# Chapter 10 HULL OUTFITTING

## Section 1 RUDDER AND MANOEUVRING ARRANGEMENT

**Symbols**

For symbols not defined in this Section, refer to **Ch 1**, **Sec 4**.

$C_R$ : Rudder force, in *N*

$Q_R$ : Rudder torque, in *N-m*

$A$ : Total movable area of the rudder, in $m^2$, measured at the mid-plane of the rudder
For nozzle rudders, $A$ is not to be taken less than 1.35 *times* the projected area of the nozzle.

$A_t$ : Area equal to $A$ + area of a rudder horn, if any, in $m^2$

$A_f$ : Portion of rudder area located ahead of the rudder stock axis, in $m^2$

$b$ : Mean height of rudder area, in *m*

$c$ : Mean breadth of rudder area, in *m*, see **Fig. 1**

$\Lambda$ : Aspect ratio of rudder area $A_t$, taken equal to:

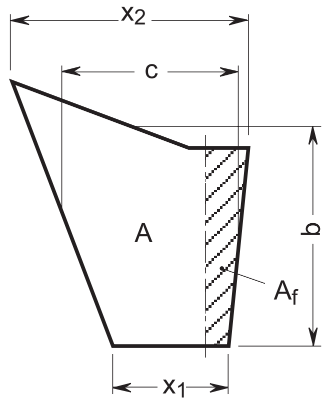
$$\Lambda = \frac{b^2}{A_t}$$

$V_0$ : Maximum ahead speed, in knots, as defined in **Ch 1**, **Sec 4**. If this speed is less than 10, $V_0$ is to be replaced by:

$$V_{\min} = \frac{(V_0 + 20)}{3}$$

$V_a$ : Maximum astern speed, in knots, to be taken not less than $0.5V_0$. For greater astern speeds special evaluation of rudder force and torque as a function of the rudder angle may be required. If no limitations for the rudder angle at astern condition is stipulated, the factor $\kappa_2$ is not to be taken less than given in **Table 1** for astern condition.

Fig. 1 Dimensions of rudder




$$c = \frac{x_1 + x_2}{2} \qquad b = \frac{A}{c}$$

### 1. General

#### 1.1 Manoeuvring arrangement

1.1.1

Each ship is to be provided with a manoeuvring arrangement which will guarantee sufficient manoeuvring capability.

The manoeuvring arrangement includes all parts from the rudder and steering gear to the steering position necessary for steering the ship.

1.1.2

Rudder stock, rudder coupling, rudder bearings and the rudder body are dealt with in this Section. The steering gear is to comply with the provisions of **Part D**.

1.1.3

The steering gear compartment shall be readily accessible and, as far as practicable, separated from the machinery space.

Note: Concerning the use of non-magnetisable material in the wheel house in way of a magnetic compass, the requirements of the national Administration concerned are to be observed.

### 1.2 Structural details

1.2.1

Effective means are to be provided for supporting the weight of the rudder body without excessive bearing pressure, e.g. by a rudder carrier attached to the upper part of the rudder stock. The hull structure in way of the rudder carrier is to be suitably strengthened.

1.2.2

Suitable arrangements are to be provided to prevent the rudder from lifting.

1.2.3

Connections of rudder blade structure with solid parts in forged or cast steel, which are used as rudder stock housing, are to be suitably designed to avoid any excessive stress concentration at these areas.

1.2.4

The rudder stock is to be carried through the hull either enclosed in a watertight trunk, or glands are to be fitted above the deepest load waterline, to prevent water from entering the steering gear compartment and the lubricant from being washed away from the rudder carrier. If the top of the rudder trunk is below the deepest waterline two separate stuffing boxes are to be provided.

1.3 Size of rudder area

In order to achieve sufficient manoeuvring capability the size of the movable rudder area $A$ is recommended to be not less than obtained, in $m^2$, from the following formula:

$$A = c_1\, c_2\, c_3\, c_4\, \frac{1.75LT}{100}$$

where:

$c_1$ : Factor taken equal to 0.9

$c_2$ : Factor for the rudder type:

- $c_2 = 1.0$ in general
- $c_2 = 0.9$ for semi-spade rudders
- $c_2 = 0.7$ for high lift rudders

$c_3$ : Factor for the rudder profile:

- $c_3 = 1.0$ for *NACA*-profiles and plate rudder
- $c_3 = 0.8$ for hollow profiles and mixed profiles

$c_4$ : Factor for the rudder arrangement:

- $c_4 = 1.0$ for rudders in the propeller jet
- $c_4 = 1.5$ for rudders outside the propeller jet

For semi-spade rudders 50% of the projected area of the rudder horn may be included into the rudder area $A$.

Where more than one rudder is arranged the area of each rudder can be reduced by 20%.

In estimating the rudder area $A$, **0** is to be considered.

### 1.4 Materials

1.4.1

For materials for rudder stock, pintles, coupling bolts etc. refer to **Part K**.

1.4.2

In general, materials having $R_{eH}$ of less than 200 $N/mm^2$, $R_m$ of less than 400 $N/mm^2$ or more than 900 $N/mm^2$ are not to be used for rudder stocks, pintles, keys and bolts. The requirements of this Section are based on a with $R_{eH}$ of

235 $N/mm^2$. If material is used having a $R_{eH}$ differing from 235 $N/mm^2$, the material factor $k_r$ is to be determined as follows:

$$k_r = \left(\frac{235}{R_{eH}}\right)^{0.75} \quad \text{for } R_{eH} > 235$$

$$k_r = \frac{235}{R_{eH}} \quad \text{for } R_{eH} \le 235$$

where:

$R_{eH}$ : Minimum yield stress of material used, in $N/mm^2$. $R_{eH}$ is not to be taken greater than $0.7R_m$ or 450 $N/mm^2$, whichever is less.

1.4.3

Before significant reductions in rudder stock diameter due to the application of steels with $R_{eH}$ exceeding 235 $N/mm^2$ are accepted, the Society may require the evaluation of the elastic rudder stock deflections. Large deflections should be avoided in order to avoid excessive edge pressures in way of bearings.

1.4.4

The permissible stresses given in **5.1** are applicable for normal strength steel. When higher strength steels are used, higher values may be used for the permissible stresses, on a case by case basis.

## 2. Rudder force and torque

### 2.1 Rudder force and torque for normal rudders

2.1.1

The rudder force is to be determined, in $N$, according to the following formula:

$$C_R = 132\, A V^2 \kappa_1\, \kappa_2\, \kappa_3\, \kappa_t$$

where:

$V$ : $V_0$ for ahead condition

$V_a$ for astern condition

$\kappa_1$ : Coefficient, depending on the aspect ratio $\Lambda$, taken equal to:

$\kappa_1 = (\Lambda + 2)/3$, where $\Lambda$ need not be taken greater than 2

$\kappa_2$ : Coefficient, depending on the type of the rudder and the rudder profile according to **Table 1**

Table 1 Coefficient $\kappa_2$

| Profile / type of rudder | $\kappa_2$ | |
|---|---|---|
| | Ahead | Astern |
| *NACA*-00 series Göttingen profiles | 1.10 | 0.80 |
| Flat side profiles | 1.10 | 0.90 |
| Mixed profiles (e. g. *HSVA*) | 1.21 | 0.90 |
| Hollow profiles | 1.35 | 0.90 |
| High lift rudders | 1.70 | to be specially considered; if not known: 1.30 |
| Fish tail | 1.40 | 0.80 |
| Single plate | 1.00 | 1.00 |

$\kappa_3$ : Coefficient, depending on the location of the rudder, taken equal to:

$\kappa_3$ = 0.80 for rudders outside the propeller jet

$\kappa_3$ = 1.00 elsewhere, including also rudders within the propeller jet

$\kappa_3$ = 1.15 for rudders aft of the propeller nozzle

$\kappa_t$ : Coefficient equal to 1.0 for rudders behind propeller. Where a thrust coefficient $C_{Th} > 1.0$, the Society may consider a coefficient $\kappa_t$ different from 1.0, on a case by case basis.

2.1.2

The rudder torque, in *Nm*, is to be determined by the following formula:

$$Q_R = C_R\, r$$

where:

$r$ : Lever of the force $C_R$, in *m*, taken equal to:

$r = c(\alpha - k_{bc})$, without being less than $0.1c$ for ahead condition

$\alpha$ : Coefficient taken equal to:

$\alpha = 0.33$ for ahead condition

$\alpha = 0.66$ for astern condition (general)

$\alpha = 0.75$ for astern condition (hollow profiles)

For parts of a rudder behind a fixed structure such as a rudder horn:

$\alpha = 0.25$ for ahead condition

$\alpha = 0.55$ for astern condition

For high lift rudders $\alpha$ is to be specially considered. If not known, $\alpha = 0.40$ may be used for the ahead condition

$k_{bc}$ : Balance factor as follows:

$$k_{bc} = \frac{A_f}{A}$$

$k_{bc} = 0.08$ for unbalanced rudders

2.1.3

Effects of the provided type of rudder/profile on choice and operation of the steering gear are to be observed.

**2.2 Rudder force and torque for rudder blades with cut-outs (semi-spade rudders)**

2.2.1

The total rudder force $C_R$ is to be calculated according to **2.1.1**. The pressure distribution over the rudder area, upon which the determination of rudder torque and rudder blade strength are to be based, is to be obtained as follows:

- the rudder area may be divided into two rectangular or trapezoidal parts with areas $A_1$ and $A_2$, see **Fig. 2**.
- the resulting force, in *N*, of each part may be taken as:

$$C_{R1} = C_R \frac{A_1}{A}$$

$$C_{R2} = C_R \frac{A_2}{A}$$

2.2.2

The resulting torque, in *N-m*, of each part is to be taken as:

$$Q_{R1} = C_{R1}\, r_1$$

$$Q_{R2} = C_{R2}\, r_2$$

where:

$r_1 = c_1(\alpha - k_{b1})$ , in *m*

$r_2 = c_2(\alpha - k_{b2})$ , in *m*

$$k_{b1} = \frac{A_{1f}}{A_1}$$

$$k_{b2} = \frac{A_{2f}}{A_2}$$

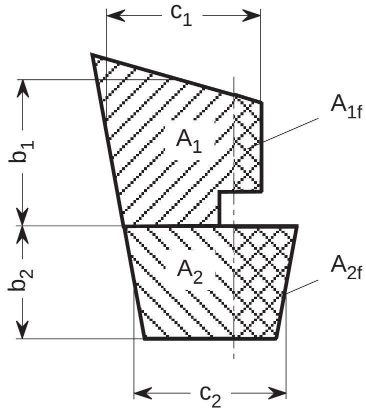
$A_{1f}$, $A_{2f}$ : As defined in **Fig. 2**

$$c_1 = \frac{A_1}{b_1}$$

$$c_2 = \frac{A_2}{b_2}$$

$b_1$, $b_2$ : Mean heights of the partial rudder areas $A_1$ and $A_2$ (see **Fig. 2**).

Fig. 2 Areas $A_1$ and $A_2$




2.2.3

The total rudder torque, in *N-m*, is to be determined according to the following formulae:

$Q_R = Q_{R1} + Q_{R2}$, without being less than $Q_{R\min} = C_R \, r_{1,2\min}$

where:

$$r_{1,2\min} = \frac{0.1}{A}\left(c_1 A_1 + c_2 A_2\right) \text{, in } m.$$

## 3. Scantlings of the rudder stock

### 3.1 Rudder stock diameter

3.1.1

The diameter of the rudder stock, in *m*, for transmitting the rudder torque is not to be less than:

$$D_t = 4.2 \sqrt[3]{Q_R \, k_r}$$

where:

$Q_R$ : As defined in **2.1.2**, **2.2.2** and **2.2.3**

The related torsional stress, in $N/mm^2$, is:

$$\tau_t = \frac{68}{k_r}$$

where:

$k_r$ : As defined in **1.4.2** and **1.4.3**.

3.1.2

The diameter of the rudder stock determined according to **3.1.1** is decisive for the steering gear, the stopper and the locking device.

3.1.3

In case of mechanical steering gear the diameter of the rudder stock in its upper part which is only intended for transmission of the torsional moment from the auxiliary steering gear may be $0.9D_t$. The length of the edge of the quadrangle for the auxiliary tiller must not be less than $0.77D_t$ and the height not less than $0.8D_t$.

3.1.4

The rudder stock is to be secured against axial sliding. The degree of the permissible axial clearance depends on the construction of the steering engine and on the bearing.

### 3.2 Strengthening of rudder stock

3.2.1

If the rudder is so arranged that additional bending stresses occur in the rudder stock, the stock diameter has to be suitably increased. The increased diameter is, where applicable, decisive for the scantlings of the coupling.

For the increased rudder stock diameter the equivalent stress of bending and torsion, in $N/mm^2$, is not to exceed the following value:

$$\sigma_v = \sqrt{\sigma_b^2 + 3\tau^2} \le \frac{118}{k_r}$$

where:

$\sigma_b$ : Bending stress, in $N/mm^2$, equal to:

$$\sigma_b = \frac{10.2\, M_b}{D_1^3}$$

$M_b$ : Bending moment at the neck bearing, in *N-m*

$\tau$ : Torsional stress, in $N/mm^2$, equal to:

$$\tau = \frac{5.1\, Q_R}{D_1^3}$$

$D_1$ : Increased rudder stock diameter, in *cm*, equal to:

$$D_1 = 0.1 D_t \sqrt[6]{1 + \frac{4}{3}\left(\frac{M_b}{Q_R}\right)^2}$$

$Q_R$ : As defined in **2.1.2**, **2.2.2** and **2.2.3**

$D_t$ : As defined in **3.1.1**.

Note: Where a double-piston steering gear is fitted, additional bending moments may be transmitted from the steering gear into the rudder stock. These additional bending moments are to be taken into account for determining the rudder stock diameter.

### 3.3 Analysis

3.3.1 General

The bending moments, shear forces and support forces for the system rudder - rudder stock are to be obtained from **3.3.2** and **3.3.3**, for rudder types as shown in **Fig. 3** to **Fig. 7**.

3.3.2 Data for the analysis

$\ell_{10}$ ,.., $\ell_{50}$ : Lengths, in *m*, of the individual girders of the system

$I_{10}$ ,.., $I_{50}$ : Moments of inertia of these girders, in $cm^4$

For rudders supported by a sole piece the length $\ell_{20}$ is the distance between lower edge of rudder body and centre of sole piece, and $I_{20}$ is the moment of inertia of the pintle in the sole piece.

Load on rudder body, in *kN/m*, (general):

$$p_R = \frac{C_R}{\ell_{10} \cdot 10^3}$$

Load on semi-spade rudders, in *kN/m*:

$$p_{R10} = \frac{C_{R2}}{\ell_{10} \cdot 10^3}$$

$$p_{R20} = \frac{C_{R1}}{\ell_{20} \cdot 10^3}$$

$C_R$, $C_{R1}$, $C_{R2}$ : As defined in **2.1** and **2.2**

$Z$ : Spring constant, in *kN/m*, of support in the sole piece or rudder horn respectively:

for the support in the sole piece (see **Fig. 3**):

$$Z = \frac{6.18\, I_{50}}{\ell_{50}^3}$$

for the support in the rudder horn (see **Fig. 4**):

$$Z = \frac{1}{f_b + f_t}$$

$f_b$ : Unit displacement of rudder horn, in *m/kN*, due to a unit force of 1 *kN* acting in the centre of support

$$f_b = \frac{1.3\, d^3\, 10^8}{3\, E\, I_n}$$

$$f_b = 0.21 \frac{d^3}{I_n} \quad \text{(guidance value for steel)}$$

$I_n$ : Moment of inertia of rudder horn, in $cm^4$, around the $x$-axis at $d/2$ (see **Fig. 4**)

$f_t$ : Unit displacement due to a torsional moment of the amount 1, in $m/kN$

$$f_t = \frac{d\,e^2}{G\,J_t}$$

$$f_t = \frac{d\,e^2 \sum u_i / t_i}{3.17 \cdot 10^8\, F_T^2} \quad \text{for steel}$$

$G$ : Modulus of rigity, $kN/m^2$:

$G = 7.92 \cdot 10^7$ for steel

$J_t$ : Torsional moment of inertia, in $m^4$

$F_T$ : Mean sectional area of rudder horn, in $m^2$

$u_i$ : Breadth, in *mm*, of the individual plates forming the mean horn sectional area

$t_i$ : Plate thickness of individual plate having breadth $u_i$, in *mm*

$e$, $d$ : Distances, in $m$, according to **Fig. 4**

$K_{11}$, $K_{22}$, $K_{12}$ : Rudder horn compliance constants calculated for rudder horn with 2-conjugate elastic supports (**Fig. 5**).The 2-conjugate elastic supports are defined in terms of horizontal displacements, $y_i$, by the following equations:

at the lower rudder horn bearing:

$$y_1 = -K_{12}\, F_{A2} - K_{22}\, F_{A1}$$

at the upper rudder horn bearing:

$$y_2 = -K_{11}\, F_{A2} - K_{12}\, F_{A1}$$

where

$y_1$, $y_2$ : Horizontal displacements, in $m$, at the lower and upper rudder horn bearings, respectively

$F_{A1}$, $F_{A2}$ : Horizontal support forces, in $kN$, at the lower and upper rudder horn bearings, respectively

$K_{11}$, $K_{22}$, $K_{12}$ : Obtained, in $m/kN$, from the following formulae:

$$K_{11} = 1.3 \frac{\lambda^3}{3EJ_{1h}} + \frac{e^2 \lambda}{GJ_{th}}$$

$$K_{12} = 1.3 \left[ \frac{\lambda^3}{3EJ_{1h}} + \frac{\lambda^2 (d-\lambda)}{2EJ_{1h}} \right] + \frac{e^2 \lambda}{GJ_{th}}$$

$$K_{22} = 1.3 \left[ \frac{\lambda^3}{3EJ_{1h}} + \frac{\lambda^2 (d-\lambda)}{EJ_{1h}} + \frac{\lambda (d-\lambda)^2}{EJ_{1h}} + \frac{(d-\lambda)^3}{3EJ_{2h}} \right] + \frac{e^2 d}{GJ_{th}}$$

$d$ : Height of the rudder horn, in $m$, defined in **Fig. 5**. This value is measured downwards from the upper rudder horn end, at the point of curvature transition, till the mid-line of the lower rudder horn pintle

$\lambda$ : Length, in $m$, as defined in **Fig. 5**. This length is measured downwards from the upper rudder horn end, at the point of curvature transition, till the mid-line of the upper rudder horn bearing. For $\lambda = 0$, the above formulae converge to those of spring constant $Z$ for a rudder horn with 1-elastic support, and assuming a hollow cross section for this part

$e$ : Rudder-horn torsion lever, in $m$, as defined in **Fig. 5** (value taken at $z = d/2$)

$J_{1h}$ : Moment of inertia of rudder horn about the $x$ axis, in $m^4$, for the region above the upper rudder horn bearing. Note that $J_{1h}$ is an average value over the length $\lambda$ (see **Fig. 5**)

$J_{2h}$ : Moment of inertia of rudder horn about the $x$ axis, in $m^4$, for the region between the upper and lower rudder horn bearings. Note that $J_{2h}$ is an average value over the length $d - \lambda$ (see **Fig. 5**)

$J_{th}$ : Torsional stiffness factor of the rudder horn, in $m^4$

For any thin wall closed section

$$J_{th} = \frac{4F_T^2}{\sum_i \frac{u_i}{t_i}}$$

$F_T$ : Mean of areas enclosed by outer and inner boundaries of the thin walled section of rudder horn, in $m^2$

$u_i$ : Length, in *mm*, of the individual plates forming the mean horn sectional area

$t_i$ : Thickness, in *mm*, of the individual plates mentioned above.

Note that the $J_{th}$ value is taken as an average value, valid over the rudder horn height.

3.3.3 Moments and forces to be evaluated

a) The bending moment $M_R$ and the shear force $Q_1$ in the rudder body, the bending moment $M_b$ in the neck bearing and the support forces $B_1$, $B_2$, $B_3$ are to be evaluated.

The so evaluated moments and forces are to be used for the stress analyses required by **3.2**, **5**, **9.1** and **9.2**.

b) For spade rudders (see **Fig. 6**) the moments, in *N-m*, and forces, in *N*, may be determined by the following formulae:

$$M_b = C_R\left(\ell_{20} + \frac{\ell_{10}(2x_1 + x_2)}{3(x_1 + x_2)}\right)$$

$$B_3 = \frac{M_b}{\ell_{30}}$$

$$B_2 = C_R + B_3$$

c) For spade rudders with rudders trunks supporting rudder force (see **Fig. 7**) the moments, in *N-m*, and forces, in *N*, may be determined by the following formulae:

$M_R$ is the greatest of the following values:

$M_R = C_{R2}\ (\ell_{10} - CG_{2Z})$

$M_R = C_{R1}\ (CG_{1Z} - \ell_{10})$

where :

$C_{R1}$ : Rudder force over the rudder blade area $A_1$

$C_{R2}$ : Rudder force over the rudder blade area $A_2$

$CG_{1Z}$ : Vertical position of the centre of gravity of the rudder blade area $A_1$

$CG_{2Z}$ : Vertical position of the centre of gravity of the rudder blade area $A_2$

$M_B = C_{R2}\ (\ell_{10} - CG_{2Z})$

$B_3 = (M_B + M_{CR1}) / (\ell_{20} + \ell_{30})$

$B_2 = C_R + B_3$

Fig. 3 Rudder supported by sole piece

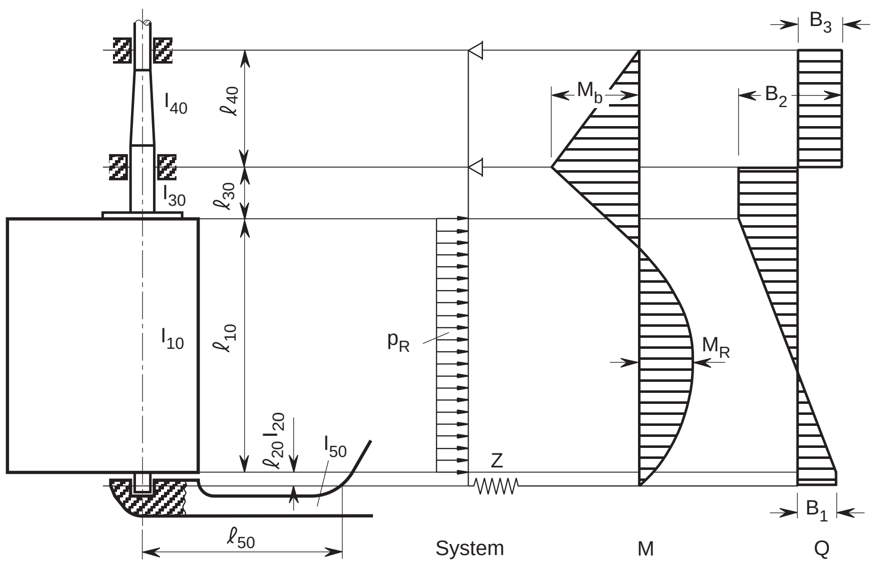


Fig. 4 Semi-spade rudder (with 1-elastic support)

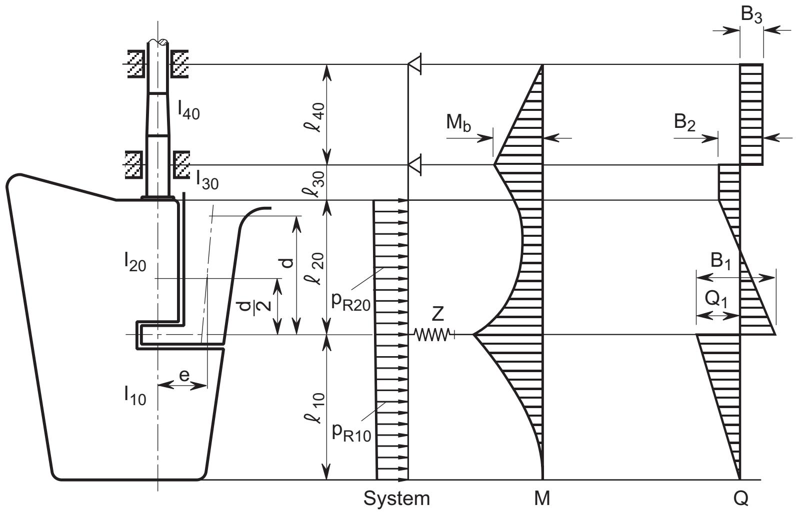



Fig. 5 Semi-spade rudder (with 2-conjugate elastic supports)

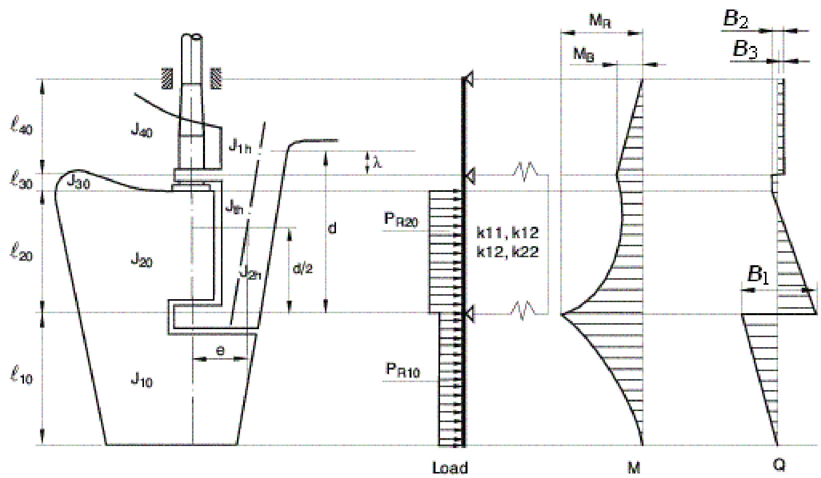



Fig. 6 Spade rudder

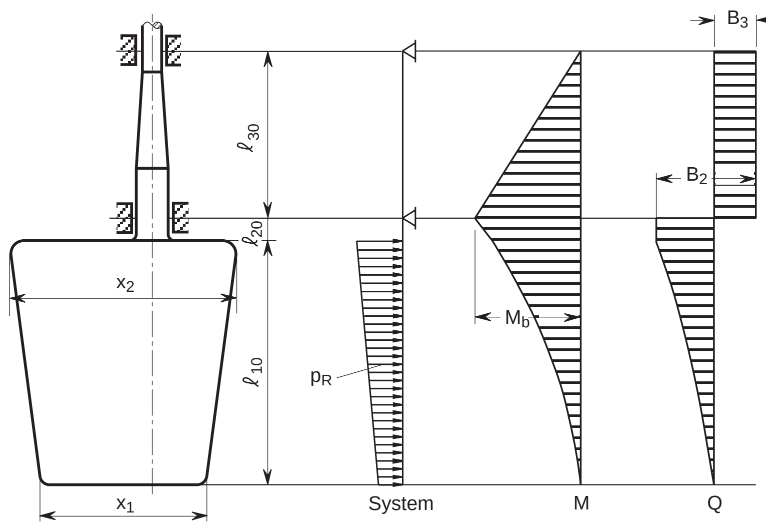
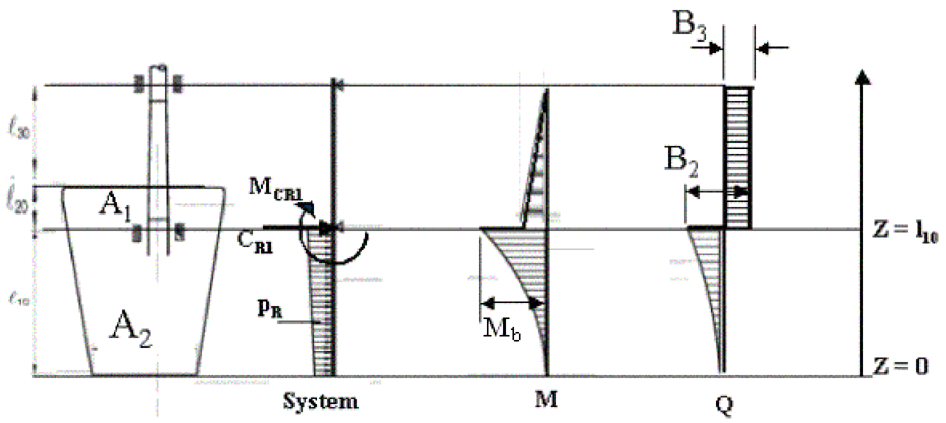


Fig. 7 Spade rudders with rudder trunks




### 3.4 Rudder trunk supporting rudder stock

3.4.1

Where the rudder stock is arranged in a trunk in such a way that the trunk is stressed by forces due to rudder action, the scantlings of the trunk are to be as such that the equivalent stress due to bending and shear does not exceed $0.35\,R_{eH}$ of the material used.

3.4.2

In case where the rudder stock is fitted with a rudder trunk which is arranged in such a way the rudder trunk is loaded by the pressure induced on the rudder blade, as given in **2.1.1**, the bending stress in the rudder trunk, in $N/mm^2$, is to be in compliance with the following formula:

$\sigma \leq 80 / k$

where the material factor $k$ for the rudder trunk is not to be taken less than 0.7.

For the calculation of the bending stress, the span to be considered is the distance between the mid-height of the lower rudder stock bearing and the point where the trunk is clamped into the shell or the bottom of the skeg.

3.4.3

The steel used for the rudder trunk is to be of weldable quality, with a carbon content not exceeding 0.23% on ladle analysis and a carbon equivalent *CEQ* not exceeding 0.41.

3.4.4

The weld at the connection between the rudder trunk and the shell or the bottom of the skeg is to be full penetration.

The fillet shoulder radius $r$, in *mm*, is to be as large as practicable and to comply with the following formulae:

$r = 60$ when $\sigma \geq 40 / k$ $N/mm^2$

$r = 0.1D_1$ when $\sigma < 40 / k$ $N/mm^2$

without being less than 30,

where $D_1$ is defined in **3.2.1**.

The radius may be obtained by grinding. If disk grinding is carried out, score marks are to be avoided in the direction of the weld.

The radius is to be checked with a template for accuracy. Four profiles at least are to be checked. A report is to be submitted to the Surveyor.

3.4.5

Before welding is started, a detailed welding procedure specification is to be submitted to the Society covering the weld preparation, welding positions, welding parameters, welding consumables, preheating, post weld heat treatment and inspection procedures. This welding procedure is to be supported by approval tests in accordance with the applicable requirements of materials and welding sections of this Part.

The manufacturer is to maintain records of welding, subsequent heat treatment and inspections traceable to the welds. These records are to be submitted to the Surveyor.

3.4.6

Non destructive tests are to be conducted at least 24 *hours* after completion of the welding. The welds are to be 100% magnetic particle tested and 100% ultrasonic tested. The welds are to be free from cracks, lack of fusion and incomplete penetration. The non destructive tests reports are to be handed over to the Surveyor.

3.4.7

Rudder trunks in materials other than steel are to be specially considered by the Society.

3.4.8

The thickness of the shell or of the bottom plate is to be compatible with the trunk thickness.

## 4. Rudder couplings

### 4.1 General

4.1.1

The couplings are to be designed in such a way as to enable them to transmit the full torque of the rudder stock.

4.1.2

The distance of the bolt axis from the edges of the flange is not to be less than 1.2 *times* the diameter of the bolt. In horizontal couplings, at least 2 bolts are to be arranged forward of the stock axis.

4.1.3

The coupling bolts are to be fitted bolts. The bolts and nuts are to be effectively secured against loosening.

4.1.4

For spade rudders, horizontal couplings according to **4.2** are permitted only where the required thickness of the coupling flanges $t_f$ is less than 50 *mm*, otherwise cone coupling according to **4.4** or **4.5**, as applicable, is to be applied. For spade rudders of the high lift type, only cone coupling according to **4.4** or **4.5**, as applicable, is permitted.

### 4.2 Horizontal couplings

4.2.1

The diameter of coupling bolts, in *mm*, is not to be less than:

$$d_b = 0.62\sqrt{\frac{D^3 k_b}{k_r\ n\ e}}$$

where:

$D$ : Rudder stock diameter according to **6**, in *mm*

$n$ : Total number of bolts, which is not to be less than 6

$e$ : Mean distance of the bolt axes from the centre of bolt system, in *mm*

$k_r$ : Material factor for the rudder stock as defined in **1.4.2**

$k_b$ : Material factor for the bolts, obtained according to **1.4.2**.

4.2.2

The thickness of the coupling flanges, in *mm*, is not to be less than determined by the following formulae:

$$t_f = 0.62\sqrt{\frac{D^3 k_f}{k_r\ n\, e}}\text{, without being less than } 0.9 d_b$$

where:

$k_f$ : Material factor for the coupling flanges, obtained according to **1.4.2**

The thickness of the coupling flanges clear of the bolt holes is not to be less than $0.65 t_f$.

The width of material outside the bolt holes is not to be less than $0.67 d_b$.

4.2.3

The coupling flanges are to be equipped with a fitted key according to *DIN* 6885 or equivalent standard for relieving the bolts.

The fitted key may be dispensed with if the diameter of the bolts is increased by 10%.

4.2.4

Horizontal coupling flanges are to be either forged together with the rudder stock or welded to the rudder stock, according to **10.1.3**.

4.2.5

For the connection of the coupling flanges with the rudder body, see also **10**.

### 4.3 Vertical couplings

4.3.1

The diameter of the coupling bolts, in *mm*, is not to be less than:

$$d_b = \frac{0.81D}{\sqrt{n}}\sqrt{\frac{k_b}{k_r}}$$

where:

$D$, $k_b$, $k_r$, $n$ are defined in **4.2.1**, where $n$ is not to be less than 8.

4.3.2

The first moment of area of the bolts, in $cm^3$, about the centre of the coupling is not to be less than:

$$S = 0.00043D^3$$

4.3.3

The thickness of the coupling flanges, in *mm*, is not to be less than $t_f = d_b$

The width of material outside the bolt holes is not to be less than $0.67d_b$.

### 4.4 Cone couplings with key

4.4.1

Cone couplings should have a taper $c$ on diameter of 1 : 8 to 1 : 12, where $c = (d_0 - d_u)/\ell$ (see **Fig. 8**).

The cone shapes are to fit very exact. The nut is to be carefully secured, e.g. by a securing plate as shown in **Fig. 8**.

4.4.2

The coupling length $\ell$ is to be, in general, not less than $1.5d_0$.

4.4.3

For couplings between stock and rudder a key is to be provided, the shear area of which, in $cm^2$, is not to be less than:

$$a_s = \frac{17.55Q_F}{d_k R_{eH1}}$$

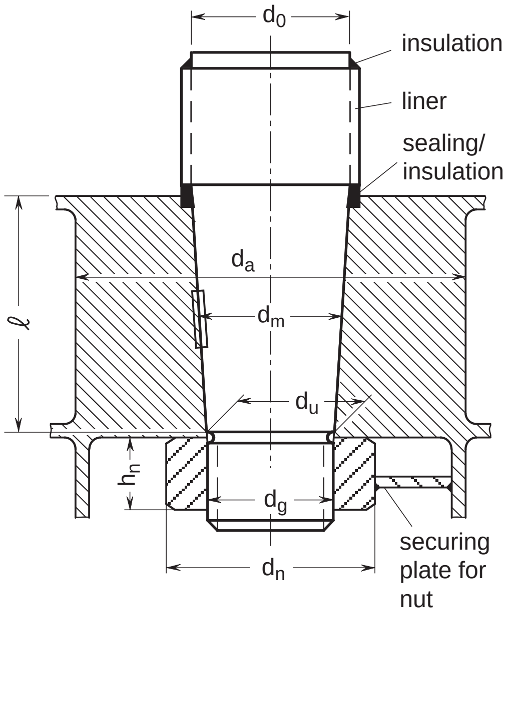
where:

$Q_F$ : Design yield moment of rudder stock, in *N-m* according to **6**

$d_k$ : Diameter of the conical part of the rudder stock, in *mm*, at the key

$R_{eH1}$ : Minimum yield stress of the key material, in $N/mm^2$

Fig. 8 Cone coupling with key



4.4.4

The effective surface area, in $cm^2$, of the key (without rounded edges) between key and rudder stock or cone coupling is not to be less than:

$$a_k = \frac{5Q_F}{d_k R_{eH2}}$$

where:

$R_{eH2}$ : Minimum yield stress of the key, stock or coupling material, in $N/mm^2$, whichever is less.

4.4.5

The dimensions of the slugging nut are to be as follows (see **Fig. 8**):

- height: $h_n = 0.6 d_g$
- outer diameter, the greater value of: $d_n = 1.2\ d_u$ or $d_n = 1.5\ d_g$
- external thread diameter: $d_g = 0.65\ d_0$

4.4.6

It is to be proved that 50% of the design yield moment will be solely transmitted by friction in the cone couplings. This can be done by calculating the required push-up pressure and push-up length according to **4.5.3** for a torsional moment $Q'_F = 0.5 Q_F$ .

## 4.5 Cone couplings with special arrangements for mounting and dismounting the couplings

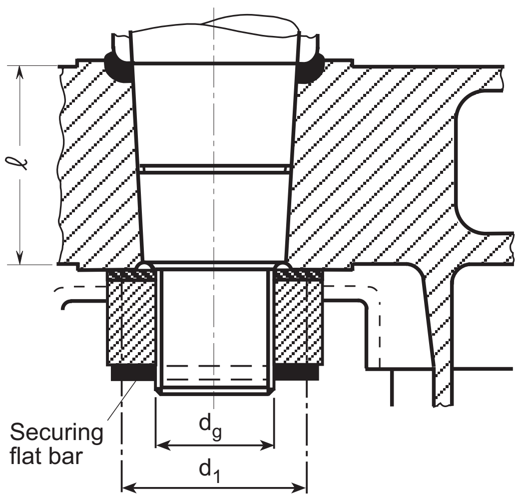
4.5.1

Where the stock diameter exceeds 200 *mm*, the press fit is recommended to be effected by a hydraulic pressure connection. In such cases the cone is to be more slender, $c \approx 1:2$ to $\approx 1:20$ .

4.5.2

In case of hydraulic pressure connections the nut is to be effectively secured against the rudder stock or the pintle. A securing plate for securing the nut against the rudder body is not to be provided (see **Fig. 9**).

Fig. 9 Cone coupling without key




Note: A securing flat bar will be regarded as an effective securing device of the nut, if its shear area, in $mm^2$, is not less than:

$$A_s = \frac{P_s \cdot \sqrt{3}}{R_{eH}}$$

where:

$P_s$ : Shear force, in *N*, as follows:

$$P_S = \frac{P_e}{2} \mu_1 \left( \frac{d_1}{d_g} - 0.6 \right)$$

$P_e$ : Push-up force according to **4.5.3**, in *N*

$\mu_1$ : Frictional coefficient between nut and rudder body, normally $\mu_1 = 0.3$

$d_1$ : Mean diameter of the frictional area between nut and rudder body

$d_g$ : Thread diameter of the nut

$R_{eH}$ : Minimum yield stress, in $N/mm^2$, of the securing flat bar material.

4.5.3 Push-up pressure and push-up length

For the safe transmission of the torsional moment by the coupling between rudder stock and rudder body the push-up length and the push-up pressure are to be determined according to **4.5.4** and **4.5.5**.

4.5.4 Push-up pressure

The push-up pressure is not to be less than the greater of the two following values:

$$p_{req1} = \frac{2Q_F}{d_m^2\, \ell\, \pi\, \mu_0} 10^3$$

$$p_{req2} = \frac{6 \cdot M_b}{\ell^2\, d_m} 10^3$$

where:

$Q_F$ : Design yield moment of rudder stock according to **6**, in *N-m*

$d_m$ : Mean cone diameter, in *mm*

$\ell$ : Cone length, in *mm*

$\mu_0$ : Frictional coefficient, equal to about 0.15

$M_b$ : Bending moment in the cone coupling (e.g. in case of spade rudders), in *N-m*

It has to be proved that the push-up pressure does not exceed the permissible surface pressure in the cone. The permissible surface pressure is to be determined by the following formula:

$$p_{perm} = \frac{0.8 R_{eH}\left(1-\alpha^2\right)}{\sqrt{3+\alpha^4}}$$

where:

$R_{eH}$ : Minimum yield stress, in $N/mm^2$, of the material of the gudgeon

$$\alpha = \frac{d_m}{d_a}$$

$d_m$ : Diameter, in *mm*, as defined in **Fig. 8**

$d_a$ : Outer diameter of the gudgeon (see **Fig. 8**), in *mm*, to be not less than $1.5 d_m$.

4.5.5 Push-up length

The push-up length, in *mm*, is not to be less than:

$$\Delta\ell_1 = \frac{p_{req} d_m}{E\left(\frac{1-\alpha^2}{2}\right)c} + \frac{0.8 R_{tm}}{c}$$

where:

$R_{tm}$ : Mean roughness, in *mm*, taken equal to about 0.01

$c$ : Taper on diameter according to **4.5.1**

The push-up length, in *mm*, is, however, not to be taken greater than:

$$\Delta\ell_2 = \frac{1.6\, R_{eH}\, d_m}{E\, c\sqrt{3+\alpha^4}} + \frac{0.8 R_{tm}}{c}$$

Note: In case of hydraulic pressure connections the required push-up force $P_e$ for the cone, in $N$, may be determined by the following formula:

$$P_e = p_{req}\, d_m \pi\, \ell\left(\frac{c}{2} + 0.02\right)$$

The value 0.02 is a reference for the friction coefficient using oil pressure. It varies and depends on the mechanical treatment and roughness of the details to be fixed.

Where due to the fitting procedure a partial push-up effect caused by the rudder weight is given, this may be taken into account when fixing the required push-up length, subject to approval by the Society.

4.5.6 Push-up pressure for pintle bearings

The required push-up pressure for pintle bearings, in $N/mm^2$, is to be determined by the following formula:

$$p_{req} = 0.4 \frac{B_1 d_0}{d_m^2 \ell}$$

where:

$B_1$ : Supporting force in the pintle bearing, in $N$ (see **Fig. 4**)

$d_m$, $\ell$ : As defined in **4.5.3**

$d_0$ : Pintle diameter, in *mm*, according to **Fig. 8**.

## 5. Rudder body, rudder bearings

### 5.1 Strength of rudder body

5.1.1

The rudder body is to be stiffened by horizontal and vertical webs in such a manner that the rudder body will be effective as a beam. The rudder should be additionally stiffened at the aft edge.

5.1.2

The strength of the rudder body is to be proved by direct calculation according to **3.3**

5.1.3

For rudder bodies without cut-outs the permissible stress are limited to:

- bending stress, in $N/mm^2$, due to $M_R$ defined in **3.3.3**:

  $$\sigma_b = 110$$

- shear stress, in $N$, due to $Q_1$ defined in **3.3.3**:

  $$\tau_t = 50$$

- equivalent stress due to bending and shear:

  $$\sigma_v = \sqrt{\sigma\frac{2}{b} + 3\tau^2} = 120$$

In case of openings in the rudder plating for access to cone coupling or pintle nut the permissible stresses according to **5.1.4** apply. Smaller permissible stress values may be required if the corner radii are less than $0.15h_o$, where $h_o$ is the height of opening.

5.1.4

In rudder bodies with cut-outs (semi-spade rudders) the following stress values are not to be exceeded:

- bending stress, $N/mm^2$, due to $M_R$:

  $$\sigma_b = 90$$

- shear stress, $N/mm^2$, due to $Q_1$:

  $$\tau = 50$$

- torsional stress, $N/mm^2$,due to $M_t$:

  $$\tau_t = 50$$

- equivalent stress, in $N/mm^2$, due to bending and shear and equivalent stress due to bending and torsion:

  $$\sigma_{v1} = \sqrt{\sigma_b^2 + 3\tau^2} = 120$$

  $$\sigma_{v2} = \sqrt{\sigma_b^2 + 3\tau_t^2} = 100$$

where:

$M_R = C_{R2} f_1 + B_1 \frac{f_2}{2}$, in *N-m*

$Q_1 = C_{R2}$, in $N$

$f_1$, $f_2$ : As defined in **Fig. 10**

$\tau_t$ : Torsional stress, in $N/mm^2$, taken equal to:

$$\tau_t = \frac{M_t}{2\,\ell\, h\, t}$$

$M_t = C_{R2}\, e$, in *N-m*

$C_{R2}$ : Partial rudder force, in $N$, of the partial rudder area $A_2$ below the cross section under consideration *e* : Lever for torsional moment, in *m* (horizontal distance between the centre of pressure of area $A_2$ and the centre line a-a of the effective cross sectional area under consideration, see **Fig. 10**. The centre of pressure is to be assumed at $0.33c_2$ aft of the forward edge of area $A_2$, where $c_2$ is the mean breadth of area $A_2$).

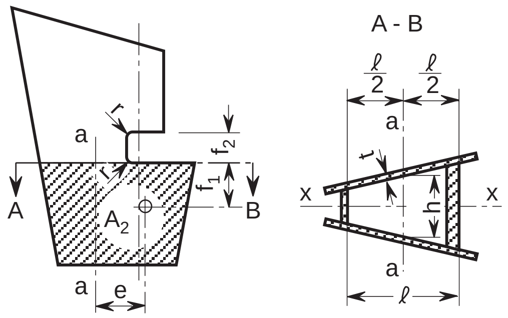
$h, \ell, t$ : Dimensions, in *cm*, as defined in **Fig. 10**.

The distance $\ell$ between the vertical webs should not exceed $1.2h$.

The radii in the rudder plating are not to be less than 4 to 5 *times* the plate thickness, but in no case less than 50 *mm*.

Note: It is recommended to keep the natural frequency of the fully immersed rudder and of local structural components at least 10 % above the exciting frequency of the propeller (number of revolutions × number of blades) or if relevant above higher order.

Fig. 10 Geometry of rudder




## 5.2 Rudder plating

5.2.1

The thickness of the rudder plating, in *mm*, is to be determined according to the following formula:

$$t_P = 1.74a\sqrt{p_R\, k} + 2.5$$

where:

$$p_R = 10T + \frac{C_R}{10^3 A}, \text{ in } kN/m^2$$

*a* : Smaller unsupported width of a plate panel, in *m*.

The influence of the aspect ratio of the plate panels may be taken into account according to **Ch 3**.

However, the thickness is to be not less than the thickness of the shell plating at aft part according to **Ch 9**, **Sec 2**.

Regarding dimensions and welding, **10.1.1** is to be comply with.

5.2.2

For connecting the side plating of the rudder to the webs tenon welding is not to be used. Where application of fillet welding is not practicable, the side plating is to be connected by means of slot welding to flat bars which are welded to the webs.

5.2.3

The thickness of the webs, in *mm*, is not to be less than 70 % of the thickness of the rudder plating according to **5.2.1**, but not less than:

$$t_{\min} = 8\sqrt{k}$$

Webs exposed to seawater are to be dimensioned according to **5.2.1**.

## 5.3 Connections of rudder blade structure with solid parts in forged or cast steel

5.3.1 General

Solid parts in forged or cast steel which ensure the housing of the rudder stock or of the pintle are in general to be connected to the rudder structure by means of two horizontal web plates and two vertical web plates.

5.3.2 Minimum section modulus of the connection with the rudder stock housing

The section modulus of the cross-section of the structure of the rudder blade which is connected with the solid part where the rudder stock is housed, which is made by vertical web plates and rudder plating, is to be not less than that obtained, in $cm^3$, from the following formula:

$$w_S = c_S\, d_1^3 \left( \frac{H_E - H_X}{H_E} \right)^2 \frac{k}{k_1} 10^{-4}$$

where:

$c_S$ : Coefficient, to be taken equal to:

$c_S = 1.0$ if there is no opening in the rudder plating or if such openings are closed by a full penetration welded plate

$c_S = 1.5$ if there is an opening in the considered cross-section of the rudder

$D_1$ : Rudder stock diameter, in *mm*, defined in **3.2.1**

$H_E$ : Vertical distance, in *m*, between the lower edge of the rudder blade and the upper edge of the solid part

$H_X$ : Vertical distance, in *m*, between the considered cross-section and the upper edge of the solid part

$k$, $k_1$ : Material factors, defined for the rudder blade plating and the rudder stock, respectively.

5.3.3 Calculation of the actual section modulus of the connection with the rudder stock housing

The actual section modulus of the cross-section of the structure of the rudder blade which is connected with the solid part where the rudder stock is housed is to be calculated with respect to the symmetrical axis of the rudder.

The breadth of the rudder plating to be considered for the calculation of this actual section modulus is to be not greater than that obtained, in *m*, from the following formula:

$$b = s_V + 2\frac{H_X}{m}$$

where:

$s_V$ : Spacing, in *m*, between the two vertical webs (see **Fig. 11**)

$H_X$ : Distance defined in **5.3.2**

$m$ : Coefficient to be taken, in general, equal to 3.

Where openings for access to the rudder stock nut are not closed by a full penetration welded plate, they are to be deducted (see **Fig. 11**).

5.3.4 Thickness of horizontal web plates

In the vicinity of the solid parts, the thickness of the horizontal web plates, as well as that of the rudder blade plating between these webs, is to be not less than the greater of the values obtained, in *mm*, from the following formulae:

$$t_H = 1.2\, t_P$$

$$t_H = 0.045 \frac{d_S^2}{s_H}$$

where:

$t_P$ : Defined in **5.2.1**

$d_S$ : Diameter, in *mm*, to be taken equal to:

$d_S = D_1$ for the solid part connected to the rudder stock

$d_S = d_a$ for the solid part connected to the pintle

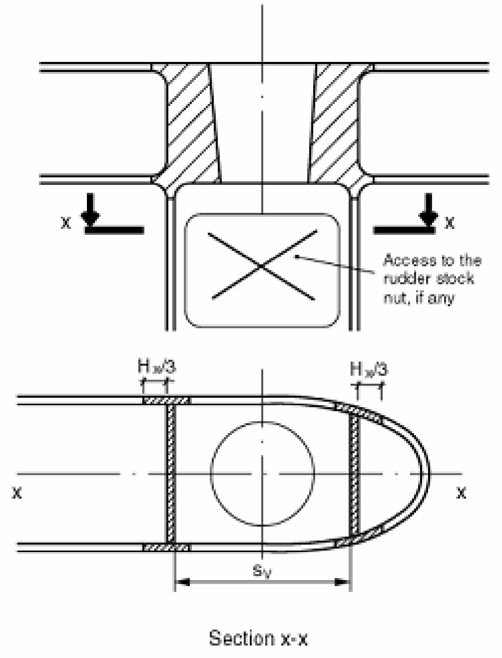
$D_1$ : Rudder stock diameter, in *mm*, defined in **3.2.1**

$d_a$ : Pintle diameter, in *mm*, defined in **5.5.1**

$s_H$ : Spacing, in *mm*, between the two horizontal web plates.

Different thickness may be accepted when justified on the basis of direct calculations submitted to the Society for approval.

Fig. 11 Cross-section of the connection between rudder blade structure and rudder stock housing




5.3.5 Thickness of side plating and vertical web plates welded to the solid part

The thickness of the vertical web plates welded to the solid part where the rudder stock is housed as well as the thickness of the rudder side plating under this solid part is to be not less than the values obtained, in *mm*, from **Table 2**.

Table 2 Thickness of side plating and vertical web plates

| Type of rudder | Thickness of vertical web plates, in *mm* | | Thickness of rudder plating, in *mm* | |
|---|---|---|---|---|
| | Rudder blade without opening | At opening boundary | Rudder blade without opening | Area with opening |
| Rudder supported by sole piece (**Fig 3**) | $1.2t_P$ | $1.6\ t_P$ | $1.2\ t_P$ | $1.4\ t_P$ |
| Semi-spade and spade rudders (**Fig 4** to **Fig 7**) | $1.4\ t_P$ | $2.0\ t_P$ | $1.3\ t_P$ | $1.6\ t_P$ |
| $t_P$ : Defined in **5.2.1** | | | | |

5.3.6 Solid part protrusions

The solid parts are to be provided with protrusions. Vertical and horizontal web plates of the rudder are to be butt welded to these protrusions.

These protrusions are not required when the web plate thickness is less than:

- 10 *mm* for web plates welded to the solid part on which the lower pintle of a semi-spade rudder is housed and for vertical web plates welded to the solid part of the rudder stock coupling of spade rudders
- 20 *mm* for the other web plates.

5.3.7

If the torque is transmitted by a prolonged shaft extended into the rudder, the latter must have the diameter $D_t$ or $D_1$, whichever is greater, at the upper 10 % of the intersection length. Downwards it may be tapered to $0.6D_t$, in spade rudders to 0.4 *times* the strengthened diameter, if sufficient support is provided for.

**5.4 Rudder bearings**

5.4.1

In way of bearings liners and bushes are to be fitted.

Their minimum thickness is equal to:

- $t_{min} = 8\ mm$ for metallic materials and synthetic material

- $t_{min} = 22\ mm$ for lignum material

Where in case of small ships bushes are not fitted, the rudder stock is to be suitably increased in diameter in way of bearings enabling the stock to be re-machined later.

5.4.2

An adequate lubrication is to be provided.

5.4.3

The bearing forces result from the direct calculation mentioned in **3.3**. As a first approximation the bearing force may be determined without taking account of the elastic supports. This can be done as follows:

- normal rudder with two supports:

  The rudder force $C_R$ is to be distributed to the supports according to their vertical distances from the centre of gravity of the rudder area.

- semi-spade rudders:

  support force in the rudder horn, in $N$:

  $$B_1 = C_R \frac{b}{c}$$

  support force in the neck bearing, in $N$:

  $$B_2 = C_R - B_1$$

  For $b$ and $c$ see **Fig. 14**.

5.4.4

The projected bearing surface $A_b$ (“bearing height”× “external diameter of liner”), in $mm^2$, is not to be less than

$$A_b = \frac{B}{q}$$

where:

$B$ : Support force, in $N$

$q$ : Permissible surface pressure according to **Table 3**.

5.4.5

Stainless and wear resistant steels, bronze and hot-pressed bronze-graphit materials have a considerable difference in potential to non-alloyed steel. Respective preventive measures are required.

5.4.6

The bearing height is to be equal to the bearing diameter, however, is not to exceed 1.2 *times* the bearing diameter. Where the bearing depth is less than the bearing diameter, bearing material is to allow higher specific surface pressures.

Table 3 Surface pressure $q$ of bearing materials

| Bearing material | $q$, in $N/mm^2$ |
|---|---|
| Lignum vitae | 2.5 |
| White metal, oil lubricated | 4.5 |
| Synthetic material(1) | 5.5 |
| Steel(2), bronze and hot-pressed bronze-graphite materials | 7.0 |

(1) Synthetic materials to be of approved type.
Surface pressures exceeding 5.5 $N/mm^2$ may be accepted in accordance with bearing manufacturer's specification and tests, but in no case more than 10 $N/mm^2$.

(2) Stainless and wear resistant steel in an approved combination with stock liner. Higher surface pressures than 7 $N/mm^2$ may be accepted if verified by tests.

The wall thickness of pintle bearings in sole piece and rudder horn is to be approximately equal to one fourth of the pintle diameter.

### 5.5 Pintles

5.5.1

Pintles are to have scantlings complying with the conditions given in **4.4** and **4.6**. The pintle diameter, in *mm*, is not to be less than:

$$d_a = 0.35\sqrt{B_1 k_r}$$

where:

$B_1$ : Support force, in $N$

$k_r$ : Material factor defined in **1.4.2**.

5.5.2

The thickness of any liner or bush, in *mm*, is neither to be less than:

$$t = 0.01\sqrt{B_1}$$

nor than the minimum thickness defined in **5.4.1**.

5.5.3

Where pintles are of conical shape, the taper on diameter is to comply with the following:

- $1:8$ to $1:12$, if keyed by slugging nut
- $1:12$ to $1:20$, if mounted with oil injection and hydraulic nut

5.5.4

The pintles are to be arranged in such a manner as to prevent unintentional loosening and falling out.

For nuts and threads the requirements of **4.4.5** and **4.5.2** apply accordingly.

### 5.6 Criteria for bearing clearances

5.6.1

For metallic bearing material the bearing clearance, in *mm*, is to be not less:

$$\frac{d_b}{1000} + 1.0$$

where:

$d_b$ : Inner diameter of bush, in *mm*.

5.6.2

If non-metallic bearing material is applied, the bearing clearance is to be specially determined considering the material's swelling and thermal expansion properties.

5.6.3

The clearance is not to be taken less than 1.5 *mm* on diameter. In case of self lubricating bushes going down below this value can be agreed to on the basis of the manufacturer's specification.

## 6. Design yield moment of rudder stock

### 6.1 General

6.1.1

The design yield moment of the rudder stock is to be determined by the following formula:

$$Q_F = 0.02664\frac{D_t^3}{k_r}$$

$D_t$ : Stock diameter, in *mm*, according to **3.1**.

Where the actual diameter $D_{ta}$ is greater than the calculated diameter $D_t$, the diameter $D_{ta}$ is to be used. However, $D_{ta}$ applied to the above formula need not be taken greater than $1.145D_t$.

## 7. Stopper, locking device

### 7.1 Stopper

7.1.1

The motions of quadrants or tillers are to be limited on either side by stoppers. The stoppers and their foundations connected to the ship's hull are to be of strong construction so that the yield point of the applied materials is not exceeded at the design yield moment of the rudder stock.

### 7.2 Locking device

7.2.1

Each steering gear is to be provided with a locking device in order to keep the rudder fixed at any position. This device as well as the foundation in the ship's hull are to be of strong construction so that the yield point of the applied materials is not exceeded at the design yield moment of the rudder stock as specified in **6**. Where the ship's speed exceeds 12 *knots*, the design yield moment need only be calculated for a stock diameter based on a speed $V_0$ = 12 *knots*.

### 7.3

7.3.1

Regarding stopper and locking device see also the applicable requirements of **15.4.8, Part D**.

## 8. Propeller nozzles

### 8.1 General

8.1.1

The following requirements are applicable to propeller nozzles having an inner diameter of up to 5 *m*. Nozzles with larger diameters will be specially considered.

8.1.2

Special attention is to be given to the support of fixed nozzles at the hull structure.

### 8.2 Design pressure

8.2.1

The design pressure for propeller nozzles, in $kN/m^2$, is to be determined by the following formula:

$$p_d = c\, p_{d0}$$

$$p_{d0} = \varepsilon \frac{N}{Ap}$$

where:

$N$ : Maximum shaft power, in $kW$

$A_p$ : Propeller disc area, in $m^2$, taken equal to:

$$A_p = D^2 \frac{\pi}{4}$$

$D$ : Propeller diameter, in $m$

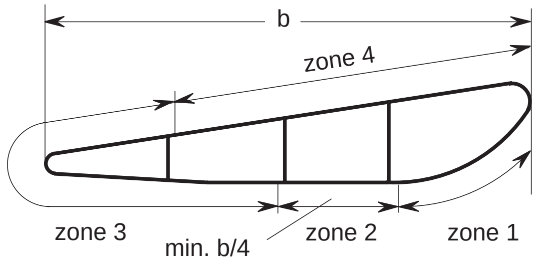
$\varepsilon$ : Factor obtained from the following formula:

$$\varepsilon = 0.21 - 2 \cdot 10^{-4} \frac{N}{A_p}$$ , without being taken less than 0.1

$c$ : Coefficient taken equal to (see **Fig. 12**):

| | |
|---|---|
| $c$ = 1.0 | in zone 2 (propeller zone) |
| $c$ = 0.5 | in zones 1 and 3 |
| $c$ = 0.35 | in zone 4. |

Fig. 12 Zones of propeller nozzle




### 8.3 Plate thickness

8.3.1

The thickness of the nozzle shell plating, in *mm*, is not to be less than:

$t = t_0 + t_k$ , without being taken less than 7.5

where:

$t_0$ : Thickness, in *mm*, obtained from the following formula:

$t_0 = 5a\sqrt{p_d}$

$a$ : Spacing of ring stiffeners, in *m*

$t_k$ : Corrosion allowance, in *mm*, taken equal to:

$t_k = 1.5$ if $t_0 \leq 10$

$t_k = \min\left[0.1\left(\frac{t_0}{\sqrt{k}} + 0.5\right), 3.0\right]$ if $t_0 > 10$

8.3.2

The web thickness of the internal stiffening rings is not to be less than the nozzle plating for zone 3, however, in no case be less than 7.5 *mm*.

### 8.4 Section modulus

8.4.1

The section modulus of the cross section shown in **Fig. 12** around its neutral axis, in $cm^3$, is not to be less than:

$w = n\,d^2\,b\,V_0^2$

where:

$d$ : Inner diameter of nozzle, in *m*

$b$ : Length of nozzle, in *m*

$n$ : Coefficient taken equal to:

$n$ = 1.0 , for rudder nozzles

$n$ = 0.7 , for fixed nozzles.

### 8.5 Welding

8.5.1

The inner and outer nozzle shell plating is to be welded to the internal stiffening rings as far as practicable by double continuous welds. Plug welding is only permissible for the outer nozzle plating.

## 9. Rudder horn and sole piece scantlings

### 9.1 Sole piece

9.1.1

The section modulus of the sole piece related to the $z$-axis, in $cm^3$, is not to be less than:

$$W_z = \frac{B_1\,x\,k}{80}$$

where:

$B_1$ : As defined in **3.3**. For rudders with two supports the support force is approximately $B_1 = C_R/2$, when the elasticity of the sole piece is ignored.

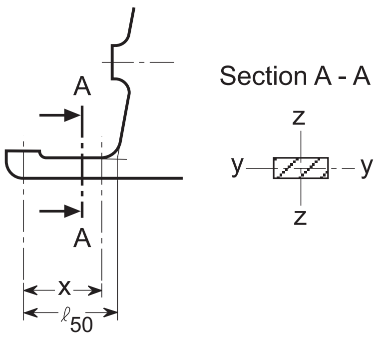
$x$ : Distance, in $m$, of the respective cross section from the rudder axis, with:

$x_{min} = 0.5\,\ell_{50}$

$x_{max} = \ell_{50}$

$\ell_{50}$ : As defined in **Fig. 13** and **3.3.2**.

Fig. 13 Sole piece




9.1.2

The section modulus related to the $y$-axis is not to be less than:

- where no rudder post or rudder axle is fitted

$$W_y = \frac{W_z}{2}$$

- where a rudder post or rudder axle is fitted

$$W_y = \frac{W_z}{3}$$

9.1.3

The sectional area, in $mm^2$, at the location $x = \ell_{50}$ is not to be less than:

$$A_s = \frac{B_1}{48} k$$

9.1.4

The equivalent stress taking into account bending and shear stresses, in $N/mm^2$, at any location within the length $\ell_{50}$ is not to exceed:

$$\sigma_v = \sqrt{\sigma_b^2 + 3\tau^2} = \frac{115}{k}$$

where:

$$\sigma_b = \frac{B_1\, x}{W_z}$$

$$\tau = \frac{B_1}{A_s}$$

**9.2 Rudder horn of semi spade rudders (case of 1-elastic support)**

9.2.1

The distribution of the bending moment, in $N$-$m$, shear force, in $N$, and torsional moment, in $N$-$m$, is to be determined according to the following formulae:

- bending moment: $M_b = B_1 z$

  $M_{b\,max} = B_1 d$

- shear force: $Q = B_1$
- torsional moment: $M_T = B_1\, e_{(z)}$

For determining preliminary scantlings the flexibility of the rudder horn may be ignored and the supporting force $B_1$, in *N*, be calculated according to the following formula:

$$B_1 = C_R \frac{b}{c}$$

where $b$, $c$, $d$, $e_{(z)}$ and $z$ are defined in **Fig. 14** and **Fig. 15**.

$b$ results from the position of the centre of gravity of the rudder area.

Fig. 14 Dimensions of rudder horn

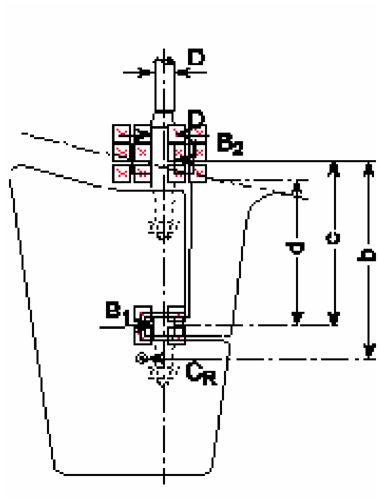
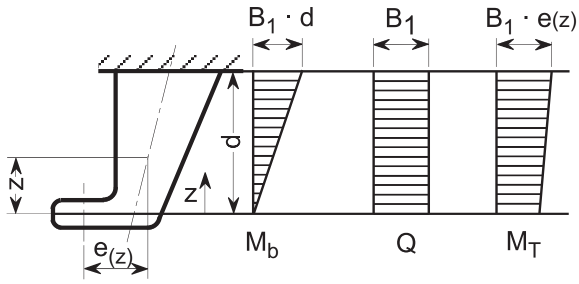



Fig. 15 Rudder horn loads




9.2.2

The section modulus of the rudder horn in transverse direction related to the horizontal $x$-axis is at any location $z$, in $cm^3$, not to be less than:

$$W_x = \frac{M_b\, k}{67}$$

9.2.3

At no cross section of the rudder horn the shear stress, in $N/mm^2$, due to the shear force $Q$ is to exceed the value:

$$\tau = \frac{48}{k}$$

The shear stress, in $N/mm^2$, is to be determined by the following formula:

$$\tau = \frac{B_1}{A_h}$$

where:

$A_h$ : Effective shear area of the rudder horn, in $mm^2$, in $y$-direction

9.2.4

The equivalent stress, in $N/mm^2$, at any location $z$ of the rudder horn is not to exceed the following value:

$$\sigma_v = \sqrt{\sigma_b^2 + 3\left(\tau^2 + \tau_T^2\right)} = \frac{120}{k}$$

where:

$$\sigma_b = \frac{M_b}{W_x}$$

$$\tau_T = \frac{M_T}{2A_T t_h} 10^3$$

$A_T$ : Sectional area, in $mm^2$, enclosed by the rudder horn at the location considered

$t_h$ : Thickness of the rudder horn plating, in *mm*.

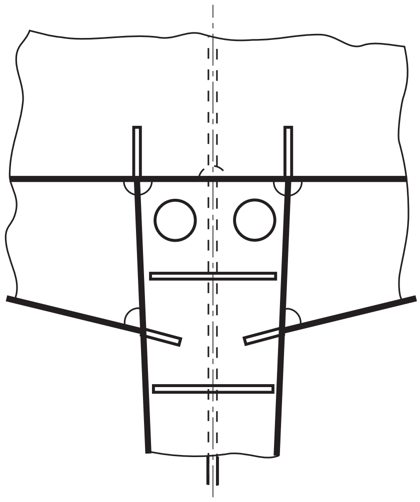
9.2.5

When determining the thickness of the rudder horn plating the provisions of **5.2** to **5.4** are to be complied with. The thickness, in *mm*, is, however, not to be less than $2.4\sqrt{LK}$ .

9.2.6

The rudder horn plating is to be effectively connected to the aft ship structure, e.g. by connecting the plating to longitudinal girders, in order to achieve a proper transmission of forces, see **Fig. 16**.

Fig. 16 Connection of rudder horn to aft ship structure



9.2.7

Transverse webs of the rudder horn are to be led into the hull up to the next deck in a sufficient number and must be of adequate thickness.

9.2.8

Strengthened plate floors are to be fitted in line with the transverse webs in order to achieve a sufficient connection with the hull. The thickness of these plate floors is to be increased by 50% above the bottom thickness determined according to **Ch 6**, **Sec 1** or **Ch 9**, **Sec 2**.

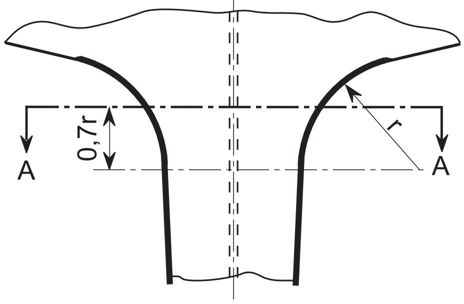
9.2.9

The centre line bulkhead (wash-bulkhead) in the after peak is to be connected to the rudder horn.

9.2.10

Where the transition between rudder horn and shell is curved, about 50% of the required total section modulus of the rudder horn is to be formed by the webs in a section A - A located in the centre of the transition zone, i.e. 0.7*r* above the beginning of the transition zone (See **Fig. 17**).

Fig. 17 Transition between rudder horn and shell




### 9.3 Rudder horn of semi spade rudders (case of 2-conjugate elastic supports)

9.3.1 Bending moment

The bending moment acting on the generic section of the rudder horn is to be obtained, in *N-m*, from the following formulae:

- between the lower and upper supports provided by the rudder horn:

  $M_H = F_{A1}\, z$

- above the rudder horn upper-support:

  $M_H = F_{A1}\, z + F_{A2}\, (z - d_{lu})$

  where:

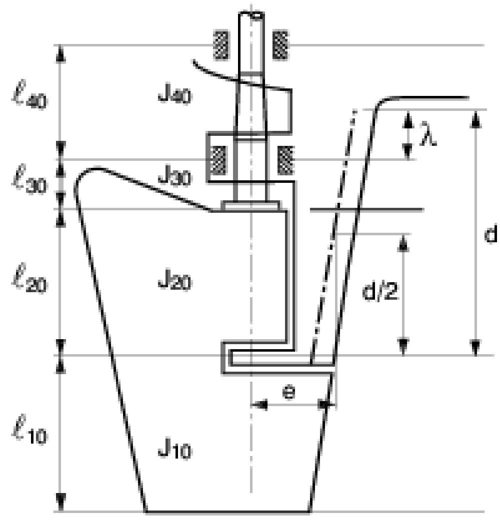
  $F_{A1}$ : Support force at the rudder horn lower-support, in *N*, to be obtained according to **Fig. 5**, and taken equal to $B_1$

  $F_{A2}$ : Support force at the rudder horn upper-support, in *N*, to be obtained according to **Fig. 5**, and taken equal to $B_2$

  $z$ : Distance, in *m*, defined in **Fig. 19**, to be taken less than the distance *d*, in *m*, defined in the same figure

  $d_{lu}$ : Distance, in *m*, between the rudder-horn lower and upper bearings (according to **Fig. 18**, $d_{lu} = d - \lambda$ ).

Fig. 18 Geometrical parameters for the calculation of the bending moment in rudder horn




9.3.2 Shear force

The shear force $Q_H$ acting on the generic section of the rudder horn is to be obtained, in *N*, from the following formulae:

- between the lower and upper rudder horn bearings:

  $Q_H = F_{A1}$

- above the rudder horn upper-bearing:

  $Q_H = F_{A1} + F_{A2}$

  where:

  $F_{A1}, F_{A2}$ : Support forces, in $N$.

9.3.3 Torque

The torque acting on the generic section of the rudder horn is to be obtained, in $N\text{-}m$, from the following formulae:

- between the lower and upper rudder horn bearings:

  $M_T = F_{A1}\, e_{(z)}$

- above the rudder horn upper-bearing:

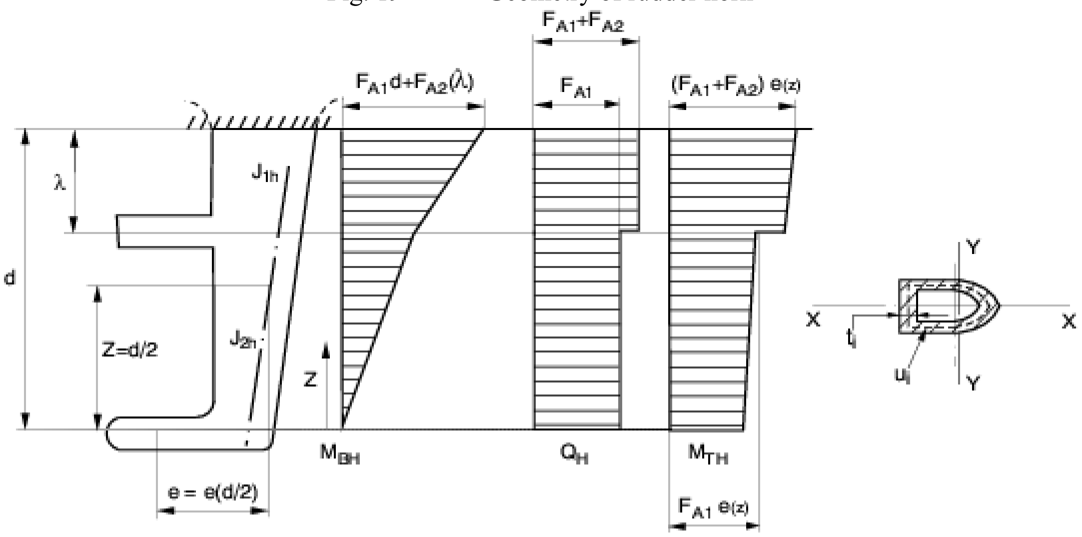
  $M_T = F_{A1}\, e_{(z)} + F_{A2}\, e_{(z)}$

  where:

  $F_{A1}, F_{A2}$ : Support forces, in $N$

  $e_{(z)}$ : Torsion lever, in $m$, defined in **Fig. 19**.

Fig. 19 Geometry of rudder horn




9.3.4 Shear stress calculation

a) For a generic section of the rudder horn, located between its lower and upper bearings, the following stresses are to be calculated:

$\tau_S$ : Shear stress, in $N/mm^2$, to be obtained from the following formula:

$$\tau_S = \frac{F_{A1}}{A_H}$$

$\tau_T$ : Torsional stress, in $N/mm^2$, to be obtained for hollow rudder horn from the following formula:

$$\tau_T = \frac{M_T 10^3}{2F_T t_H}$$

For solid rudder horn, $\tau_T$ is to be considered by the Society on a case by case basis

b) For a generic section of the rudder horn, located in the region above its upper bearing, the following stresses are to be calculated:

$\tau_S$ : Shear stress, in $N/mm^2$, to be obtained from the following formula:

$$\tau_S = \frac{F_{A1} + F_{A2}}{A_H}$$

$\tau_T$ : Torsional stress, in $N/mm^2$, to be obtained for hollow rudder horn from the following formula:

$$\tau_T = \frac{M_T 10^3}{2F_T t_H}$$

For solid rudder horn, $\tau_T$ is to be considered by the Society on a case by case basis

where:

$F_{A1}, F_{A2}$ : Support forces, in $N$

$A_H$ : Effective shear sectional area of the rudder horn, in $mm^2$, in $y$-direction

$M_T$ : Torque, in $N\text{-}m$

$F_T$ : Mean of areas enclosed by outer and inner boundaries of the thin walled section of rudder horn, in $m^2$

$t_H$ : Plate thickness of rudder horn, in $mm$. For a given cross section of the rudder horn, the maximum value of $\tau_T$ is obtained at the minimum value of $t_H$.

9.3.5 Bending stress calculation

For the generic section of the rudder horn within the length $d$, defined in **Fig. 14**, the following stresses are to be calculated:

$\sigma_B$ : Bending stress, in $N/mm^2$, to be obtained from the following formula:

$$\sigma_B = \frac{M_H}{W_X}$$

$M_H$ : Bending moment at the section considered, in $N\text{-}m$

$W_X$ : Section modulus, in $cm^3$, around the $X$-axis (see **Fig. 19**).

9.3.6 General remarks

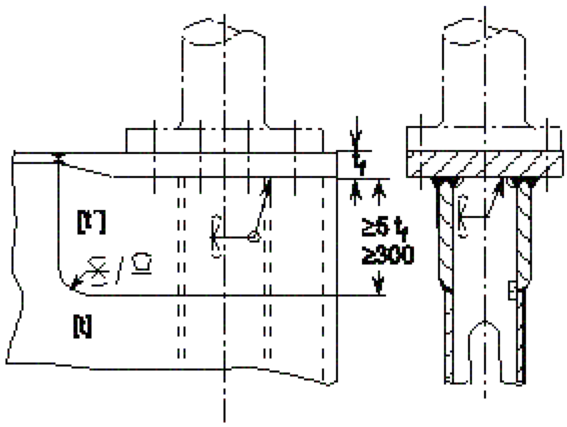
Requirements **9.2.5** to **9.2.10** also apply to rudder horn with 2-conjugate elastic supports.

## 10. Rudder coupling flanges

10.1.1

Unless forged or cast steel flanges with integrally forged or cast welding flanges are used, horizontal rudder coupling flanges are to be joined to the rudder body by plates of graduated thickness and full penetration single or double-bevel welds as prescribed in **Ch 11**, **Sec 1** (see **Fig. 20**).

Fig. 20 Horizontal rudder coupling flanges




$t$ = thickness of rudder plating, in $mm$

$t_f$ = actual flange thickness, in $mm$

$t' = \dfrac{t_f}{3} + 5$, in $mm$, where $t_f < 50$ $mm$

$t' = 3\sqrt{t_f}$, in $mm$, where $t_f \geq 50$ $mm$

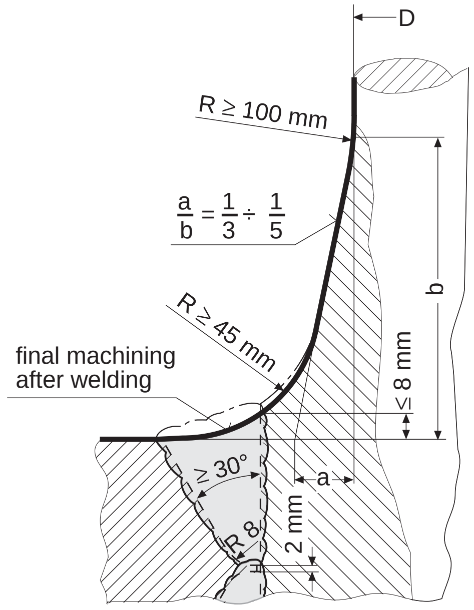
10.1.2

Allowance is to be made for the reduced strength of the coupling flange in the thickness direction. In case of doubt, proof by calculation of the adequacy of the welded connection shall be produced.

10.1.3

The welded joint between the rudder stock (with thickened collar) and the flange is to be made in accordance with **Fig. 21**.

Fig. 21 Welded joint between rudder stock and coupling flange




## 11. Azimuth propulsion system

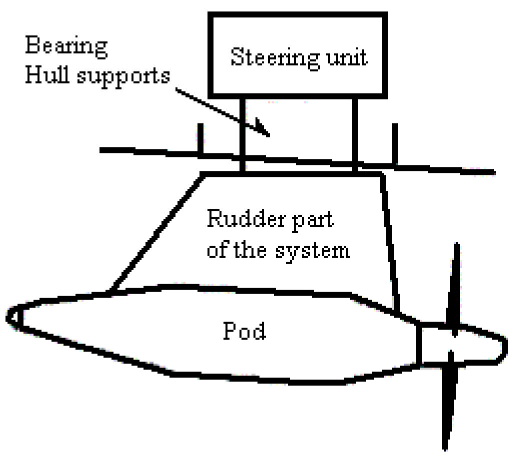
### 11.1 General

11.1.1 Arrangement

The azimuth propulsion system is constituted by the following sub-systems (see **Fig. 22**):

- the steering unit
- the bearing
- the hull supports
- the rudder part of the system
- the pod, which contains the electric motor in the case of a podded propulsion system.

Fig. 22 Azimuth propulsion system




11.1.2 Application

The requirements of this Article apply to the scantlings of the hull supports, the rudder part and the pod.

The steering unit and the bearing are to comply with the relevant requirements of **Part D**.

11.1.3 Operating conditions

The maximum angle at which the azimuth propulsion system can be oriented on each side when the ship navigates at its maximum speed is to be specified by the Designer. Such maximum angle is generally to be less than 35° on each side.

In general, orientations greater than this maximum angle may be considered by the Society for azimuth propulsion systems during manoeuvres, provided that the orientation values together with the relevant speed values are submitted to the Society for approval.

### 11.2 Arrangement

11.2.1 Plans to be submitted

In addition to the plans showing the structural arrangement of the pod and the rudder part of the system, the plans showing the arrangement of the azimuth propulsion system supports are to be submitted to the Society for approval. The scantlings of the supports and the maximum loads which act on the supports are to be specified in these drawings.

11.2.2 Locking device

The azimuth propulsion system is to be mechanically lockable in a fixed position, in order to avoid rotations of the system and propulsion in undesirable directions in the event of damage.

### 11.3 Design loads

11.3.1

The lateral pressure to be considered for scantling of plating and ordinary stiffeners of the azimuth propulsion system is to be determined for an orientation of the system equal to the maximum angle at which the azimuth propulsion system can be oriented on each side when the ship navigates at its maximum speed.

- The total force which acts on the azimuth propulsion system is to be obtained by integrating the lateral pressure on the external surface of the system.
- The calculations of lateral pressure and total force are to be submitted to the Society for information.

### 11.4 Plating

11.4.1 Plating of the rudder part of the azimuth propulsion system

The thickness of plating of the rudder part of the azimuth propulsion system is to be not less than that obtained, in *mm*, from the formulae in **5.2.1**, in which the term $C_R/A$ is to be replaced by the lateral pressure calculated according to **11.3**.

11.4.2 Plating of the pod

The thickness of plating of the pod is to be not less than that obtained, in *mm*, from the formulae in **Ch 6**, **Sec 1** or **Ch 9**, **Sec 2**, where the lateral pressure is to be calculated according to **11.3**.

11.4.3 Webs

The thickness of webs of the rudder part of the azimuth propulsion system is to be determined according to **5.2.3**, where the lateral pressure is to be calculated according to **11.3**.

### 11.5 Ordinary stiffeners

11.5.1 Ordinary stiffeners of the pod

The scantlings of ordinary stiffeners of the pod are to be not less than those obtained from the formulae in **Ch 6**, **Sec 2** or **Ch 9**, **Sec 2**, where the lateral pressure is to be calculated according to **11.3**.

### 11.6 Primary supporting members

11.6.1 Analysis criteria

The scantlings of primary supporting members of the azimuth propulsion system are to be obtained by the Designer through direct calculations, to be carried out according to the following requirements:

- the structural model is to include the pod, the rudder part of the azimuth propulsion system, the bearing and the hull supports
- the boundary conditions are to represent the connections of the azimuth propulsion system to the hull structures
- the loads to be applied are those defined in **11.6.2**.
- The direct calculation analyses (structural model, load and stress calculation, strength checks) carried out by the Designer are to be submitted to the Society for information.

11.6.2 Loads

The following loads are to be considered by the Designer in the direct calculation of the primary supporting members of the azimuth propulsion system:

- gravity loads
- buoyancy
- maximum loads calculated for an orientation of the system equal to the maximum angle at which the azimuth propulsion system can be oriented on each side when the ship navigates at its maximum speed
- maximum loads calculated for the possible orientations of the system greater than the maximum angle at the relevant speed (see **11.1.3**)
- maximum loads calculated for the crash stop of the ship obtained through inversion of the propeller rotation
- maximum loads calculated for the crash stop of the ship obtained through a 180° rotation of the pod.

11.6.3 Strength check

It is to be checked that the Von Mises equivalent stress $\sigma_E$ in primary supporting members, calculated, in $N/mm^2$, for the load cases defined in **11.6.2**, is in compliance with the following formula:

$$\sigma_E \le \sigma_{ALL}$$

where:

$\sigma_{ALL}$ : Allowable stress, in $N/mm^2$, to be taken equal to the lesser of the following values:

$$\sigma_{ALL} = 0.275 R_m$$

$$\sigma_{ALL} = 0.55 R_{eH}$$

**11.7 Hull supports of the azimuth propulsion system**

11.7.1 Analysis criteria

The scantlings of hull supports of the azimuth propulsion system are to be obtained by the Designer through direct calculations, to be carried out in accordance with the requirements in **11.6.1**.

11.7.2 Loads

The loads to be considered in the direct calculation of the hull supports of the azimuth propulsion system are those specified in **11.6.2**.

11.7.3 Strength check

It is to be checked that the Von Mises equivalent stress $\sigma_E$ in hull supports, in $N/mm^2$, calculated for the load cases defined in **11.6.2**, is in compliance with the following formula:

$$\sigma_E \le \sigma_{ALL}$$

where:

$\sigma_{ALL}$ : Allowable stress, in $N/mm^2$, equal to $65 / k_r$

$k_r$ : Material factor, defined in **1.4.2**

Values of $\sigma_E$ greater than $\sigma_{ALL}$ may be accepted by the Society on a case by case basis, depending on the localisation of $\sigma_E$ and on the type of direct calculation analysis.

# Section 2 BULWARKS AND GUARD RAILS

## 1. General

### 1.1 Introduction

1.1.1

The requirements of this Section apply to the arrangement of bulwarks and guard rails provided at boundaries of the freeboard deck, superstructure decks and tops of the first tier of deckhouses located on the freeboard deck.

### 1.2 General

1.2.1

Efficient bulwarks or guard rails are to be fitted at the boundaries of all exposed parts of the freeboard deck and superstructure decks directly attached to the freeboard deck, as well as the first tier of deckhouses fitted on the freeboard deck and the superstructure ends.

1.2.2

The height of the bulwarks or guard rails is to be at least 1 *m* from the deck. However, where their height would interfere with the normal operation of the ship, a lesser height may be accepted, if adequate protection is provided and subject to any applicable statutory requirement.

1.2.3

Where superstructures are connected by trunks, open rails are to be fitted for the whole length of the exposed parts of the freeboard deck.

1.2.4

In type *B*-100 ships, open rails on the weather parts of the freeboard deck for at least half the length of the exposed parts are to be fitted.

Alternatively, freeing ports complying with **Ch 9**, **Sec 6**, **5.5.2** are to be fitted.

1.2.5

In ships with bulwarks and trunks of breadth not less than 0.6*B*, which are included in the calculation of freeboard, open rails on the weather parts of the freeboard deck in way of the trunk for at least half the length of the exposed parts are to be fitted.

Alternatively, freeing ports complying with **Ch 9**, **Sec 6**, **5.3.1** are to be fitted.

1.2.6

In ships having superstructures which are open at either or both ends, adequate provision for freeing the space within such superstructures is to be provided.

1.2.7

The freeing port area in the lower part of the bulwarks is to be in compliance with the applicable requirements of **Ch 9**, **Sec 6**, **5**.

## 2. Bulwarks

### 2.1 General

2.1.1

As a rule, plate bulwarks are to be stiffened at the upper edge by a suitable bar and supported either by stays or plate brackets spaced not more than 2.0 *m* apart.

The free edge of the stay or the plate bracket is to be stiffened.

Stays and brackets of bulwarks are to be aligned with the beams located below or are to be connected to them by means of local transverse stiffeners.

As an alternative, the lower end of the stay and bracket may be supported by a longitudinal stiffener.

2.1.2

In type *B*-60 and *B*-100 ships, the spacing forward of 0.07*L* from the fore end of brackets and stays is to be not greater than 1.2 *m*.

2.1.3

Where bulwarks are cut completely, the scantlings of stays or brackets at ends are to be increased with respect to those given in **2.2**.

2.1.4

As a rule, bulwarks are not to be connected either to the upper edge of the sheerstrake plate or to the stringer plate.

Failing this, the detail of the connection will be examined by the Society.

### 2.2 Scantlings

2.2.1

The gross thickness of bulwarks on the freeboard deck not exceeding 1 *m* in height is to be not less than 6.5 *mm*.

Where the height of the bulwark is equal to or greater than 1.8 *m*, its thickness is to be equal to that calculated for the side of a superstructure situated in the same location as the bulwark.

For bulwarks between 1 *m* and 1.8 *m* in height, their thickness is to be calculated by linear interpolation.

2.2.2

Bulwark plating and stays are to be adequately strengthened in way of eye plates used for shrouds or other tackles in use for cargo gear operation, as well as in way of hawser holes or fairleads provided for mooring or towing.

2.2.3

At the ends of partial superstructures and for the distance over which their side plating is tapered into the bulwark, the latter is to have the same thickness as the side plating. Where openings are cut in the bulwark at these positions, adequate compensation is to be provided either by increasing the thickness of the plating or by other suitable means.

2.2.4

The gross section modulus of stays in way of the lower part of the bulwark is to be not less than the value obtained, in $cm^3$, from the following formula:

$$w = 77 s h_B^2$$

where:

$s$ : Spacing of stays, in *m*

$h_B$ : Height of bulwark, in *m*, measured from the top of the deck plating to the upper edge.

The actual section of the connection between stays and deck structures is to be taken into account when calculating the above section modulus.

To this end, the bulb or face plate of the stay may be taken into account only where welded to the deck; in this case the beam located below is to be connected by double continuous welding.

For stays with strengthening members not connected to the deck, the calculation of the required section modulus is considered by the Society on a case by case basis.

At the ends of the ship, where the bulwark is connected to the sheerstrake, an attached plating having a width not exceeding 600 *mm* may also be included in the calculation of the actual gross section modulus of stays.

2.2.5

Openings in bulwarks are to be arranged so that the protection of the crew is to be at least equivalent to that provided by the horizontal courses in **3.1.2**.

For this purpose, vertical rails or bars spaced approximately 230 *mm* apart may be accepted in lieu of rails or bars arranged horizontally.

2.2.6

In the case of ships intended for the carriage of timber deck cargoes, the specific provisions of the freeboard regulations are to be complied with.

## 3. Guard rails

### 3.1 General

3.1.1

Where guard rails are provided, the upper edge of sheerstrake is to be kept as low as possible.

3.1.2

The opening below the lowest course is to be not more than 230 *mm*. The other courses are to be not more than 380 *mm* apart.

3.1.3

In the case of ships with rounded gunwales or sheerstrake, the stanchions are to be placed on the flat part of the deck.

3.1.4

Fixed, removable or hinged stanchions are to be fitted about 1.5 *m* apart. At least every third stanchion is to be supported by a bracket or stay.

Removable or hinged stanchions are to be capable of being locked in the upright position.

3.1.5

Wire ropes may only be accepted in lieu of guard rails in special circumstances and then only in limited lengths. Wires are to be made taut by means of turnbuckles.

3.1.6

Chains may only be accepted in short lengths in lieu of guard rails if they are fitted between two fixed stanchions and/or bulwarks.

# Section 3 EQUIPMENT

**Symbols**

For symbols not defined in this Section, refer to **Ch 1**, **Sec 4**.

*EN* : Equipment number defined in **2.1**

## 1. General

### 1.1 General

1.1.1

The requirements in this Section apply to temporary mooring of a ship within or near harbour, or in a sheltered area, when the ship is awaiting a berth, the tide, etc.

Therefore, the equipment complying with the requirements in this Section is not intended for holding a ship off fully exposed coasts in rough weather or for stopping a ship which is moving or drifting.

1.1.2

The equipment complying with the requirements in this Section is intended for holding a ship in good holding ground, where the conditions are such as to avoid dragging of the anchor. It should be noted that, in poor holding ground, the holding power of the anchors is to be significantly reduced.

1.1.3

The equipment number *EN* formula for anchoring equipment required here under is based on an assumed current speed of 2.5 *m/s*, wind speed of 25 *m/s* and a scope of chain cable between 6 and 10, the scope being the ratio between length of chain paid out and water depth.

1.1.4

It is assumed that under normal circumstances a ship will use one anchor only.

## 2. Equipment number

### 2.1 Equipment number

2.1.1 General

All ships are to be provided with equipment in anchors and chain cables (or ropes according to **3.3.5**), to be obtained from **Table 1**, based on their equipment number *EN*.

In general, stockless anchors are to be adopted.

For ships with *EN* greater than 16000, the determination of the equipment will be considered by the Society on a case by case basis.

Table 1 Equipment

| Equipment number *EN* $A < EN \le B$ | | Stockless anchors | | Stud link chain cables for anchors | | | |
|---|---|---|---|---|---|---|---|
| | | $N$ [(1)] | Mass per anchor, in *kg* | Total length, in *m* | Diameter, in *mm* | | |
| *A* | *B* | | | | Grade 1 | Grade 2 | Grade 3 |
| 50 | 70 | 2 | 180 | 220.0 | 14.0 | 12.5 | |
| 70 | 90 | 2 | 240 | 220.0 | 16.0 | 14.0 | |
| 90 | 110 | 2 | 300 | 247.5 | 17.5 | 16.0 | |
| 110 | 130 | 2 | 360 | 247.5 | 19.0 | 17.5 | |
| 130 | 150 | 2 | 420 | 275.0 | 20.5 | 17.5 | |
| 150 | 175 | 2 | 480 | 275.0 | 22.0 | 19.0 | |
| 175 | 205 | 2 | 570 | 302.5 | 24.0 | 20.5 | |

| Equipment number $EN$ $A < EN \leq B$ | | Stockless anchors | | Stud link chain cables for anchors | | | |
|---|---|---|---|---|---|---|---|
| | | $N$ [(1)] | Mass per anchor, in $kg$ | Total length, in $m$ | Diameter, in $mm$ | | |
| $A$ | $B$ | | | | Grade 1 | Grade 2 | Grade 3 |
| 205 | 240 | 3 | 660 | 302.5 | 26.0 | 22.0 | 20.5 |
| 240 | 280 | 3 | 780 | 330.0 | 28.0 | 24.0 | 22.0 |
| 280 | 320 | 3 | 900 | 357.5 | 30.0 | 26.0 | 24.0 |
| 320 | 360 | 3 | 1020 | 357.5 | 32.0 | 28.0 | 24.0 |
| 360 | 400 | 3 | 1140 | 385.0 | 34.0 | 30.0 | 26.0 |
| 400 | 450 | 3 | 1290 | 385.0 | 36.0 | 32.0 | 28.0 |
| 450 | 500 | 3 | 1440 | 412.5 | 38.0 | 34.0 | 30.0 |
| 500 | 550 | 3 | 1590 | 412.5 | 40.0 | 34.0 | 30.0 |
| 550 | 600 | 3 | 1740 | 440.0 | 42.0 | 36.0 | 32.0 |
| 600 | 660 | 3 | 1920 | 440.0 | 44.0 | 38.0 | 34.0 |
| 660 | 720 | 3 | 2100 | 440.0 | 46.0 | 40.0 | 36.0 |
| 720 | 780 | 3 | 2280 | 467.5 | 48.0 | 42.0 | 36.0 |
| 780 | 840 | 3 | 2460 | 467.5 | 50.0 | 44.0 | 38.0 |
| 840 | 910 | 3 | 2640 | 467.5 | 52.0 | 46.0 | 40.0 |
| 910 | 980 | 3 | 2850 | 495.0 | 54.0 | 48.0 | 42.0 |
| 980 | 1060 | 3 | 3060 | 495.0 | 56.0 | 50.0 | 44.0 |
| 1060 | 1140 | 3 | 3300 | 495.0 | 58.0 | 50.0 | 46.0 |
| 1140 | 1220 | 3 | 3540 | 522.5 | 60.0 | 52.0 | 46.0 |
| 1220 | 1300 | 3 | 3780 | 522.5 | 62.0 | 54.0 | 48.0 |
| 1300 | 1390 | 3 | 4050 | 522.5 | 64.0 | 56.0 | 50.0 |
| 1390 | 1480 | 3 | 4320 | 550.0 | 66.0 | 58.0 | 50.0 |
| 1480 | 1570 | 3 | 4590 | 550.0 | 68.0 | 60.0 | 52.0 |
| 1570 | 1670 | 3 | 4890 | 550.0 | 70.0 | 62.0 | 54.0 |
| 1670 | 1790 | 3 | 5250 | 577.5 | 73.0 | 64.0 | 56.0 |
| 1790 | 1930 | 3 | 5610 | 577.5 | 76.0 | 66.0 | 58.0 |
| 1930 | 2080 | 3 | 6000 | 577.5 | 78.0 | 68.0 | 60.0 |
| 2080 | 2230 | 3 | 6450 | 605.0 | 81.0 | 70.0 | 62.0 |
| 2230 | 2380 | 3 | 6900 | 605.0 | 84.0 | 73.0 | 64.0 |
| 2380 | 2530 | 3 | 7350 | 605.0 | 87.0 | 76.0 | 66.0 |
| 2530 | 2700 | 3 | 7800 | 632.5 | 90.0 | 78.0 | 68.0 |
| 2700 | 2870 | 3 | 8300 | 632.5 | 92.0 | 81.0 | 70.0 |
| 2870 | 3040 | 3 | 8700 | 632.5 | 95.0 | 84.0 | 73.0 |
| 3040 | 3210 | 3 | 9300 | 660.0 | 97.0 | 84.0 | 76.0 |
| 3210 | 3400 | 3 | 9900 | 660.0 | 100.0 | 87.0 | 78.0 |
| 3400 | 3600 | 3 | 10500 | 660.0 | 102.0 | 90.0 | 78.0 |
| 3600 | 3800 | 3 | 11100 | 687.5 | 105.0 | 92.0 | 81.0 |
| 3800 | 4000 | 3 | 11700 | 687.5 | 107.0 | 95.0 | 84.0 |
| 4000 | 4200 | 3 | 12300 | 687.5 | 111.0 | 97.0 | 87.0 |
| 4200 | 4400 | 3 | 12900 | 715.0 | 114.0 | 100.0 | 87.0 |
| 4400 | 4600 | 3 | 13500 | 715.0 | 117.0 | 102.0 | 90.0 |

(1) See **3.2.4**.

2.1.2 Equipment number

The equipment number $EN$ is to be obtained from the following formula:

$EN = \Delta^{2/3} + 2\,h\,B + 0.1\,A$

where:.

$\Delta$ : Moulded displacement of the ship, in $t$, to the summer load waterline

$h$ : Effective height, in m, from the summer load waterline to the top of the uppermost house, to be obtained in accordance with the following formula:

$h = a + \Sigma\, h_n$

When calculating h, sheer and trim are to be disregarded

$a$ : Freeboard amidships from the summer load waterline to the upper deck, in $m$

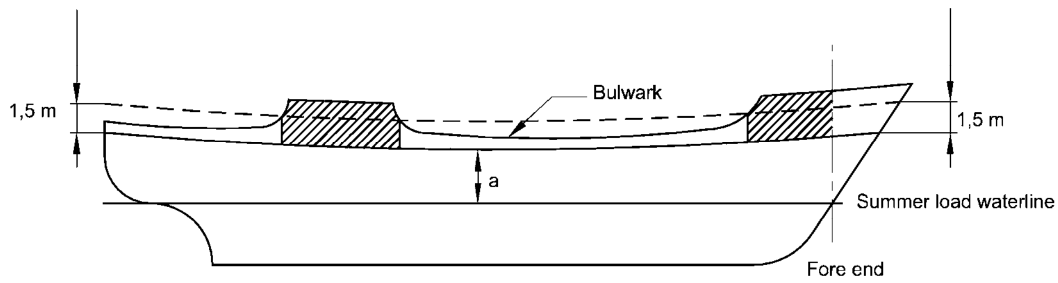
$h_n$ : Height, in $m$, at the centreline of tier "$n$" of superstructures or deckhouses having a breadth greater than $B/4$. Where a house having a breadth greater than $B/4$ is above a house with a breadth of $B/4$ or less, the upper house is to be included and the lower ignored

$A$ : Area, in $m^2$, in profile view, of the parts of the hull, superstructures and houses above the summer load waterline which are within the length $L$ and also have a breadth greater than $B/4$

Fixed screens or bulwarks 1.5 $m$ or more in height are to be regarded as parts of houses when determining $h$ and $A$. In particular, the hatched area shown in **Fig. 1** is to be included.

The height of hatch coamings and that of any deck cargo, such as containers, may be disregarded when determining $h$ and $A$.

Fig. 1 Effective area of bulwarks or fixed screen to be included in the equipment number




## 3. Equipment

### 3.1 General

3.1.1

All anchoring equipment, towing bitts, mooring bollards, fairlead cleats and eyebolts are to be so constructed and attached to the hull that, in use up to design loads, the integrity of the ship will not be impaired.

3.1.2

The anchoring arrangement is to be such as to prevent the cable from being damaged and fouled. Adequate arrangement is to be provided to secure the anchor under all operational conditions.

### 3.2 Anchors

3.2.1 General

The scantlings of anchors are to be in compliance with the following requirements.

Anchors are to be constructed and tested in compliance with approved plans.

3.2.2 Ordinary anchors

The required mass for each anchor is to be obtained from **Table 1**.

The individual mass of a main anchor may differ by ±7% from the mass required for each anchor, provided that the total mass of anchors is not less than the total mass required in **Table 1**.

The mass of the head of an ordinary stockless anchor, including pins and accessories, is to be not less than 60% of the total mass of the anchor.

Where a stock anchor is provided, the mass of the anchor, excluding the stock, is to be not less than 80% of the mass required in Tab 1 for a stockless anchor. The mass of the stock is to be not less than 25% of the mass of the anchor without the stock but including the connecting shackle.

3.2.3 High and very high holding power anchors

High holding power (*HHP*) and very high holding power (*VHHP*) anchors, i.e. anchors for which a holding power higher than that of ordinary anchors has been proved according to the applicable requirements of **Chapter 2, Part L**, do not require testing regarding to **3.2.5** or **3.2.6**, respectively.

Where *HHP* or *VHHP* anchors are used as bower anchors, the mass of each anchor is to be not less than 75% or 50%, respectively, of that required for ordinary stockless anchors in **Table 1**.

The mass of *VHHP* anchors is to be, in general, less than or equal to 1500 *kg*.

3.2.4 Third anchor

Where three anchors are provided, two are to be connected to their own chain cables and positioned on board always ready for use.

The third anchor is intended as a spare and is not required for the purpose of classification.

3.2.5 Test for high holding power anchors approval

For approval and/or acceptance as a *HHP* anchor, comparative tests are to be performed on various types of sea bottom.

Such tests are to show that the holding power of the *HHP* anchor is at least twice the holding power of an ordinary stockless anchor of the same mass.

For approval and/or acceptance as a *HHP* anchor of a whole range of mass, such tests are to be carried out on anchors whose sizes are, as far as possible, representative of the full range of masses proposed. In this case, at least two anchors of different sizes are to be tested. The mass of the maximum size to be approved is to be not greater than 10 *times* the maximum size tested. The mass of the smallest is to be not less than 0.1 *times* the minimum size tested.

3.2.6 Test for very high holding power anchors approval

For approval and/or acceptance as a *VHHP* anchor, comparative tests are to be performed at least on three types of sea bottom: soft mud or silt, sand or gravel and hard clay or similar compounded material. Such tests are to show that the holding power of the *VHHP* anchor is to be at least four *times* the holding power of an ordinary stockless anchor of the same mass or at least twice the holding power of a previously approved *HHP* anchor of the same mass. The holding power test load may be less than or equal to the proof load of the anchor, where deemed as appropriate by the Society.

For approval and/or acceptance as a *VHHP* anchor of a whole range of mass, such tests are to be carried out on anchors whose sizes are, as far as possible, representative of the full range of masses proposed. In this case, at least three anchors of different sizes are to be tested, relevant to the bottom, middle and top of the mass range.

3.2.7 Specification for test on high holding power and very high holding power anchors

Tests are generally to be carried out from a tug. Shore based tests may be accepted by the Society on a case by case basis.

Alternatively, sea trials by comparison with a previous approved anchor of the same type (*HHP* or *VHHP*) of the one to be tested may be accepted by the Society on a case by case basis.

For each series of sizes, the two anchors selected for testing (ordinary stockless and *HHP* anchors for testing *HHP* anchors, ordinary stockless and *VHHP* anchors or, when ordinary stockless anchors are not available, *HHP* and *VHHP* anchors for testing *VHHP* anchors) are to have the same mass.

The length of chain cable connected to each anchor, having a diameter appropriate to its mass, is to be such that the pull on the shank remains practically horizontal. For this purpose a value of the ratio between the length of the chain cable paid out and the water depth equal to 10 is considered normal. A lower value of this ratio may be accepted by the Society on a case by case basis.

Three tests are to be carried out for each anchor and type of sea bottom.

The pull is to be measured by dynamometer; measurements based on the *RPM*/bollard pull curve of tug may, however, be accepted instead of dynamometer readings.

Note is to be taken where possible of the stability of the anchor and its ease of breaking out.

### 3.3 Chain cables for anchors

3.3.1 Material

The chain cables are classified as grade 1, 2 or 3 depending on the type of steel used and its manufacture.

The characteristics of the steel used and the method of manufacture of chain cables are to be approved by the Society for each manufacturer. The material from which chain cables are manufactured and the completed chain cables themselves are to be tested in accordance with the applicable requirements of **Part K** and **Chapter 3, Part L**.

Chain cables made of grade 1 may not be used with high holding power and very high holding power anchors.

3.3.2 Scantlings of stud link chain cables

The mass and geometry of stud link chain cables, including the links, are to be in compliance with the requirements in the applicable requirements of **Chapter 3, Part L**.

The diameter of stud link chain cables is to be not less than the value in **Table 1**.

3.3.3 Studless link chain cables

For ships with *EN* less than 90, studless short link chain cables may be accepted by the Society as an alternative to stud link chain cables, provided that the equivalence in strength is based on proof load, defined in the applicable requirements of **Chapter 3, Part L** and that the steel grade of the studless chain is equivalent to the steel grade of the stud chains it replaces, as defined in **3.3.1**.

3.3.4 Chain cable arrangement

Chain cables are to be made by lengths of 27.5 *m* each, joined together by Dee or lugless shackles.

The total length of chain cable, required in **Table 1**, is to be divided in approximately equal parts between the two anchors ready for use.

Where different arrangements are provided, they are considered by the Society on a case by case basis.

Where the ship may anchor in areas with current speed greater than 2.5 *m/s*, the Society may require a length of heavier chain cable to be fitted between the anchor and the rest of the chain in order to enhance anchor bedding.

### 3.4 Attachment pieces

3.4.1 General

Where the lengths of chain cable are joined to each other by means of shackles of the ordinary Dee type, the anchor may be attached directly to the end link of the first length of chain cable by a Dee type end shackle.

A detachable open link in two parts riveted together may be used in lieu of the ordinary Dee type end shackle; in such case the open end link with increased diameter, defined in **3.4.2**, is to be omitted.

Where the various lengths of chain cable are joined by means of lugless shackles and therefore no special end and increased diameter links are provided, the anchor may be attached to the first length of chain cable by a special pear-shaped lugless end shackle or by fitting an attachment piece.

3.4.2 Scantlings

The diameters of the attachment pieces, in *mm*, are to be not less than the values indicated in **Table 2**.

Attachment pieces may incorporate the following items between the increased diameter stud link and the open end link:

- swivel, having a diameter equal to 1.2*d*
- increased stud link, having a diameter equal to 1.1*d*

Where different compositions are provided, they will be considered by the Society on a case by case basis.

Table 2 Diameters of attachment pieces

| Attachment piece | Diameter, in *mm* |
|---|---|
| End shackle | 1.4*d* |
| Open end link | 1.2*d* |
| Increased stud link | 1.1*d* |
| Common stud link | *d* |
| Lugless shackle | *d* |
| where:<br>*d* : Diameter, in *mm*, of the common link. | |

3.4.3 Material

Attachment pieces, joining shackles and end shackles are to be of such material and design as to provide strength equivalent to that of the attached chain cable, and are to be tested in accordance with the applicable requirements of the applicable requirements of **Chapter 3, Part L**.

3.4.4 Spare attachment pieces

A spare pear-shaped lugless end shackle or a spare attachment piece is to be provided for use when the spare anchor is fitted in place.

### 3.5 Towlines and mooring lines

3.5.1 General

The towlines having the characteristics defined in **Table 3** are intended as those belonging to the ship to be towed by a tug or another ship.

3.5.2 Materials

Towlines and mooring lines may be of wire, natural or synthetic fibre or a mixture of wire and fibre.

The breaking loads defined in **Table 3** refer to steel wires or natural fibre ropes.

Steel wires and fibre ropes are to be tested in accordance with the applicable requirements in the applicable requirements of **Chapter 4** and **Chapter 5, Part L**.

3.5.3 Steel wires

Steel wires are to be made of flexible galvanised steel and are to be of types defined in **Table 4**.

Where the wire is wound on the winch drum, steel wires to be used with mooring winches may be constructed with an independent metal core instead of a fibre core. In general such wires are to have not less than 186 threads in addition to the metallic core.

Table 3 Towlines and mooring lines

| Equipment number $EN$ $A < EN \leq B$ | | Towline [1] | | Mooring lines | | |
|---|---|---|---|---|---|---|
| $A$ | $B$ | Minimum length, in $m$ | Breaking load, in $kN$ | $N$ [2] | Length of each line, in $m$ | Breaking load, in $kN$ |
| 50 | 70 | 180 | 98.1 | 3 | 80 | 34 |
| 70 | 90 | 180 | 98.1 | 3 | 100 | 37 |
| 90 | 110 | 180 | 98.1 | 3 | 110 | 39 |
| 110 | 130 | 180 | 98.1 | 3 | 110 | 44 |
| 130 | 150 | 180 | 98.1 | 3 | 120 | 49 |
| 150 | 175 | 180 | 98.1 | 3 | 120 | 54 |
| 175 | 205 | 180 | 112 | 3 | 120 | 59 |
| 205 | 240 | 180 | 129 | 4 | 120 | 64 |
| 240 | 280 | 180 | 150 | 4 | 120 | 69 |
| 280 | 320 | 180 | 174 | 4 | 140 | 74 |
| 320 | 360 | 180 | 207 | 4 | 140 | 78 |
| 360 | 400 | 180 | 224 | 4 | 140 | 88 |
| 400 | 450 | 180 | 250 | 4 | 140 | 98 |
| 450 | 500 | 180 | 277 | 4 | 140 | 108 |
| 500 | 550 | 190 | 306 | 4 | 160 | 123 |
| 550 | 600 | 190 | 338 | 4 | 160 | 132 |
| 600 | 660 | 190 | 371 | 4 | 160 | 147 |
| 660 | 720 | 190 | 406 | 4 | 160 | 157 |
| 720 | 780 | 190 | 441 | 4 | 170 | 172 |
| 780 | 840 | 190 | 480 | 4 | 170 | 186 |
| 840 | 910 | 190 | 518 | 4 | 170 | 201 |
| 910 | 980 | 190 | 550 | 4 | 170 | 216 |
| 980 | 1060 | 200 | 603 | 4 | 180 | 230 |
| 1060 | 1140 | 200 | 647 | 4 | 180 | 250 |
| 1140 | 1220 | 200 | 692 | 4 | 180 | 270 |
| 1220 | 1300 | 200 | 739 | 4 | 180 | 284 |
| 1300 | 1390 | 200 | 786 | 4 | 180 | 309 |
| 1390 | 1480 | 200 | 836 | 4 | 180 | 324 |
| 1480 | 1570 | 220 | 889 | 5 | 190 | 324 |
| 1570 | 1670 | 220 | 942 | 5 | 190 | 333 |
| 1670 | 1790 | 220 | 1024 | 5 | 190 | 353 |
| 1790 | 1930 | 220 | 1109 | 5 | 190 | 378 |
| 1930 | 2080 | 220 | 1168 | 5 | 190 | 402 |
| 2080 | 2230 | 240 | 1259 | 5 | 200 | 422 |
| 2230 | 2380 | 240 | 1356 | 5 | 200 | 451 |
| 2380 | 2530 | 240 | 1453 | 5 | 200 | 481 |
| 2530 | 2700 | 260 | 1471 | 6 | 200 | 481 |
| 2700 | 2870 | 260 | 1471 | 6 | 200 | 490 |
| 2870 | 3040 | 260 | 1471 | 6 | 200 | 500 |
| 3040 | 3210 | 280 | 1471 | 6 | 200 | 520 |
| 3210 | 3400 | 280 | 1471 | 6 | 200 | 554 |
| 3400 | 3600 | 280 | 1471 | 6 | 200 | 588 |
| 3600 | 3800 | 300 | 1471 | 6 | 200 | 612 |
| 3800 | 4000 | 300 | 1471 | 6 | 200 | 647 |

| Equipment number *EN* $A < EN \leq B$ | | Towline [1] | | Mooring lines | | |
|---|---|---|---|---|---|---|
| *A* | *B* | Minimum length, in *m* | Breaking load, in *kN* | $N^{(2)}$ | Length of each line, in *m* | Breaking load, in *kN* |
| 4000 | 4200 | 300 | 1471 | 7 | 200 | 647 |
| 4200 | 4400 | 300 | 1471 | 7 | 200 | 657 |
| 4400 | 4600 | 300 | 1471 | 7 | 200 | 667 |
| 4600 | 4800 | 300 | 1471 | 7 | 200 | 677 |
| 4800 | 5000 | 300 | 1471 | 7 | 200 | 686 |
| 5000 | 5200 | 300 | 1471 | 8 | 200 | 686 |
| 5200 | 5500 | 300 | 1471 | 8 | 200 | 696 |
| 5500 | 5800 | 300 | 1471 | 8 | 200 | 706 |
| 5800 | 6100 | 300 | 1471 | 9 | 200 | 706 |
| 6100 | 6500 | | | 9 | 200 | 716 |
| 6500 | 6900 | | | 9 | 200 | 726 |
| 6900 | 7400 | | | 10 | 200 | 726 |
| 7400 | 7900 | | | 11 | 200 | 726 |
| 7900 | 8400 | | | 11 | 200 | 735 |
| 8400 | 8900 | | | 12 | 200 | 735 |
| 8900 | 9400 | | | 13 | 200 | 735 |
| 9400 | 10000 | | | 14 | 200 | 735 |
| 10000 | 10700 | | | 15 | 200 | 735 |
| 10700 | 11500 | | | 16 | 200 | 735 |
| 11500 | 12400 | | | 17 | 200 | 735 |
| 12400 | 13400 | | | 18 | 200 | 735 |
| 13400 | 14600 | | | 19 | 200 | 735 |
| 14600 | 16000 | | | 21 | 200 | 735 |

(1) The towline is not compulsory. It is recommended for ships having length not greater than 180 *m*.
(2) See **3.5.4**.

Table 4 Steel wire composition

| Breaking load *BL*, in *kN* | Steel wire components | | |
|---|---|---|---|
| | Number of threads | Ultimate tensile strength of threads, in $N/mm^2$ | Composition of wire |
| *BL*< 216 | 72 | 1420 ÷ 1570 | 6 strands with 7-fibre core |
| 216 < *BL*< 490 | 144 | 1570 ÷ 1770 | 6 strands with 7-fibre core |
| *BL* > 490 | 216 or 222 | 1770 ÷ 1960 | 6 strands with 1-fibre core |

3.5.4 Number of mooring lines

When the breaking load of each mooring line is greater than 490 *kN*, either a greater number of mooring lines than those required in **Table 3** having lower strength, or a lower number of mooring lines than those required in **Table 3** having greater strength may be used, provided the total breaking load of all lines aboard the ship is greater than the value defined in **Table 3**.

In any case, the number of lines is to be not less than 6 and the breaking load of each line is to be greater than 490 *kN*.

3.5.5 Length of mooring lines

The length of individual mooring lines may be reduced by up to 7% of the length defined in **Table 3**, provided that the total length of mooring lines is greater than that obtained by adding the lengths of the individual lines defined in **Table 3**.

3.5.6 Equivalence between the breaking loads of synthetic and natural fibre ropes

Generally, fibre ropes are to be made of polyamide or other equivalent synthetic fibres.

The equivalence between the breaking loads of synthetic fibre ropes $B_{LS}$ and of natural fibre ropes $B_{LN}$ is to be obtained, in *kN*, from the following formula:

$$B_{LS} = 7.4\,\delta\left(B_{LN}\right)^{8/9}$$

where:

$\delta$ : Elongation to breaking of the synthetic fibre rope, to be assumed not less than 30%.

### 3.6 Hawse pipes

3.6.1

Hawse pipes are to be built according to sound marine practice.

Their position and slope are to be so arranged as to create an easy lead for the chain cables and efficient housing for the anchors, where the latter are of the retractable type, avoiding damage to the hull during these operations.

For this purpose chafing lips of suitable form with ample lay-up and radius adequate to the size of the chain cable are to be provided at the shell and deck. The shell plating in way of the hawse pipes is to be reinforced as necessary.

3.6.2

In order to obtain an easy lead of the chain cables, the hawse pipes may be provided with rollers. These rollers are to have a nominal diameter not less than 10 *times* the size of the chain cable where they are provided with full imprints, and not less than 12 *times* its size where provided with partial imprints only.

3.6.3

All mooring units and accessories, such as timbler, riding and trip stoppers are to be securely fastened to the Surveyor's satisfaction.

### 3.7 Windlass

3.7.1 General

The windlass, which is generally single, is to be power driven and suitable for the size of chain cable and the mass of the anchors.

The windlass is to be fitted in a suitable position in order to ensure an easy lead of the chain cables to and through the hawse pipes. The deck in way of the windlass is to be suitably reinforced.

3.7.2 Assumptions for the calculation of the continuous duty pull

The calculation of the continuous duty pull $P_C$ that the windlass unit prime mover is to be able to supply is based on the following assumptions:

- ordinary stockless anchors
- wind force equal to 6 on Beaufort Scale
- water current velocity 3 *knots*
- anchorage depth 100 *m*
- $P_C$ includes the influences of buoyancy and hawse pipe efficiency; the latter is assumed equal to 70%
- the anchor masses assumed are those defined in the applicable requirements of **Chapter 2, Part L**, excluding tolerances
- only one anchor is assumed to be raised at a time.

Owing to the buoyancy, the chain masses assumed are smaller than those defined in the applicable requirements of **Chapter 3, Part L**, and are obtained, per unit length of the chain cable, in *kg/m*, from the following formula:

$$m_L = 0.0218\, d^2$$

where $d$ is the chain cable diameter, in *mm*.

3.7.3 Calculation of the continuous duty pull

According to the assumptions in **3.7.2**, the windlass unit prime mover is to be able to supply for a least 30 *minutes* a continuous duty pull $P_C$ to be obtained, in *kN*, from **Table 5**.

Table 5 Continuous duty pull

| Material of chain cables | Continuous duty pull, in *kN* |
|---|---|
| Mild steel | $P_C = 0.0375\ d^2$ |
| High tensile strength steel | $P_C = 0.0425\ d^2$ |
| Very high tensile strength steel | $P_C = 0.0475\ d^2$ |
| where:<br>$d$ : Chain cable diameter, in *mm*. | |

3.7.4 Temporary overload capacity

The windlass unit prime mover is to provide the necessary temporary overload capacity for breaking out the anchor.

The temporary overload capacity, or short term pull, is to be not less than 1.5 *times* the continuous duty pull $P_C$ and it is to be provided for at least two minutes.

The speed in this overload period may be lower than the nominal speed specified in **3.7.5**.

3.7.5 Nominal hoisting speed

The nominal speed of the chain cable when hoisting the anchor and cable, to be assumed as an average speed, is to be not less than 0.15 *m/s*.

The speed is to be measured over two shots of chain cable during the entire trip; the trial is to commence with 3 shots (82.5 *m*) of chain fully submerged.

3.7.6 Windlass brake

A windlass brake is to be provided having sufficient capacity to stop the anchor and chain cable when paying out the latter with safety, in the event of failure of the power supply to the prime mover. Windlasses not actuated by steam are also to be provided with a non-return device.

A windlass with brakes applied and the cable lifter declutched is to be able to withstand a pull of 45% of the breaking load of the chain without any permanent deformation of the stressed parts or brake slip.

3.7.7 Chain stoppers

Where a chain stopper is fitted, it is to be able to withstand a pull of 80% of the breaking load of the chain.

Where a chain stopper is not fitted, the windlass is to be able to withstand a pull of 80% of the breaking load of the chain without any permanent deformation of the stressed part or brake slip.

3.7.8 Green sea loads

Where the height of the exposed deck in way of the item is less than $0.1L$ or 22 *m* above the summer load waterline, whichever is the lesser, the securing devices of windlasses located within the forward quarter length of the ship are to resist green sea forces.

The green sea pressure and associated areas are to be taken equal to (see **Fig. 2**):

- 200 $kN/m^2$ normal to the shaft axis and away from the forward perpendicular, over the projected area in this direction
- 150 $kN/m^2$ parallel to the shaft axis and acting both inboard and outboard separately, over the multiple of $f$ *times* the projected area in this direction,

where:

$f$ : Coefficient taken equal to

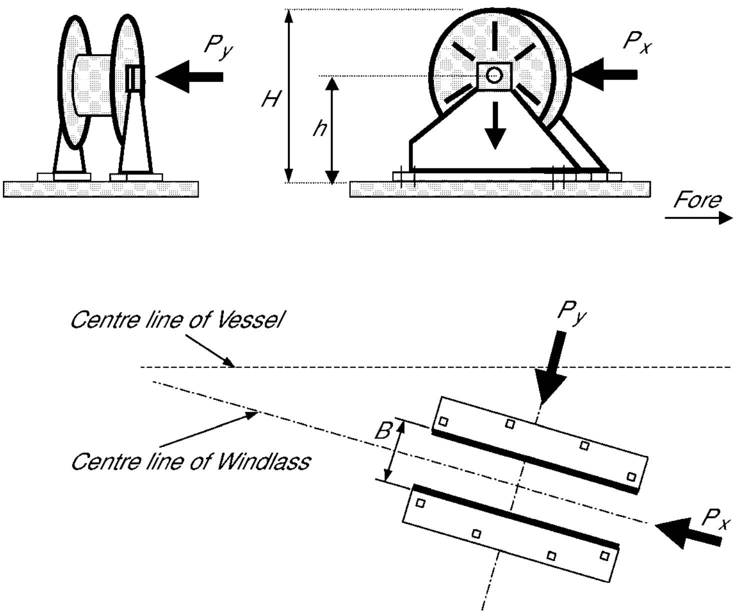
$$f = 1 + \frac{B}{H}$$ , but not greater than 2.5

$B$ : Width of windlass measured parallel to the shaft axis,

$H$ : Overall height of windlass.

Where mooring winches are integral with the anchor windlass, they are to be considered as part of the windlass.

Fig. 2 Direction of forces and weight




*Note: Py to be examined from both inboard and outboard directions separately - see [3.7.8]. The sign convention for yi is reversed when Py is from the opposite direction as shown.*

3.7.9 Forces in the securing devices of windlasses due to green sea loads

Forces in the bolts, chocks and stoppers securing the windlass to the deck are to be calculated by considering the green sea loads specified in **3.7.8**.

The windlass is supported by $N$ bolt groups, each containing one or more bolts (see also **Fig. 3**).

The axial force $R_i$ in bolt group (or bolt) $i$, positive in tension, is to be obtained, in $kN$, from the following formulae:

- $R_{xi} = P_{\mathrm{x}}\, h_{xi}\, A_{\mathrm{i}} / I_{\mathrm{x}}$
- $R_{yi} = P_y\, h_{yi}\, A_i / I_y$
- $R_i = R_{xi} + R_{yi} - R_{si}$

where:

$P_x$ : Force, in $kN$, acting normal to the shaft axis

$P_y$ : Force, in $kN$, acting parallel to the shaft axis, either inboard or outboard, whichever gives the greater force in bolt group $i$

$H$ : Shaft height, in $cm$, above the windlass mounting

$x_i$, $y_i$ : $X$ and $Y$ co-ordinates, in $cm$, of bolt group $i$ from the centroid of all $N$ bolt groups, positive in the direction opposite to that of the applied force

$A_i$ : Cross-sectional area, in $cm^2$, of all bolts in group $i$

$I_x$, $I_y$ : Inertias, for $N$ bolt groups, equal to:

$I_x = \Sigma A_i\, x_i^2$

$I_y = \Sigma A_i\, y_i^2$

$R_i$ : Static reaction force, in $kN$, at bolt group $i$, due to weight of windlass.

Shear forces $F_{xi}$, $F_{yi}$ applied to the bolt group $i$, and the resultant combined force $F_i$ are to be obtained, in $kN$, from the following formulae:

- $F_{xi} = (P_i - \alpha\, g\, M) / N$
- $F_{yi} = (P_y - \alpha\, g\, M) / N$
- $F_i = (F_{xi}^2 + F_{yi}^2)^{0.5}$

where:

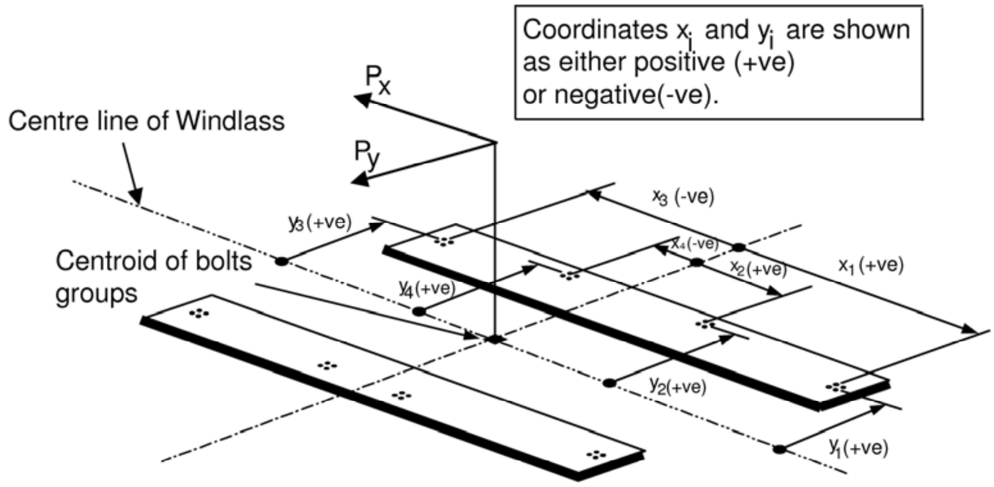
$\alpha$ : Coefficient of friction, to be taken equal to 0.5

$M$ : Mass, in $t$, of windlass

$N$ : Number of bolt groups.

Axial tensile and compressive forces and lateral forces calculated according to these requirements are also to be considered in the design of the supporting structure.

Fig. 3 Sign convention




3.7.10 Strength criteria for windlass subject to anchor and chain loads

The stresses on the parts of the windlass, its frame and stopper are to be less than the yield stress of the material used.

For the calculation of the above stresses, special attention is to be paid to:

- stress concentrations in keyways and other stress raisers
- dynamic effects due to sudden starting or stopping of the prime mover or anchor chain
- calculation methods and approximation.

3.7.11 Strength criteria for securing devices of windlass

Tensile axial stresses in the individual bolts in each bolt group *i* are to be calculated according to the requirements specified in **3.7.9**. The horizontal forces $F_{xi}$ and $F_{yi}$, to be calculated according to the requirements specified in **3.7.9**, are normally to be reacted by shear chocks.

Where "fitted" bolts are designed to support these shear forces in one or both directions, the equivalent Von Mises stress $\sigma$, in $N/mm^2$, in the individual bolt is to comply with following formula:

$$\sigma \leq 0.5\ \sigma_{BPL}$$

where $\sigma_{BPL}$ is the stress in the bolt considered as being loaded by the proof strength.

Where pourable resins are incorporated in the holding down arrangements, due account is to be taken in the calculations.

3.7.12 Connection with deck

The windlass, its frame and the stoppers are to be efficiently bedded to the deck.

### 3.8 Chain stoppers

3.8.1

A chain stopper is generally to be fitted between the windlass and the hawse pipe in order to relieve the windlass of the pull of the chain cable when the ship is at anchor. A chain stopper is to be capable of withstanding a pull of 80% of the breaking load of the chain cable. The deck at the chain stopper is to be suitably reinforced.

For the same purpose, a piece of chain cable may be used with a rigging screw capable of supporting the weight of the anchor when housed in the hawse pipe or a chain tensioner. Such arrangements are not to be considered as chain stoppers.

3.8.2

Where the windlass is at a distance from the hawse pipes and no chain stoppers are fitted, suitable arrangements are to be provided to lead the chain cables to the windlass.

### 3.9 Chain locker

3.9.1

The capacity of the chain locker is to be adequate to stow all chain cable equipment and provide an easy direct lead to the windlass.

3.9.2

Where two chains are used, the chain lockers are to be divided into two compartments, each capable of housing the full length of one line.

3.9.3

The inboard ends of chain cables are to be secured to suitably reinforced attachments in the structure by means of end shackles, whether or not associated with attachment pieces.

Generally, such attachments are to be able to withstand a force not less than 15% of the breaking load of the chain cable.

In an emergency, the attachments are to be easily released from outside the chain locker.

3.9.4

Where the chain locker is arranged aft of the collision bulkhead, its boundary bulkheads are to be watertight and a drainage system is to be provided.

### 3.10 Fairleads and bollards

3.10.1

Fairleads and bollards of suitable size and design are to be fitted for towing, mooring and warping operations.

## 4. Shipboard fittings and supporting hull structures associated with towing and mooring

### 4.1 Towing

4.1.1 Application

The strength of shipboard fittings i.e. bollard/fairlead/chocks used for normal and emergency operations at bow, sides and stern and their supporting structures are to comply with the following requirements

4.1.2 Arrangement

Shipboard fittings for towing are to be located on longitudinals, beams and/or girders, which are part of the deck construction so as to facilitate efficient distribution of the towing load.

4.1.3 Load considerations

The design loads to be used are specified below:

- not less than twice the maximum breaking strength of the tow line anticipated to be used throughout the service life of the ship is to be applied
- where the maximum breaking strength of the tow line is not provided then, at a minimum, 1.5 *times* the breaking strength of the tow line for the ship's corresponding *EN*, as specified in **Table 3**, is to be applied.

4.1.4 Deck fittings

The size of deck fittings is to be in accordance with a standard (e.g. *ISO*3913 Shipbuilding Welded Steel Bollards) recognized by the Society. The design loads used to assess deck fittings and their attachment to the ship are to be in accordance with **4.1.3**.

4.1.5 Supporting hull structure: arrangement

Arrangement of the reinforced members (carling) beneath is to consider any variation of direction (laterally and vertically) of the towing forces, which is to be not less than the design load obtained from **4.1.3**, acting through the arrangement of connection to the towing fittings.

4.1.6 Supporting hull structure: acting point of towing force

The acting point of the towing force on deck fittings is to be taken at the attachment point of a towing line.

4.1.7 Supporting hull structure: allowable stress

The allowable bending stress is to be taken equal to $R_{eH}$ of the material used.

The allowable shearing stress is to be taken as the 0.6 $R_{eH}$ of the material used.

4.1.8 Safe working load (*SWL*)

The *SWL* is not to exceed one half of the design load per **4.1.3**.

The *SWL* of each fitting that is designed for normal and emergency use with tugs is to be marked (by weld bead) on the deck fittings for towing.

The *SWL* is to be noted in the general arrangement drawing or other information available on board for the guidance of Master.

**4.2 Mooring**

4.2.1 Application

Mooring equipment is to be in accordance with the requirements in **4.1**.

However when mooring equipment is only used for mooring, the requirements in **4.2** can be substituted for the corresponded requirements in **4.1**.

4.2.2 Load considerations

The design loads to be used are specified below:

- not less than twice the maximum breaking strength of the mooring line anticipated to be used throughout the service life of the ship is to be applied.
- where the maximum breaking strength of the tow line is not provided then, at a minimum, 1.5 *times* the breaking strength of the tow line for the ship's corresponding *EN*, as specified in **Table 3**, is to be applied.