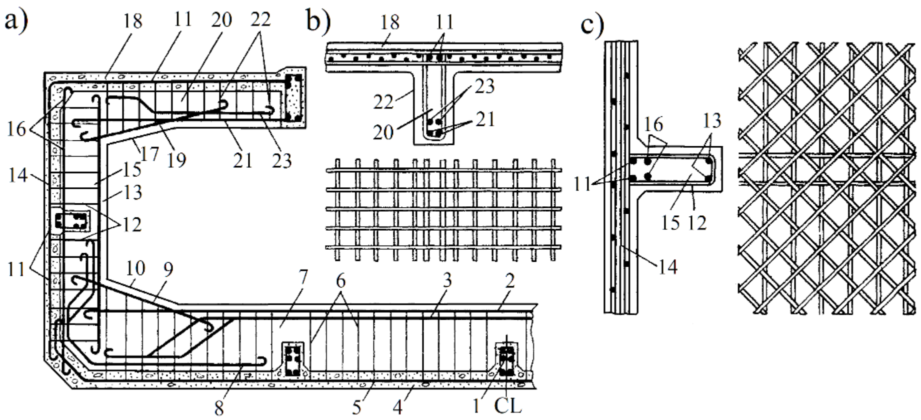

Fig. 79. Structure of the Ferroconcrete Ship's Hull [Authors, 4]

a) main frame; b) transverse and deck plate (a bottom transverse and bottom plates); c) structure of a side frame and side plates; 1 - axial keel, 2 - straight rods of a function fitting of the transverse, 3 - bent bars of a function fitting, 4 - bottom plate, 5 - straight rods of the transverse positioned in the bottom plate, 6 - yokes of the transverse, 7 - transverse, 8 - bent bars reeved from a transverse and side frame, 9 - "floating" bar of a bilge bevel, 10 - bilge bevel, 11 - reeved bars from a side frame into a half-transverse, 12 - yokes of a side frame, 13 - straight fitting bar of the side frame, 14 - side plate, 15 - side frame, 16 - straight fitting bars of a side frame, 17 - deck bracket, 18 - deck plate, 19 - "floating" fitting bar of a deck bracket, 20 - half-transverse, 21 - bent bars of a function fitting of a half-transverse, 22 - yokes, 23 - bent bars of a half-transverse

Rods of outer nets in plates on the side plating and longitudinal bulkheads are mostly placed at the angle of 45° to the plane of the bottom or the deck (a skew net). The reason is that in the side and longitudinal bulkhead by the overall bending there arise tangent and normal stresses which together evoke main tension and pressure stresses directed approx. in the direction of the angle $\pm 45°$. Besides there are also placed vertical central rods in the net.

To protect the rods against corrosion it is required to cover the entire fitting of girders and plates reliably with a layer of concrete with the minimum thickness of 10 mm.

In order to increase the strength (of the anchoring) all ends of working rods of the fitting of plates and girders are terminated with a so called "bell-crank (bending)" or they are left to be anchored into a compressed zone. In corner angles there must always be a reinforced bracket.

Structure girders are positioned analogically to steel ships. The body of the hull is usually constructed according to the longitudinal, transverse or mixed system of stiffening. Frame spacing is 1 m to 5 m. The distance between longitudinal girders is 1 m to 2 m. The ratio of a rectangular girder is 1:3 to 1:6 and the thickness of the plating plate is between 40 mm and 60 mm.

A ferroconcrete ship can be manufactured either as a monolithical ship or with a so called panel - section method.

In case of a monolithical method all works performed during manufacturing the hull are to be done in a strict sequence on the slipway. Here the fitting is bound and welded into a form which is closed into a mould (shuttering). The mould is made from boards, plywood or metal sheets and it corresponds to the outer form of the hull. The space between the edges of the mould shell is filled with a concrete mixture. When the concrete is hardened the mould is dismantled and potential defects of concrete are removed.

Shortcomings of a monolithical building:

- impossibility to diversify the work "queue" anyhow,
- limited possibility to mechanise the work,
- big consumption of specialised materials,
- difficulties with ensuring a thorough placing ("compacting") of concrete mainly into narrow spaces of the mould,
- seasonal character of the ship.

In case of a panel-section method of the shipbuilding the structure is often manufactured on the slipway and the greater part there in a shed. Monolithic elements manufactured on the slipway include plates and keelsons of the bottom as well as an underdeck girder, i.e. the most exposed structural elements of the hull girder. Sides, transverse bulkheads and deck plates can be manufactured in the shed. They are spliced through welding of protruding fitting parts on the slipway. Concrete is applied on splices; a quick-hardening, expanding (tiling plaster-aluminium) cement is used.

Advantages:

- reduced operoseness of the hull works to 20 % to 40 %,
- reduced consumption of metal for "shuttering" to 50 % to 80 %,
- 2 to 4-times shorter production cycle (the slipway occupation is reduced in 10 to 15 days).

The weight of hulls of ferroconcrete ships can be reduced using light plastic materials (ceramsite), reinforced cement and prestressed structures.

Ceramsite is gained through kilning of various kinds of clay and bentonite at a high temperature. Particles of ceramsite feature a tough shell from outside and contain air space inside. The density of material is $0.6\ \mathrm{t}/m^3$ to $0{,}8\ \mathrm{t}/m^3$ and the strength is sufficient for concrete with the overall strength between 25 MPa to 35 MPa and the overall density of $1{,}8\ \mathrm{t}/m^3$. The usage of ceramsite reduces the overall weight of the ship in 15 % to 30 %.

Reinforced cement structures are reinforced with a netting when the wire thickness is of 0.5 mm to 1.5 mm and concrete structures comprise a cement-sand mixture. Dimensions of the grid are 8 mm to 15 mm with nettings put into several layers with the spacing of 4 mm to 10 mm.

Reinforced cement plates are flexible, lighter, they transfer hits and deliver the capability of eliminating the plating thickness of the hull and superstructures between 10 mm and 20 mm. The consumption of steel in reinforced cement plates is higher than in standard ferroconcrete structures and it ranges from $450\ \mathrm{kg}/m^3$ to $500\ \mathrm{kg}/m^3$. As a result of eliminated thickness it is possible to achieve an overall saving of concrete and steel in reinforced cement structures and thus to reduce the overall weight of the hull up to 40 %.

Prestressed concrete represents structures where the fitting is safely anchored and prestressed. When the concrete hardens and the stiffener loosens the stiffener occurs in the state of a thrust, however, the concrete is in the area of pressure. This way the functional capability of concrete with regard to pressure and thrust significantly equalises. The fitting is stretched to the stress of at least $300MPa$ and $\sigma_r \geq 500MPa$, depending on the intended strength characteristics of both concrete and fitting. A so called thermal prestress of the fitting to the temperature of $300°C$ to $350°C$ is allowed.

Advantages:

- reducing the weight of the hull in approx. 20 %,
- reducing the consumption of fitting in 30 % to 60 %,
- reducing the consumption of cement in 5 % to 40 %, whereas the hull is manufactured without any ruptures.

7.11. SHIPS MADE OF PLASTICS

In the shipbuilding polyester and epoxy resins hardened at normal temperatures are usually used for the hull construction. The advantage of these materials, which ultimately allows for their usage, is their "flame retardance" which does not support self-burning after preventing a direct flame. Another advantage lies in their permanency of form (they do not become softer while warming). The process of hardening occurs after supplementing a small amount (3 % to 8 %) of special catalysts and accelerators of hardening into the resin.

Hardened non-reinforced plastics do not feature corresponding strength characteristics. The thrust strength is 40 MPa on average and the pressure strength is 100 MPa. To increase the strength of a structure plastics are reinforced using a glass fabric or other fibre materials. Plastics reinforced with glass fibres are called fibreglass laminates. The strength under the thrust of such materials reaches 200 MPa to 400 MPa and under pressure 150 MPa to 300 MPa. These values grow depending on the rate of increasing the coefficient of reinforcement μ_a which represents the ratio of the weight of a glass fibre to the weight of the overall structure of the given 1-pressure volume. Despite, however, the increase of $\mu_a$to a value greater than 60 % to 70 % is not possible since the solution of resin could not cover the surface of fibres reliably from all sides.

Main Advantages of a Glass Fibre:

- low density which enables to reduce the weight of the hull for ships with the length of up to 10 m approximately 4-times when compared to aluminium alloys, and for ships with a greater length the weight can be reduced 1,5 up to 3-times,
- ships made of plastics do not require coatings, they do not corrode and they have a smooth surface.

In spite of their advantages fibreglass laminates do also have some imperfections:

- low modulus of elasticity,
- ageing,
- high price.

As a result of these shortcomings this material is only used for building of not too big ships with the length up to max. 20 m (rescue boats, small sporting ships and purpose-built crafts).

The ratio of the strength modulus to the allowable strength for fibreglass laminates is 4 to 6-times smaller than for steel, and 3-times smaller than for aluminium alloys.

This means that in order to ensure the same stiffness as for steel ships or to ensure the same ratio values of critical stresses in compressed structures it is necessary to significantly increase the transverse section of structural elements of the ship, and of plastics. However, such a solution is not economically profitable.

The ageing of material or the loss of strength over the time is another big shortcoming. According to preliminary data the strength decreases in 20 % even in case of an unloaded structure in the course of 10 years. Under the effect of a permanent load the loss of strength increases constantly. Under the effect of water environment the loss of the strength grows even more. Also the rise of temperature above $40°C$ reduces the strength of fibreglass laminate. Moreover the issue of joining individual structural elements of fibreglass laminate is not solved sufficiently.

Fibreglass laminate ships can be divided into the following types by the method of their construction:

- shell ships without a stiffening (the smallest boats),
- ships stiffened with (transverse and longitudinal) girders with a one-layer plating,
- structures without a stiffening with a multilayer plating. The space between platings is filled with light porous materials (rescue boats, sailboards, etc.).

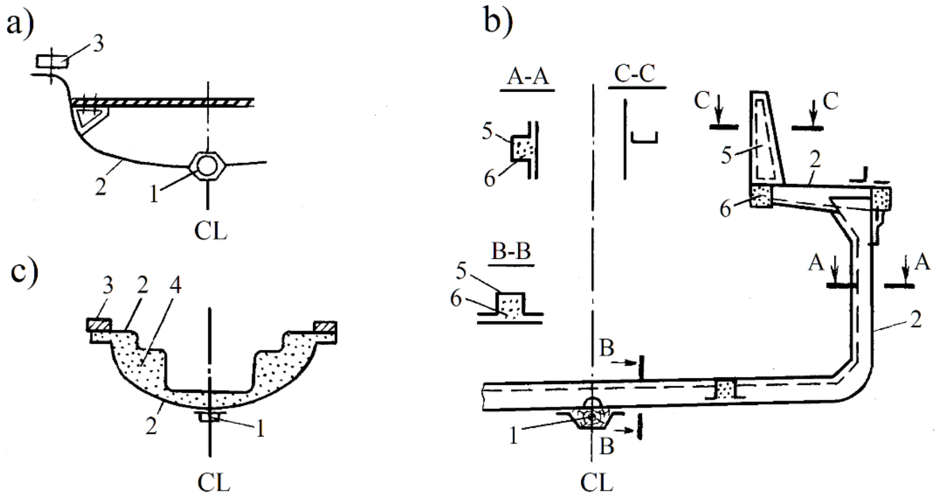

Fig. 80. Construction Scheme of Ships Made of Plastics [Authors, 4]

a) structure of a cruise boat without a stiffening; b) three-layer structure of a rescue boat; c) one-layer plating stiffened with transverse and longitudinal girders of a cargo self-propelled ship; 1 - wooden keel laminated with glass-plastic, 2 - outer and innerboard plating made of glass-plastic, 3 - wooden handle, 4 - fully plastic filler, 5 - glass-plastic, 6 - formed glass-plastic (filler) of skeleton girders

As a result of big thicknesses multilayer structures feature a greater stiffness and stability, and thus they are the most perspective structures from the point of view of development and building.

To build fibreglass laminate vessels monolithic and section methods of building are applied. Lamination can be done manually or using mechanised devices, however, often they are even combined. Lamination of steel structural elements is not suitable due to a big difference of the degree of expansion. Wooden elements, etc., are more suitable.

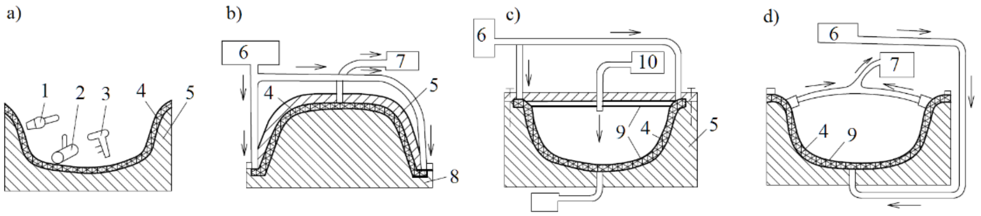

Fig. 81. Technological Schemes of Building Plastic Ships [Authors, 4]

a) method of contact forming (sequential layering); b) vacuum method of forming; c) method of a rubber bellow with pressing the air into the bellow; d) method of a rubber bellow with exhausting the air from the space between the bellow and moulding head; 1 - brush, 2 - hand roller, 3 - sprinkler for the resin application, 4 - formed plating of the ship's hull, 5 - matrix, 6 - air reservoir, 7 - vacuum pump for the air exhaustion, 8 - form, 9 - rubber bellow, 10 - pressing pump to supply air into the bellow

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REVIEWS

The textbook entitled "Construction and Shipbuilding" deals with the physics of vessels under the conditions of sea and inland navigation. In seven chapters authors of this publication from Faculty of Operation and Economics of Transport and Communication, University of Zilina presented the problems of vessels design in the areas of dimensions, buoyancy, stability, unsinkability, manoeuvrability and oscillations, so as a strenght and constructions of ships.

The structure of this textbook matches well the problem definition, purpose and logic of the process of ship construction. The reader will find both calculations of construction parameters as well as ship design related issues in the related problems.

The textbookh will strongly contribute the reader's knowledge on the trends in ship design researches, including both scientific and practical approaches. For that reason I would like to express my strong recommendation to published it as a texbook for students.

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The professional textbook entitled "Construction of Ships and Shipbuilding" is focused on the technical parameters of a variety of ships from the perspective of maritime, sea and inland waterway transport. The group of authors is appropriately composed and sufficiently prepared from the theory and erudition point of view. They describe various issues of ship constructions in terms of their division, parameters, stability, buoyancy, unsinkability, maneuverability, as well as their strength.

The proposed textbook will definitely contribute to readers' erudition and their knowledge related to the current development trends in terms of research specialized on the ship constructions. The potential readers will be able to educate themselves within the particular topics covering ship specific construction dimension calculation examples as well as technical design of individual ship parameters.

Given the aforementioned, it can be stated that the structure of the reviewed textbook perfectly comply with the addressed issue, objective and desired outcome in regard to the ship constructions. Based on previously stated assumptions and declarations, I would like to recommend and suggest publishing the reviewed literature material in a form of textbook for students at individual universities, colleges, faculties and other institutions.

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