RULES

PUBLICATION 122/P

REQUIREMENTS FOR BALTIC ICE CLASS AND POLAR CLASS FOR SHIPS UNDER PRS SUPERVISION

July
2024

Publications P (Additional Rule Requirements) issued by Polski Rejestr Statków complete or extend the Rules and are mandatory where applicable.

GDAŃSK

Publication 122/P – Requirements for Baltic Ice Class and Polar Class for Ships under PRS Supervision – July 2024 are based on Finnish-Swedish Ice Class regulations 2017, as amended together with Guidelines for the Application of the Finnish-Swedish Ice Class Rules and IACS Unified Requirements I series.

The *Publication* was approved by the PRS Board on 27 June 2024 and enters into force on 1 July 2024.

---

* *Polish Register of Shipping means Polski Rejestr Statków S.A., seated in Gdańsk, al. gen. Józefa Hallera 126, 80-416 Gdańsk, Poland,* registered in the Register of Entrepreneurs of the National Court Register, under entry number 0000019880. Polish Register of Shipping, its affiliates and subsidiaries, their respective officers, employees or agents are, individually and collectively, referred to as Polish Register of Shipping or as PRS for short.

PRS/RP, 05/2024

CONTENTS

Page

1 APPLICATION

1.1 The requirements specified in Part I apply to ships with class notation **L1A**, **L1**, **L2** and **L3** intended occasionally or primarily for navigation in the northern Baltic in winter or areas with similar ice conditions.

1.1.1 Application of the requirements specified in Part I to ships other than cargo vessels of 2000 TDW or smaller ships like tugs or workboats is to be subject of separate consideration by PRS.

1.1.2 Technical requirements for ships intended to perform icebreaking within its operational profile where notation mark ICEBREAKER is to be assigned are subject of separate consideration by PRS in each particular case assuming compliance with the requirements specified in Part II.

1.2 The requirements specified in Part II apply to ships with class notation **PC1**, **PC2**, **PC3**, **PC4**, **PC5**, **PC6**, **PC7** intended for navigation in ice-infested polar waters. All ships operating in polar waters shall also fulfil requirements of International Code for Ships Operating in Polar Waters (Polar Code), adopted by IMO Res. MSC.385 (94). SOLAS ships are required to keep on board Polar Ship Certificate and Polar Water Operational Manual (PWOM) specific to the ship operating in polar waters.

1.3 The requirements specified in Supplement apply to ships with class notation **(L4)** and E intended for navigation in waters with light ice conditions and light localized drift ice, in river mouths and coastal areas.

1.4 The scope for additional ice class notation specifies requirements for hull strength, machinery systems and equipment.

2 GENERAL REQUIREMENTS

2.1 The decision on necessity to provide ice strengthenings in ship rests with the Owner.

2.2 The requirements specified in the present Chapter shall be regarded as supplementary to the basic requirements set forth in Chapters 1÷17 *Part II – Hull* and *Part VI – Machinery Installations and Refrigerating Plants.*

2.3 Definition[1]

First-year ice means sea ice of not more than one winter growth developing from young ice with thickness from 0.3 m to 2.0 m. May be subdivided into thin first-year ice/white ice, medium first-year ice and thick first-year ice.

Thin first-year ice means first-year ice 30 cm to 70 cm thick.

Medium first-year ice means first-year ice of 70 cm to 120 cm thick.

Thick first-year ice means first-year ice over 120 cm thick.

Old ice means sea ice which has survived at least one summer's melt, typical thickness up to 3 m or more. It is subdivided into residual first-year ice, second-year ice and multi-year ice.

Residual first-year ice means first-year ice that has survived the summer's melt and is now in the new cycle of growth. It is 30 to 180 cm thick depending on the region where it was in summer. After 1 January (in the Southern hemisphere after 1 July), this is called second-year ice.

[1] Ice description is based on WMO Sea Ice Nommenclature.

Second-year ice means old ice which has survived only one summer's melt, typical thickness up to 2.5 m and sometimes more. Because it is thicker than first-year ice, it stands higher out of the water. Ridged features as a result of melting during the preceding summer attain a smoothed rounded shape. In summer, numerous puddles of extending irregular shape form on its surface. Bare ice patches and puddles are usually greenish-blue.

Multi-year ice means old ice up to 3 m or more thick that has survived at least two summer's melt. Hummocks are even smoother than in second-year ice and attain a look of mounds and hills. The surface on multiyear ice fields in places not subject to deformations is also hillocky due to non-uniform multiple melting. The ice is almost salt-free. Its color, where bare, is usually blue. As a result of melting, round puddles appear at its surface in summer and a well-developed drainage system is formed.

Polar waters means both Arctic waters and/or Antarctic area.

Antarctic area means the sea area south of latitude 60°S (see Fig. 2.3-1).

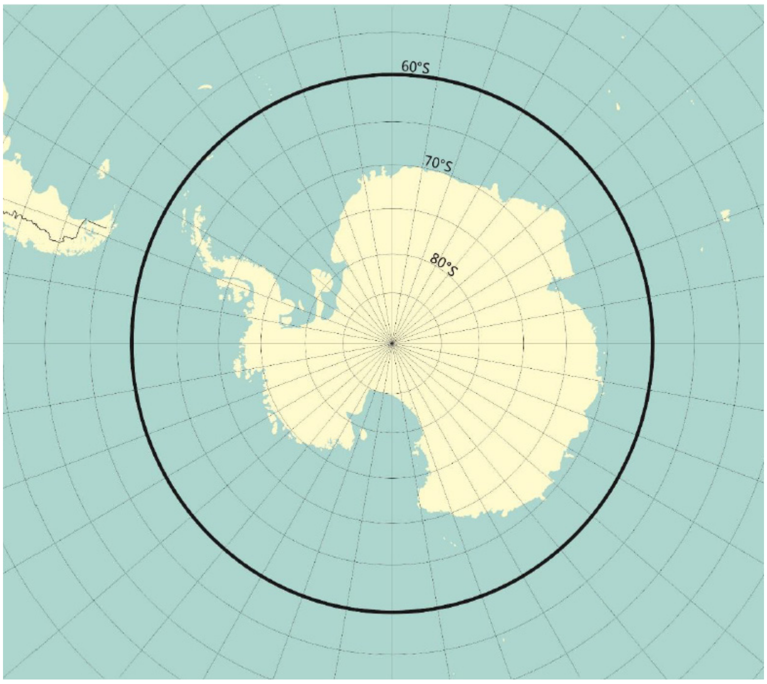


Fig. 2.3-1
Maximum extent of Antarctic area

Arctic waters means those waters which are located north of a line from the latitude 58°00'.0 N and longitude 042°00'.0 W to latitude 64°37'.0 N, longitude 035°27'.0 W and thence by a rhumb line to latitude 67°03'.9 N, longitude 026°33'.4 W and thence by a rhumb line to the latitude 70°49'.56 N and longitude 008°59'.61 W (Sørkapp, Jan Mayen) and by the southern shore of Jan Mayen to 73°31'.6 N and 019°01'.0 E by the Island of Bjørnøya, and thence by a great circle line to the latitude 68°38'.29 N and longitude 043 º23'.08 E (Cap Kanin Nos) and hence by the northern shore of the Asian Continent eastward to the Bering Strait and thence from the Bring Strait westward to latitude 60°N as far as I1'pyrskiy and following the 60 th North parallel eastward as far as and including Etolin Strait and thence by the northern shore of the North American continent as far south as latitude 60°N and thence eastward along parallel of latitude 60°N, to longitude 056°37'.1 W and thence to the latitude 58°00'.0 N, longitude 042°00'.0 W (see fig. 2.3-2).

Rys. 2.3-2 Maximum extent of Arctic waters

3 SURVEY AND CLASSIFICATION

3.1 Ships built in accordance with the requirements specified in Part I applicable to ships intended for unaided navigation in winter in the Baltic Sea or other non-Arctic seas with similar ice conditions may be assigned the following ice strengthening marks appropriate to the ice strengthenings applied:

3.1.1 **L3**, for unaided navigation on in light ice conditions, with the assistance of icebreakers – where necessary (see *PRS Rules* paragraph 3.4.4.2.2, *Part I – Classification Regulations*),

3.1.2 **L2**, for unaided navigation in medium ice conditions, with the assistance of icebreakers – where necessary (see *PRS Rules* paragraph 3.4.4.2.2, *Part I – Classification Regulations*),

3.1.3 **L1**, for navigation in heavy ice conditions, with the assistance of icebreakers – where necessary (see *PRS Rules* paragraph 3.4.4.2.1, *Part I – Classification Regulations*),

3.1.4 **L1A**, for navigation in heavy ice conditions without the assistance of icebreakers – (see *PRS Rules* paragraph 3.4.4.2.1, *Part I – Classification Regulations*).

3.2 Ships built in accordance with the requirements specified in Part II as well as in the Polar Code, applicable to ships intended for navigation in ice-infested polar waters, may be assigned the following ice strengthening marks appropriate to the ice strengthenings applied:

- **.1** **PC7**, for unaided summer/autumn operation in thin first-year ice which may include old ice inclusions,
- **.2** **PC6**, for unaided summer/autumn operation in medium first-year ice which may include old ice inclusions
- **.3** **PC5**, for unaided year-round operation in medium first-year ice which may include old ice inclusions

.4 **PC4**, for unaided year-round operation in thick first-year ice which may include old ice inclusions
.5 **PC3**, for unaided navigation in second-year ice which may include multi-year ice inclusions
.6 **PC2**, for unaided year-round operation in moderate multi-year ice inclusions,
.7 **PC1**, for unaided year-round operation in all polar waters.

3.3 Ships built in accordance with the requirements specified in Supplement intended for navigation in waters with light ice conditional and light localized drift ice, in river mouths and coastal areas may be assigned the following ice strengthening marks appropriate to the ice strengthenings applied:
.1 **(L4)**, for unaided occasional navigation in very light ice conditions (in fine ice pieces), In each case the hull structure strength need not exceed that required for ice strengthening **L3**,
.2 **E**, for unaided navigation in drift ice in mouths of rivers and coastal regions.

4 TECHNICAL DOCUMENTATION

4.1 The classification documentation submitted to PRS shall contain details related to ice classes with respect to design, arrangement and strength.

4.2 On the shell expansion drawing, lines separating the forward, midship and aft regions of the ice belt, UIWL and LIWL, as well as the upper and lower boundaries of minimum extension of frames strengthened for ice conditions shall be indicated.

4.3 The displacement D_s and maximum continuous power N_s shall be indicated on the drawings of midship section and shell expansion.

PART I

SECTION 1

PRS Ice Class Regulations 2017 and the Application Thereof

1 GENERAL

1.1 Purpose of the regulations

These regulations constitute detailed regulations on the requirements concerning the structure, engine output and other ice navigation properties of ships belonging to different ice classes, on the methods for determining ice classes, and on differences between ice classes, referred to in section 4.1 of the Act on the Ice Classes of Ships and Icebreaker Assistance (1121/2005).

1.2 Application of the 2017 Ice Class Regulations

The Ice Class Regulations of 2017 apply to ships contracted for construction on or after 1 January 2019.

As from 1 December 2017, the Ice Class Regulations of 2017 may, however, also be applied to ships contracted for construction on or after 1 December 2017.

The provisions in section 1.8 (Ice classes) and chapter 2 (Ice Class Draught) in the 2017 Ice Class Regulations apply to all ships irrespective of their year of build.

1.3 Application of the 2010 Ice Class Regulations

The Ice Class Regulations of 2010 apply to ships contracted for construction on or after 1 January 2012 but before 1 January 2019.LINK

1.4 Application of the 2008 Ice Class Regulations

The Ice Class Regulations of 2008 apply to ships contracted for construction on or after 1 January 2010 but before 1 January 2012.

1.5 Application of the 2002 Ice Class Regulations

The Ice Class Regulations of 2002 apply to ships the keels of which have been laid or which have been at a similar stage of construction on or after 1 September 2003 but which have been contracted for construction before 1 January 2010.

1.6 Application of the 1985 Ice Class Regulations

The Ice Class Regulations of 1985, as amended, apply to ships the keels of which have been laid or which have been at a similar stage of construction on or after 1 November 1986 but before 1 September 2003. On the owner's request, the requirements of the 2008 Ice Class Regulations may, however, be applied to the engine output of such ships.

However, ships of ice class **L1A** and **L1** the keels of which have been laid or which have been at a similar stage of construction before 1 September 2003 shall comply with the requirements in section 3.2.2 or 3.2.4 of the Ice Class Regulations of 2017 not later than 1 January in the year when twenty years have elapsed since the year the ship was delivered.

1.7 Application of the 1971 Ice Class Regulations

The requirements of Annex I or, depending on the ship's age, section 10 of the Board of Navigation Rules for Assigning Ships Separate Ice-Due Classes of 1971 (6.4.1971 No.1260/71/307), as amended, apply to ships the keels of which have been laid or which have been at a similar stage of construction before 1 November 1986. On the owner's request, the requirements of the 1985 Ice Class Rules or the 2008 Ice Class Regulations may, however, be applied to the engine output of such ships.

However, ships of ice class **L1A** and **L1** the keels of which have been laid or which have been at a similar stage of construction before 1 September 2003 shall comply with the requirements in section 3.2.2 or 3.2.4 of the Ice Class Regulations of 2017 not later than 1 January in the year when twenty years have elapsed since the year the ship was delivered.

1.8 Ice classes

PRS Baltic Ice Class notation requirements are fully aligned with Finish-Swedish Ice Class Rules published by TRAFI.

The requirements for ships having ice class notation applicable to ships classed by PRS and built before 2010 can be found in relevant Parts of PRS Rules in web page archive of PRS Rules (https://www.prs.pl/prs-rules-and-publications/classification-rules.html)

2 ICE CLASS DRAUGHT

2.1 Upper and Lower Ice Waterlines

The upper ice waterline (UIWL) shall be the envelope of the highest points of the waterlines at which the ship is intended to operate in ice. The line may be a broken line.

The lower ice waterline (LIWL) shall be the envelope of the lowest points of the waterlines at which the ship is intended to operate in ice. The line may be a broken line.

2.2 Maximum and Minimum Draught Fore and Aft

The maximum and minimum ice class draughts at fore and aft perpendiculars shall be determined in accordance with the upper and lower ice waterlines and the draught of the ship at fore and aft perpendiculars, when ice conditions require the ship to be ice-strengthened, shall always be between the upper and lower ice waterlines.

Restrictions on draughts when operating in ice shall be documented and kept on board readily available to the master. The maximum and minimum ice class draughts fore, amidships and aft shall be indicated in the class certificate. For ships built on or after 1 July 2007, if the summer load line in fresh water is anywhere located at a higher level than the UIWL, the ship's sides are to be provided with a warning triangle and with an ice class draught mark at the maximum permissible ice class draught amidships (see Annex III). Ships built before 1 July 2007 shall be provided with such a marking, if the UIWL is below the summer load line, not later than the first scheduled dry docking after 1 July 2007.

The draught and trim, limited by the UIWL, must not be exceeded when the ship is navigating in ice. The salinity of the sea water along the intended route shall be taken into account when loading the ship.

The ship shall always be loaded down at least to the draught of LIWL amidships when navigating in ice. Any ballast tank, situated above the LIWL and needed to load down the ship to this water line, shall be equipped with devices to prevent the water from freezing. In determining the LIWL, regard shall be paid to the need to ensure a reasonable degree of ice-going capability in ballast. The highest point of the propeller shall be submerged and if possible at a depth of at least h_i below the water surface in all loading conditions. The forward draught shall be at least:

$$(2 + 0.00025\,\Delta)h_i\,[\mathrm{m}],\ \text{but need not exceed } 4h_i, \tag{2.2}$$

where:
Δ is the displacement of the ship [t] determined from the waterline on the UIWL (see section 2.1). Where multiple waterlines are used for determining the UIWL, the displacement must be determined from the waterline corresponding to the greatest displacement.

h_i is the level ice thickness [m] according to section 4.2.1.

3 ENGINE OUTPUT

3.1 Definition of Engine Output

The engine output P is the total maximum output the propulsion machinery can continuously deliver to the propeller(s). If the output of the machinery is restricted by technical means or by any regulations applicable to the ship, P shall be taken as the restricted output. If additional power sources are available for propulsion power (e.g. shaft motors), in addition to the power of the main engine(s), they shall also be included in the total engine output.

3.2 Required Engine Output for Ice Classes L1A, L1, L2 and L3

The engine output shall not be less than that determined by the formula below and in no case less than 1,000 kW for ice class **L1A**, **L1**, **L2** and **L3**, and no less than 2,800 kW for **L1A**.

3.2.1 Definitions

The dimensions of the ship and some other parameters are defined as follows:

L	m	the length of the ship between the perpendiculars
L_{BOW}	m	the length of the bow
L_{PAR}	m	the length of the parallel midship body
B	m	the maximum breadth of the ship
T	m	the actual ice class draughts of the ship according to 3.2.2
A_{wf}	m^2	the area of the waterline of the bow
α	degree	the angle of the waterline at $B/4$
φ_1	degree	the rake of the stem at the centerline
φ_2	degree	the rake of the bow at $B/4$
Ψ	degree	the flare angle calculated as $\Psi = \tan^{-1}\left(\frac{\tan\phi}{\sin\alpha}\right)$ using local angles α and φ at each location. For chapter 3, the flare angle is calculated using $\varphi = \varphi_2$
D_P	m	the diameter of the propeller
H_M	m	the thickness of the brash ice in mid channel
H_F	m	the thickness of the brash ice layer displaced by the bow.

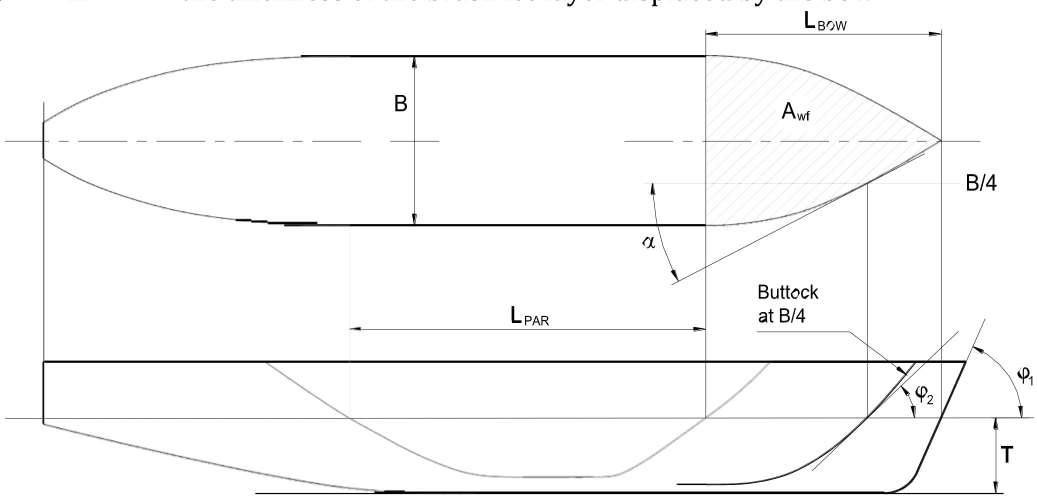


Figure 3-1.
Determination of the geometric quantities of the hull. If the ship has a bulbous bow, then φ_1 = 90°.

3.2.2 New ships

To qualify for ice class **L1A**, **L1**, **L2** or **L3**, a ship the keel of which is laid or which is at a similar stage of construction on or after 1 September 2003 shall comply with the following requirements regarding its engine output. The engine output requirement shall be calculated for two draughts by formula 3.1. Draughts to be used are the maximum draught amidships referred to as UIWL and the minimum draught amidships referred to as LIWL, as defined in section 2.2. In the calculations, the ship's parameters which depend on the draught must be determined at the appropriate draught, but L and B must be determined only at the UIWL. The engine output shall be no less than the greater of these two outputs.

$$P_{\min} = K_e \frac{(\frac{R_{CH}}{1000})^{3/2}}{D_p} \text{ [kW]} \tag{3.1}$$

where Ke shall be given a value according to Table 3-1.

Table 3-1: Values of K_e for conventional propulsion systems

Number of propellers	CP propeller or electric or hydraulic propulsion machinery	FP propeller
1 propeller	2.03	2.26
2 propellers	1.44	1.60
3 propellers	1.18	1.31

These K_e values apply to conventional propulsion systems. Other methods may be used for determining the required power for advanced propulsion systems (see 3.2.5)

R_{CH} is the ice resistance in Newton of the ship in a channel with brash ice and a consolidated surface layer.

$$R_{CH} = C_1 + C_2 + C_3 C\mu (H_F + H_M)^2 (B + C_\Psi H_F) + C_4 L_{PAR} H_F^2 + C_5 (\frac{LT}{B^2})^3 \frac{A_{wf}}{L} \tag{3.2}$$

where:

$C\mu$ = 0.15cos φ_2 + sinψsinα, C_μ has a value equal to or larger than 0.45
$C\psi$ = 0.047ψ–2.115 and C_ψ = 0 if $\psi \le 45°$
H_F = 0.26+$(H_M B)^{0.5}$
H_M = 1.0 m for ice classes **L1** and **L1A**
= 0.8 m for ice class **L2**
= 0.6 m for ice class **L3**

C_1 and $C2$ take account of a consolidated upper layer of brash ice. C_1=0 and C_2=0 for ice classes **L1**, **L2** and **L3**.

For ice class **L1A**:

$$C_1 = f_1 \frac{BL_{PAR}}{2\frac{T}{B}+1} + (1 + 0.021\phi_1)(f_2 B + f_3 L_{BOW} + f_4 L_{BOW})$$

$$C_2 = (1 + 0.063\phi_1)(g_1 + g_2 B) + g_3 (1 + 1.2\frac{T}{B})\frac{B^2}{\sqrt{L}}$$

For a ship with a bulbous bow, φ_1= 90°.

Coefficients f_1-f_4 and g_1-g_3 are given in Table 3-2.

Table 3-2: Values of coefficients f_1-f_4 and g_1-g_3 for the determination of C_1 and C_2

f_1 = 23 N/m^2 f_2 = 45.8 N/m f_3 = 14.7 N/m f_4 = 29 N/m^2	g_1 = 1530 N g_2 = 170 N/m g_3 = 400 N/m$^{1.5}$

C_3 = 845 kg/(m^{2}s^2)
C_4 = 42 kg/(m^{2}s^2)
C_5 = 825 kg/s^2
$\psi = \tan^{-1}\left(\frac{\tan\phi_2}{\sin\alpha}\right)$

If the value of the term $\left(\frac{LT}{B^2}\right)^3$ is less than 5, the value 5 shall be used and if the value of the term is more than 20, the value 20 shall be used.

3.2.3 Existing ships of ice class L2 or L3

In order to retain ice class L2 or L3 a ship, to which ice class regulations 1985, as amended apply, shall comply with the required minimum engine output as defined in section 3.2.1 of the ice class regulations 1985. For ease of reference, the provisions for ice classes L2 and L3 of section 3.2.1 of the ice class regulations 1985 are given in Annex II of these regulations.

3.2.4 Existing ships of ice class L1 or L1A

In order to retain ice class **L1A** or **L1** a ship, the keel of which has been laid or which has been at a similar stage of construction before 1 September 2003, shall comply with the requirements in section 3.2.2 above not later than 1 January in the year when twenty years have elapsed since the year the ship was delivered.

If the ship does not comply with the requirements in section 3.2.2 on the date given above, the highest lower ice class for which the engine output is sufficient can be confirmed for the ship.

When, for an existing ship, values for some of the hull form parameters required for the calculation method in section 3.2.2 are difficult to obtain, the following alternative formulae can be used:

$$R_{CH} = C_1 + C_2 + C_3(H_F + H_M)^2(B + 0.658H_F) + C_4LH_F^2 + C_5\left(\frac{LT}{B^2}\right)\frac{B}{4} \qquad (3.3)$$

where for ice class **L1**, C_1=0 and C_2=0.

For ice class **L1A**, ship without a bulb, C_1 and C_2 shall be calculated as follows:

$$C_1 = f_1\frac{BL}{2\frac{T}{B}+1} + 1{,}84(f_2B + f_3L + f_4BL)$$

$$C_2 = 3.52(g_1 + g_2B) + g_3(1 + 1.2\frac{T}{B})\frac{B^2}{\sqrt{L}}$$

For ice class **L1A**, ship with a bulb, C_1 and C_2 shall be calculated as follows:

$$C_1 = f_1\frac{BL}{2\frac{T}{B}+1} + 2.89(f_2B + f_3L + f_4BL)$$

$$C_2 = 6.67(g_1 + g_2B) + g_3(1 + 1.2\frac{T}{B})\frac{B^2}{\sqrt{L}}$$

Coefficients f_1-f_4 and g_1-g_3 are given in Table 3-3.

Table 3-3: Values of coefficients f_1-f_4 and g_1-g_3 for the determination of C_1 and C_2

f_1 = 10.3 N/m^2 f_2 = 45.8 N/m f_3 = 2.94 N/m f_4 = 5.8 N/m^2	g_1 = 1530 N g_2 = 170 N/m g_3 = 400 N/m $^{1.5}$

C_3 = 460 kg/(m^{2}s^2)
C_4 = 18.7 kg/(m^{2}s^2)
C_5 = 825 kg/s^2

If the value of the term $\left(\frac{LT}{B^2}\right)^3$ is less than 5, the value 5 shall be used and if the value of the term is more than 20, the value 20 shall be used.

3.2.5 Other methods of determining K_e or R_{CH}

For an individual ship, in lieu of the K_e or R_{CH} values defined in sections 3.2.2 and 3.2.3, the use of K_e or R_{CH} values based on more precise calculations or values based on model tests may be approved. Such approval will be given on the understanding that it can be revoked if experience of the ship's performance provides grounds for this in practice.

The design requirement for ice classes is a minimum speed of 5 knots in the following brash ice channels:

L1A H_M = 1.0 m and a 0.1 m thick consolidated layer of ice
L1 = 1.0 m
L2 = 0.8 m
L3 = 0.6 m

4 HULL STRUCTURAL DESIGN

4.1 General

The method for determining hull scantlings is based on certain assumptions concerning the nature of the ice load on the structure. These assumptions are based on full-scale observations made in the northern Baltic.

It has thus been observed that the local ice pressure on small areas can reach rather high values. This pressure may well be in excess of the normal uniaxial crushing strength of sea ice. This is explained by the fact that the stress field is in fact multiaxial.

Furthermore, it has been observed that the ice pressure on a frame can be higher than on the shell plating at the midspacing between frames. This is due to the different flexural stiffness of frames and shell plating. The load distribution is assumed to be as shown in Figure 4-1.

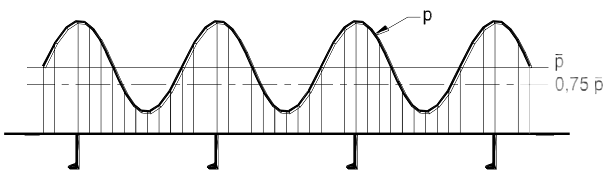


Figure 4-1.
Ice load distribution on a ship's side.

Direct analysis may be substituted for the formulae and values given in this section if they are deemed to PRS to be invalid or inapplicable for a given structural arrangement or detail.

Otherwise, direct analysis is not to be used as an alternative to the analytical procedures prescribed by the explicit requirements in sections 4.3 – 4.5.

Direct analyses are to be carried out using the load patch defined in section 4.2 (p, h and l_a). The pressure to be used is $1.8p$ where p is determined according to 4.2.2. The load patch must be applied at locations where the capacity of the structure under the combined effects of bending and shear is minimised. In particular, the structure must be checked with a load centred at the UIWL, $0.5h_0$ below the LIWL, and positioned at several vertical locations in between. Several horizontal locations shall also be checked, especially the locations centred at the mid-span or spacing. Furthermore, if the load length l_a cannot be determined directly from the arrangement of the structure, several values of l_a shall be checked using corresponding values for c_a.

The acceptance criterion for designs is that the combined stresses from bending and shear, when using the von Mises yield criterion, are lower than the yield point σ_y. When the direct calculation is based on beam theory, the allowable shear stress must be no larger than $0.9 \cdot \tau_y$, where $\tau y = \sigma_y / \sqrt{3}$.

If scantlings derived from these regulations are less than those required by the PRS for a ship that has not been ice strengthened, the latter shall be used.

NB1. The frame spacings and spans defined in the following text are normally (in accordance with the appropriate PRS rules for the ship in question) assumed to be measured along the plate and perpendicular to the axis of the stiffener for plates, along the flange for members with a flange, and along the free edge for flat bar stiffeners. For curved members the span (or spacing) is defined as the chord length between span (or spacing) points. The span points are defined by the intersection between the flange or upper edge of the member and the supporting structural element (stringer, web frame, deck or bulkhead). Figure 4-2 illustrates the determination of the span and spacing for curved members.

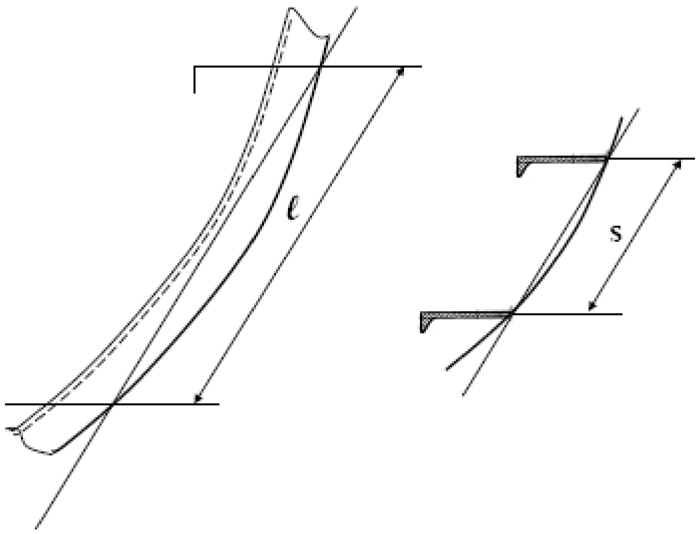


Figure 4-2. Definition of the frame span (left) and frame spacing (right) for curved members.

NB2. The effective breadth of the attached plate to be used for calculating the combined section modulus of the stiffener, stringer and web frame and attached plate must be given the value which the appropriate PRS rules require. The effective breadth shall in no case be more than what is stated in the appropriate PRS rules for the ship in question.

NB3. The requirements for the section modulus and shear area of the frames, stringers and web frames in 4.4, 4.5 and 4.6 are in accordance with the effective member cross section. For cases where the member is not normal to the plating, the section properties must be calculated in accordance with the appropriate PRS rules for the ship in question.

4.1.1 Hull regions

For the purpose of this section, the ship's hull is divided into regions as follows (see also Figure 4-3):

Bow region: From the stem to a line parallel to and 0.04·*L* aft of the forward borderline of the part of the hull where the waterlines run parallel to the centerline. For ice classes **L1A** and **L1**, the overlap over the borderline need not exceed 6 metres, for ice classes **L2** and **L3** this overlap need not exceed 5 metres.

Midbody region: From the aft boundary of the Bow region to a line parallel to and 0.04·*L* aft of the aft borderline of the part of the hull where the waterlines run parallel to the centerline. For ice classes **L1A** and **L1**, the overlap over the borderline need not exceed 6 metres, for ice classes **L2** and **L3** this overlap need not exceed 5 metres.

Stern region: From the aft boundary of the Midbody region to the stern.

L shall be taken as the ship's rule length used by PRS.

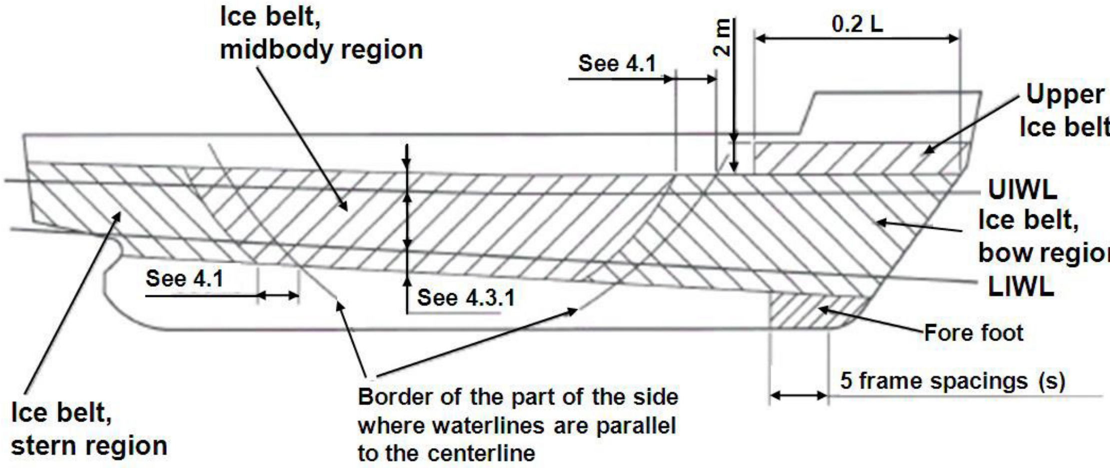


Figure 4-3. Ice strengthened regions of the hull

4.2 Ice load

4.2.1 Height of the ice load area

An ice-strengthened ship is assumed to operate in open sea conditions corresponding to a level ice thickness not exceeding h_i. The design ice load height (h) of the area actually under ice pressure at any particular point of time is, however, assumed to be only a fraction of the ice thickness. The values for h_i and h are given in Table 4-1.

Table 4-1: Values of h_i and h for the different ice classes

Ice Class	h_i [m]	h [m]
L1A	1.0	0.35
L1	0.8	0.30
L2	0.6	0.25
L3	0.4	0.22

4.2.2 Ice pressure

The design ice pressure is determined by the formula:

$$p = c_d \, c_p c_a p_0 \text{ [MPa]} \tag{4.1}$$

where:

c_d is a factor which takes account of the influence of the size and engine output of the ship. The value of this factor is a maximum of c_d = 1. It is calculated using the formula:

$$c_d = \frac{ak+b}{1000} \tag{4.2}$$

where

$$k = \frac{\sqrt{\Delta P}}{1000} \tag{4.3}$$

The values of a and b are given in Table 4-2.

Table 4-2: Values of a and b for different hull areas

	Bow		Midbody and Stern	
	k≤12	k>12	k≤12	k>12
a	30	6	8	2
b	230	518	214	286

Δ is the displacement of the ship at a maximum ice class draught [t] (see section 2.2).

P is the actual continuous engine output of the ship [kW] (see section 3.1) available when sailing in ice. If additional power sources are available for propulsion power (e.g. shaft motors) in addition to the power of the main engine(s), they shall also be included in the total engine output used as the basis for hull scantling calculations. The engine output used for the calculation of the hull scantlings shall be clearly stated on the shell expansion drawing.

c_p is a factor that reflects the magnitude of the load expected in the hull area in question relative to the bow area.

The value of c_p is given in Table 4-3.

Table 4-3: Values of c_p for different hull areas

	Bow	Midbody	Stern
L1A	1.0	1.0	0.75
L1	1.0	0.85	0.65
L2	1.0	0.70	0.45
L3	1.0	0.50	0.25

c_a is a factor which takes account of the probability that the full length of the area under consideration will be under pressure at the same time. It is calculated using the formula:

$$c_a = \sqrt{\frac{l_0}{l_a}}, \text{ maximum 1.0, minimum 0.35, } l_0 = 0.6 \text{ m.} \tag{4.4}$$

Values of l_a are given in Table 4-4.

p_o is the nominal ice pressure; the value 5.6 MPa shall be used.

Table 4-4: Values of la for different structural elements

Structure	Type of framing	l_a [m]
Shell	Transverse	Frame spacing
	Longitudinal	1.7×Frame spacing
Frames	Transverse	Frame spacing
	Longitudinal	Span of frame
Ice Stringer	-	Span of stringer
Web frame	-	2×Web frame spacing

4.3 Shell plating

4.3.1 Vertical extension of ice strengthening for plating (ice belt)

The vertical extension of the ice belt shall be as given in Table 4-5 (see Figure 4-3).

Table 4-5: Vertical extension of the ice belt

Ice class	Hull region	Above UIWL	Below LIWL
L1A	Bow	0.60 m	1.20 m
	Midbody		1.20 m
	Stern		1.00 m
L1	Bow	0.50 m	0.90 m
	Midbody		0.75 m
	Stern		0.75 m
L2 and **L3**	Bow	0.40 m	0.70 m
	Midbody		0.60 m
	Stern		0.60 m

In addition, the following areas shall be strengthened:

Fore foot: For ice class L1A, the shell plating below the ice belt from the stem to a position five main frame spacings abaft of the point where the bow profile departs from the keel line shall be ice-strengthened in the same way as the bow region.

Upper bow ice belt: For ice classes **L1A** and **L1** on ships with an open water service speed equal to or exceeding 18 knots, the shell plate from the upper limit of the ice belt to 2 m above it and from the stem to a position at least 0.2 *L* abaft of the forward perpendicular shall be ice-strengthened in the same way as the midbody region. A similar strengthening of the bow region is also advisable for a ship with a lower service speed when, on the basis of the model tests, for example, it is evident that the ship will have a high bow wave.

Sidescuttles shall not be situated in the ice belt. If the weather deck on any part of the ship is situated below the upper limit of the ice belt (e.g. in the way of the well of a raised quarter decker), the bulwark shall be provided with at least the same strength as is required for the shell in the ice belt. The strength of the construction of the freeing ports shall meet the same requirements.

4.3.2 Plate thickness in the ice belt

For transverse framing, the thickness of the shell plating shall be determined by the formula:

$$t = 667s\sqrt{\frac{f_1 p_{pi}}{\sigma_y}} + t_c \quad [\text{mm}] \tag{4.5}$$

and for longitudinal framing, the thickness of the shell plating shall be determined by the formula:

$$t = 667s\sqrt{\frac{p}{f_2 \sigma_y}} + t_c \ [\text{mm}] \tag{4.6}$$

where:

s is the frame spacing [m]

p_{pl} =0.75p [MPa], where p is as given in 4.2.2

$f_1 = 1.3 - \frac{4.2}{(\frac{h}{s}+1.8)^2}$, maximum 1.0,

$$f_2 = \begin{cases} 0.6 + \dfrac{0.4}{\frac{h}{s}}, \text{when} \dfrac{h}{s} \leq 1 \\ 1.4 - 0.4\left(\dfrac{h}{s}\right), \text{when } 1 \leq \dfrac{h}{s} \leq 1.8 \end{cases}$$

where h is as given in section 4.2.1.

σ_y is the yield stress of the material [N/mm^2], for which the following values shall be used:
σ_y = 235 N/mm^2 for normal-strength hull structural steel
σ_y = 315 N/mm^2 or higher for high-strength hull structural steel

If steels with different yield stress are used, the actual values may be substituted for the above ones if accepted by the PRS.

t_c is the increment for abrasion and corrosion [mm]; t_c shall normally be 2 mm; if a special surface coating, shown by experience to be capable of withstanding abrasion by ice, is applied and maintained, lower values may be approved.

4.4 Frames

4.4.1 Vertical extension of ice strengthening for framing

The vertical extension of the ice strengthening of framing shall be at least as given in Table 4-6.

Table 4-6: Vertical extension of the ice strengthening of framing

Ice class	Hull region	Above UIWL	Below LIWL
L1A	Bow	1.2 m	Down to tank top or below top of the floors
	Midbody		2.0 m
	Stern		1.6 m
L1, **L2** and **L3**	Bow	1.0 m	1.6 m
	Midbody		1.3 m
	Stern		1.0 m

Where an upper bow ice belt is required (see 4.3.1), the ice-strengthened part of the framing shall be extended to at least the top of this ice belt.

Where the ice-strengthening would go beyond a deck, the top or bottom plating of a tank or tank top by no more than 250 mm, it can be terminated at that deck, top or bottom plating of the tank or tank top.

4.4.2 Transverse frames

4.4.2.1 Section modulus and shear area

The section modulus of a main or intermediate transverse frame shall be calculated using the formula:

$$Z = \frac{pshl}{m_t \sigma_y} 10^6 \text{ [cm}^3\text{]}, \tag{4.7}$$

and the effective shear area will be calculated from

$$A = \frac{\sqrt{3} f_3 phs}{2\sigma_y} 10^4 \text{ [cm}^2\text{]}, \tag{4.8}$$

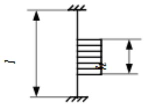
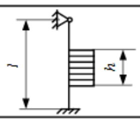
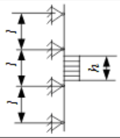
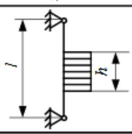
where

p is the ice pressure as given in 4.2.2 [MPa]
s is the frame spacing [m]
h is the height of the load area as given in 4.2.1 [m]
l is the span of the frame [m]

$m_t = \frac{7m_0}{7-5h/l}$,

f_3 is a factor which takes account of the maximum shear force versus the load location and the shear stress distribution, f_3 = 1.2
σ_y is the yield stress as in 4.3.2 [N/mm²]
m_o takes the boundary conditions into account. The values of m_o are given in Table 4-7.

Table 4-7: Values of m_0 for different boundary conditions

Boundary conditions for the main and intermediate frames	Value of m_0	Example
	7	Frames in a bulk carrier with top wing tanks
	6	Frames extending from the tank top to the main deck of a single-decked vessel
	5.7	Continuous frames between several decks or stringers
	5	Frames extending between two decks only

The boundary conditions are those for the main and intermediate frames. Load is applied at mid span.

Where less than 15% of the span, l, of the frame is situated within the ice-strengthening zone for frames as defined in 4.4.1, ordinary frame scantlings may be used.

4.4.2.2 Upper end of transverse framing

The upper end of the strengthened part of a main frame and of an intermediate ice frame shall be attached to a deck, top or bottom plating of a tank or an ice stringer (section 4.5).

Where a frame terminates above a deck or a stringer which is situated at or above the upper limit of the ice belt (section 4.3.1), the part above the deck or stringer may have the scantlings required by the PRS for an non ice-strengthened ship and the upper end of an intermediate frame may be connected to the adjacent frames by a horizontal member with the same scantlings as the main frame.

4.4.2.3 Lower end of transverse framing

The lower end of the strengthened part of a main frame and of an intermediate ice frame shall be attached to a deck, top or bottom plating of a tank, tank top or an ice stringer (section 4.5).

Where an intermediate frame terminates below a deck, top or bottom plating of a tank, tank top or ice stringer which is situated at or below the lower limit of the ice belt (section 4.3.1), the lower end may be connected to the adjacent main frames by a horizontal member of the same scantlings as the main frames. Note that the main frames below the lower edge of the ice belt must be ice strengthened, see 4.4.1.

4.4.3 Longitudinal frames

The following requirements are intended for longitudinal frames with all end conditions.

The section modulus of a longitudinal frame shall be calculated using the formula:

$$Z = \frac{f_4 p h l^2}{m \sigma_y} 10^6 \ [\text{cm}^3] \qquad (4.9)$$

The effective shear area of a longitudinal frame shall be:

$$A = \frac{\sqrt{3} f_4 f_5 p h l}{2\sigma_y} 10^4 \ [\text{cm}^3] \qquad (4.10)$$

In calculating the actual shear area of the frames, the shear area of the brackets should not be taken into account.

In the formulae given above:

f_4 is a factor which takes account of the load distribution over adjacent frames:

$$f_4 = (1 - 0.2\ h/s)$$

f_5 is a factor which takes account of the maximum shear force versus the load location and the shear stress distribution:

$$f_5 = 2.16$$

p is the ice pressure as given in section 4.2.2 [MPa]
h is the height of load area as given in section 4.2.1 [m]
s is the frame spacing [m]
l is the total span of the frame [m]
m is a boundary condition factor and m = 13.3 for a continuous beam with brackets; where the boundary conditions deviate significantly from those of a continuous beam with brackets, e.g. in an end field, a smaller boundary condition factor may be required.
σ_y is the yield stress as in 4.3.2 [N/mm²].

4.4.4 General on framing

4.4.4.1 The attachment of frames to supporting structures

Within the ice-strengthened area, all frames shall be effectively attached to all of the supporting structures. A longitudinal frame shall be attached by brackets to all supporting web frames and bulkheads. When a transversal frame terminates at a stringer or deck, a bracket or similar construction must be fitted. When a frame is running through the supporting structure, both sides of the web plate of the frame must be connected to the structure (by direct welding, collar plate or lug). When a bracket is installed, it must have at least the same thickness as the web plate of the frame and the edge must be appropriately stiffened against buckling.

4.4.4.2 Support of frames against instability, in particular tripping

The frames shall be attached to the shell by a double continuous weld. No scalloping is allowed (except when crossing shell plate butts).

The web thickness of the frames shall be at least the maximum of the following:

a) $\frac{h_w\sqrt{\sigma_y}}{C}$, h_w is the web height and C = 805 for profiles and C = 282 for flat bars;
b) half of the net thickness of the shell plating, $t - t_c$. For the purpose of calculating the minimum web thickness of frames, the required thickness of the shell plating must be calculated according to 4.3.2 using the yield strength σ_y of the frames;
c) 9 mm.

Where there is a deck, top or bottom plating of a tank, tank top or bulkhead in lieu of a frame, the plate thickness of it shall be calculated as above, to a depth corresponding to the height of the adjacent frames. In such a case, the material properties of the deck, top or bottom plating of the tank, tank top or bulkhead and the frame height h_W of the adjacent frames shall be used in the calculations, and the constant C shall be 805.

Asymmetrical frames and frames which are not at right angles to the shell (web less than 90 degrees to the shell) shall be supported against tripping by brackets, intercoastals, stringers or similar, at a distance not exceeding 1,300 mm. For frames with spans greater than 4 m, the extent of antitripping supports must be applied to all regions and for all ice classes. For frames with spans less than or equal to 4 m, the extent of antitripping supports must be applied to all regions for ice class **L1A**, to the bow and midbody regions for ice class **L1**, and to the bow region for ice classes **L2** and **L3**. Direct calculation methods may be applied to demonstrate the equivalent level of support provided by alternative arrangements.

4.5 Ice stringers

4.5.1 Stringers within the ice belt

The section modulus of a stringer situated within the ice belt (see 4.3.1) shall be calculated using the formula:

$$Z = \frac{f_6 f_7 p h l^2}{m \sigma_y} 10^6 \ [\text{cm}^3] \tag{4.11}$$

and the effective shear area shall be:

$$A = \frac{\sqrt{3} f_6 f_7 f_8 p h l}{2\sigma_y} 10^4 \ [\text{cm}^2] \tag{4.12}$$

where:
p is the ice pressure as given in section 4.2.2 [MPa]
h is the height of the load area as given in section 4.2.1 [m]

If the product ph is less than 0.15, value 0.15 [MN/m] shall be used.
l is the span of the stringer [m]
m is a boundary condition factor as defined in section 4.4.3
f_6 is a factor which takes account of the distribution of load over the transverse frames;
$f_6 = 0.9$
f_7 is the safety factor of the stringers; $f_7 = 1.8$
f_8 is a factor that takes account of the maximum shear force versus the load location and the shear stress distribution; $f_8 = 1.2$
σ_y is the yield stress as in section 4.3.2.

4.5.2 Stringers outside the ice belt

The section modulus of a stringer situated outside the ice belt but supporting ice-strengthened frames shall be calculated using the formula:

$$Z = \frac{f_9 f_{10} p h l^2}{m \sigma_y}\left(1 - \frac{h_s}{l_s}\right)10^6 \ [\mathrm{cm}^3] \tag{4.13}$$

and the effective shear area shall be:

$$A = \frac{\sqrt{3} f_9 f_{10} f_{11} p h l}{2\sigma_y}\left(1 - \frac{h_s}{l_s}\right)10^4 \ [\mathrm{cm}^2] \tag{4.14}$$

where:
p is the ice pressure as given in section 4.2.2 [MPa]
h is the height of the load area as given in section 4.2.1 [m]
If the product $p \cdot h$ is less than 0.15, value 0.15 [MN/m] shall be used.
l is the span of the stringer [m]
m is the boundary condition factor as defined in section 4.4.3
l_s is the distance to the adjacent ice stringer [m]
h_s is the distance to the ice belt [m]
f_9 is a factor which takes account of the distribution of load over the transverse frames;
$f_9=0.80$
f_{10} is the safety factor of the stringers; $f_{10}=1.8$
f_{11} is a factor that takes account of the maximum shear force versus the load location and shear stress distribution; $f_{11} = 1.2$
σ_y is the yield stress of material as in section 4.3.2.

4.5.3 Deck strips

Narrow deck strips abreast of hatches and serving as ice stringers shall comply with the section modulus and shear area requirements given in 4.5.1 and 4.5.2 respectively. In the case of very long hatches, the PRS may permit the product $p \cdot h$ to be given a value of less than 0.15 but in no case less than 0.10.

Regard shall be paid to the deflection of the ship's sides due to ice pressure with respect to very long (more than $B/2$) hatch openings, when designing weatherdeck hatch covers and their fittings.

4.6 Web frames

4.6.1 Ice load

The ice load transferred to a web frame from an ice stringer or from longitudinal framing shall be calculated using the formula:

$$F = f_{12} phS \ [\mathrm{MN}] \tag{4.15}$$

where:
p is the ice pressure as given in section 4.2.2 [MPa], in calculating c_a, however, l_a shall be 2S.
h is the height of the load area as given in section 4.2.1 [m]
If the product $p \cdot h$ is less than 0.15, value 0.15 [MN/m] shall be used.
S is the distance between the web frames [m]

f_{12} is the safety factor of web frames; f_{12}=1.8.

If the supported stringer is outside the ice belt, the force F shall be multiplied by $(1- h_s/l_s)$, where h_s and l_s shall be as defined in section 4.5.2.

4.6.2 Section modulus and shear area

The section modulus and shear area of the web frames shall be calculated using the formulae:

The effective shear area:

$$A = \frac{\sqrt{3}\alpha f_{13} Q}{\sigma_y} 10^4 \quad [\mathrm{cm}^2] \tag{4.16}$$

where:

Q is the maximum calculated shear force under the ice load F, as given in section 4.6.1
f_{13} is a factor that takes account of the shear force distribution, f_{13} = 1.1.
α is as given in Table 4-8
σ_y is the yield stress of the material as in section 4.3.2.

Section modulus:

$$Z = \frac{M}{\sigma_y}\sqrt{\frac{1}{1-(\frac{\gamma A}{A_a})^2}} 10^6 \quad [\mathrm{cm}^3] \tag{4.17}$$

where:

M is the maximum calculated bending moment under the ice load F; this must be given the value $M = 0.193Fl$;
γ is given in Table 4-8;
A is the required shear area;
A_a is the actual cross-sectional area of the web frame, $A_a = A_f + A_w$.

Table 4-8: Values of factors α and γ

A_f/A_w	0	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
α	1.5	1.23	1.16	1.11	1.09	1.07	1.06	1.05	1.05	1.04	1.04
γ	0	0.44	0.62	0.71	0.76	0.80	0.83	0.85	0.87	0.88	0.89

where:

A_f is the actual cross-sectional area of the free flange;
A_w is the actual effective cross-sectional area of the web plate.

4.7 Stem

The stem shall be made of rolled, cast or forged steel, or of shaped steel plates as shown in Figure 4-4.

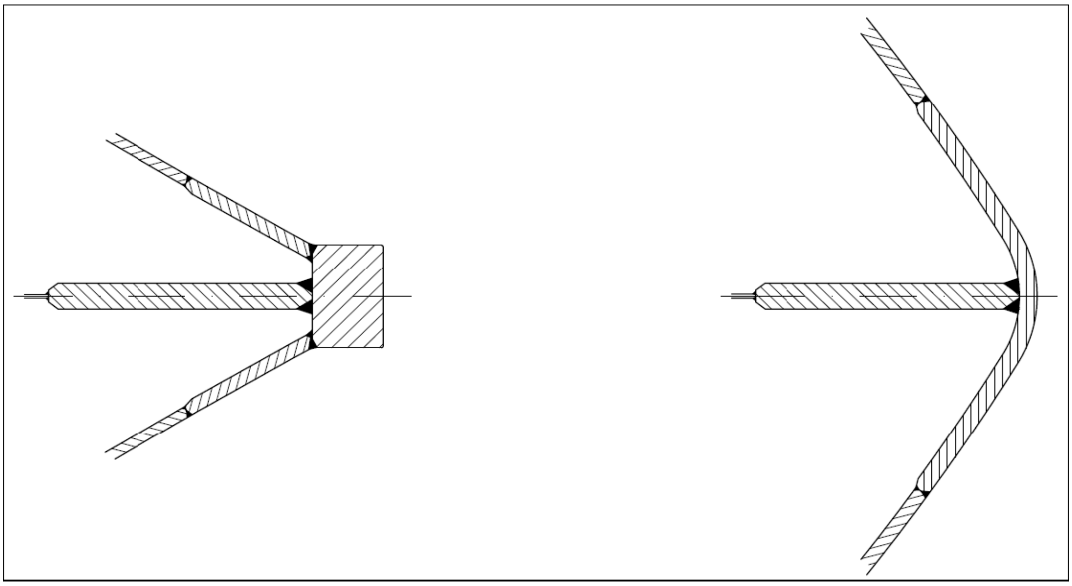


Figure 4-4. Examples of suitable stems.

The plate thickness of a shaped plate stem and, in the case of a blunt bow, any part of the shell where $\alpha \geq 30^{\text{o}}$ and $\psi \geq 75^{\text{o}}$ (see section 3.2.1 for angle definitions), shall be calculated according to formula 4.5, assuming that:

s is the spacing of elements supporting the plate [m];
$p_{PL} = p$ [MPa] (see section 4.3.2);
l_a is the spacing of vertical supporting elements [m].

The stem and the part of a blunt bow defined above shall be supported by floors or brackets spaced no more than 0.6 m apart and with a thickness of at least half the plate thickness. The reinforcement of the stem shall extend from the keel to a point 0.75 m above the UIWL or, if an upper bow ice belt is required (see section 4.3.1), to the upper limit of this.

4.8 Stern

The introduction of new propulsion arrangements with azimuthing thrusters, which provide improved manoeuvrability, will result in increased ice loading of the Stern region and the stern area. This fact should be considered in the design of the aft/stern structure.

In order to avoid very high loads on propeller blade tips, the minimum distance between the propeller(s) and the hull (including stern frame) should be no less than h_0 (see 4.2.1).

On twin and triple screw ships, the ice strengthening of the shell and framing shall be extended to the tank top 1.5 metres forward and aft of the side propellers.

The shafting and stern tubes of side propellers shall normally be enclosed within plated bossings. If detached struts are used, due consideration shall be taken of their design, strength and attachments to the hull.

5 RUDDER AND STEERING ARRANGEMENTS

The scantlings of the rudder post, rudder stock, pintles, steering engine etc. as well as the capability of the steering engine shall be determined according to the rules of the PRS. The maximum service speed of the ship to be used in these calculations shall, however, not be given a value lower than that stated below:

L1A 20 knots
L1 18 knots
L2 16 knots
L3 14 knots

If the actual maximum service speed of the ship is higher, that speed shall be used.

The local scantlings of rudders must be determined assuming that the whole rudder belongs to the ice belt. Furthermore, the rudder plating and frames are to be designed using the ice pressure *p* for the plating and frames in the midbody region.

For ice classes **L1** and **L1A**, the rudder (the rudder stock and the upper part of the rudder) shall be protected from direct contact with intact ice by an ice knife that extends below the LIWL, if practicable (or equivalent means). Special consideration shall be given to the design of the rudder and the ice knife for ships with flap-type rudders.

For ice classes **L1** and **L1A**, due regard shall be paid to the large loads that arise when the rudder is forced out of the midship position when sailing astern in ice or into ice ridges. Suitable arrangements such as rudder stoppers shall be installed to absorb such loads.

Relief valves for the hydraulic pressure in rudder turning mechanism(s) shall be installed. The components of the steering gear (e.g. rudder stock, rudder coupling, rudder horn etc.) shall be dimensioned to withstand loads causing yield stresses within the required diameter of the rudder stock.

6 PROPULSION MACHINERY

6.1 Scope

These regulations apply to propulsion machinery covering open- and ducted-type propellers with a controllable pitch or fixed pitch design for ice classes **L1A**, **L1**, **L2** and **L3**. The given propeller loads are the expected ice loads for the entire ship's service life under normal operational conditions, including loads resulting from the changing rotational direction of FP propellers. However, these loads do not cover off-design operational conditions, for example when a stopped propeller is dragged through ice. However, the load models of the regulations do not include propeller/ice interaction loads when ice enters the propeller of a turned azimuthing thruster from the side (radially).

The regulations also apply to azimuthing and fixed thrusters for main propulsion, taking consideration of loads resulting from propeller/ice interaction and loads on the thruster body/ice interaction. The given azimuthing thruster body loads are the expected ice loads for the ship's service life under normal operational conditions. The local strength of the thruster body shall be sufficient to withstand local ice pressure when the thruster body is designed for extreme loads.

The thruster global vibrations caused by blade order excitation on the propeller may cause significant vibratory loads.

6.2 Definitions

c	m	chord length of blade section
$c_{0.7}$	m	chord length of blade section at 0.7R propeller radius
CP		controllable pitch
D	m	propeller diameter
d	m	external diameter of propeller hub (at propeller plane)
D_{limit}	m	limit value for propeller diameter

EAR		expanded blade area ratio
F_b	kN	maximum backward blade force for the ship's service life
F_{ex}	kN	ultimate blade load resulting from blade loss through plastic bending
F_f	kN	maximum forward blade force during the ship's service life
F_{ice}	kN	ice load
$(F_{ice})_{max}$	kN	maximum ice load during the ship's service life
FP		fixed pitch
h_0	m	depth of the propeller centreline from the lower ice waterline
H_{ice}	m	thickness of the maximum design ice block entering the propeller
I_e	kgm^2	equivalent mass moment of inertia of all parts on the engine side of the component under consideration
I_t	kgm^2	equivalent mass moment of inertia of the whole propulsion system
k		Shape parameter for Weibull distribution
LIWL	m	ower ice waterline
m		slope for SN curve in log/log scale
M_{BL}	kNm	blade bending moment
MCR		maximum continuous rating
n	rev./s	propeller rotational speed
n_n	rev./s	nominal propeller rotational speed at MCR in free running condition
N_{class}		reference number of impacts per nominal propeller rotational speed per ice class
N_{ice}		total number of ice loads on the propeller blade for the ship's service life
N_R		reference number of load for the equivalent fatigue stress (10^8 cycles)
N_Q		number of propeller revolutions during a milling sequence
$P_{0,7}$	m	propeller pitch at 0.7R radius
$P_{0.7n}$	m	propeller pitch at 0.7R radius at MCR in free running condition
$P_{0.7b}$	m	propeller pitch at 0.7R radius at MCR in bollard condition
Q	kNm	torque
Q_{emax}	kNm	maximum engine torque
Q_{max}	kNm	maximum torque on the propeller resulting from propeller/ice interaction
$Q_{max}{}^n$	kNm	maximum torque on the propeller resulting from propeller/ice interaction reduced to the rotational speed in question
Q_{motor}	kNm	electric motor peak torque
Q_n	kNm	nominal torque at MCR in free running condition
Q_r	kNm	response torque along the propeller shaft line
Q_{peak}	kNm	maximum of the response torque Qr
Q_{smax}	kNm	maximum spindle torque of the blade for the ship's service life
Q_{sex}	kNm	maximum spindle torque due to blade failure caused by plastic bending
Q_{vib}	kNm	vibratory torque at considered component, taken from frequency domain open water torque vibration calculation (TVC)
R	m	propeller radius
r	m	blade section radius
T	kN	propeller thrust
T_b	kN	maximum backward propeller ice thrust during the ship's service life

T_f	kN	maximum forward propeller ice thrust during the ship's service life
T_n	kN	propeller thrust at MCR in free running condition
T_r	kN	maximum response thrust along the shaft line
t	m	maximum blade section thickness
Z		number of propeller blades
α_i	deg	duration of propeller blade/ice interaction expressed in rotation angle
α_1	deg	phase angle of propeller ice torque for blade order excitation component
α_2	deg	phase angle of propeller ice torque for twice the blade order excitation
$\gamma_{\epsilon 1}$		the reduction factor for fatigue; scatter effect
$\gamma_{\epsilon 2}$		the reduction factor for fatigue; test specimen size effect
γ_v		the reduction factor for fatigue; variable amplitude loading effect
γ_m		the reduction factor for fatigue; mean stress effect
ρ		a reduction factor for fatigue correlating the maximum stress amplitude to the equivalent fatigue stress for 108 stress cycles
$\sigma_{0.2}$	MPa	proof yield strength (at 0.2% offset) of blade material
σ_{exp}	MPa	mean fatigue strength of blade material at 108 cycles to failure in sea water
σ_{fat}	MPa	equivalent fatigue ice load stress amplitude for 108 stress cycles
σ_{fl}	MPa	characteristic fatigue strength for blade material
σ_{ref1}	MPa	reference strength $\sigma ref1$=0,6σ0.2+0,4σu
σ_{ref2}	MPa	reference strength $\sigma ref2$=0,7σu or $\sigma ref2$=0,6σ0.2+0,4σu, whichever is less
σ_{st}	MPa	maximum stress resulting from Fb or Ff
σ_u	MPa	ultimate tensile strength of blade material
$(\sigma_{ice})_{bmax}$	MPa	principal stress caused by the maximum backward propeller ice load
$(\sigma_{ice})_{fmax}$	MPa	principal stress caused by the maximum forward propeller ice load
$(\sigma_{ice})_{max}$	MPa	maximum ice load stress amplitudę.
$(\sigma_{ice})_{max}$	MPa	maximum ice load stress amplitudę.

Table 6-1: Definition of loads

	Definition	Use of the load in design process
F_b	The maximum lifetime backward force on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to the 0.7R chord line. See Figure 6-1.	Design force for strength calculation of the propeller blade
F_f	The maximum lifetime forward force on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to the 0.7R chord line.	Design force for calculation of strength of the propeller blade
Q_{smax}	The maximum lifetime spindle torque on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade.	When designing the propeller strength, the spindle torque is automatically taken into account because the propeller load is acting on the blade in the form of distributed pressure on the leading edge or tip area.
T_b	The maximum lifetime thrust on a propeller (all blades) resulting from propeller/ice interaction. The direction of the thrust is the propeller shaft direction and the force is opposite to the hydrodynamic thrust	Is used for estimating the response thrust Tr. Tb can be used as an estimate of excitation in axial vibration calculations. However, axial vibration calculations are not required by the rules.

	Definition	Use of the load in design process
T_f	The maximum lifetime thrust on a propeller (all blades) resulting from propeller/ice interaction. The direction of the thrust is the propeller shaft direction acting in the direction of hydrodynamic thrust.	Is used for estimating the response thrust T_r. T_f can be used as an estimate of excitation in axial vibration calculations. However, axial vibration calculations are not required by the rules.
Q_{max}	The maximum ice-induced torque resulting from propeller/ice interaction on one propeller blade, including hydrodynamic loads on that blade.	Is used for estimating the response torque (Qr) along the propulsion shaft line and as excitation for torsional vibration calculations.
F_{ex}	Ultimate blade load resulting from blade loss through plastic bending. The force that is needed to cause total failure of the blade so that a plastic hinge appears in the root area. The force is acting on 0.8R. The spindle arm is 2/3 of the distance between the axis of blade rotation and the leading/trailing edge (whichever is the greater) at the 0.8R radius.	Blade failure load is used to dimension the blade bolts, pitch control mechanism, propeller shaft, propeller shaft bearing and trust bearing. The objective is to guarantee that total propeller blade failure does not lead to damage to other components.
Q_r	Maximum response torque along the propeller shaft line, taking account of the dynamic behaviour of the shaft line for ice excitation (torsional vibration) and the hydrodynamic mean torque on the propeller.	Design torque for propeller shaft line components.
T_r	Maximum response thrust along the shaft line, taking account of the dynamic behaviour of the shaft line for ice excitation (axial vibration) and the hydrodynamic mean thrust on the propeller	Design thrust for propeller shaft line components.
F_{ti}	Maximum response force caused by ice block impacts on the thruster body or the propeller hub	Design load for thruster body and slewing bearings.
F_{tr}	Maximum response force on the thruster body caused by ice ridge/thruster body interaction	Design load for thruster body and slewing bearings.

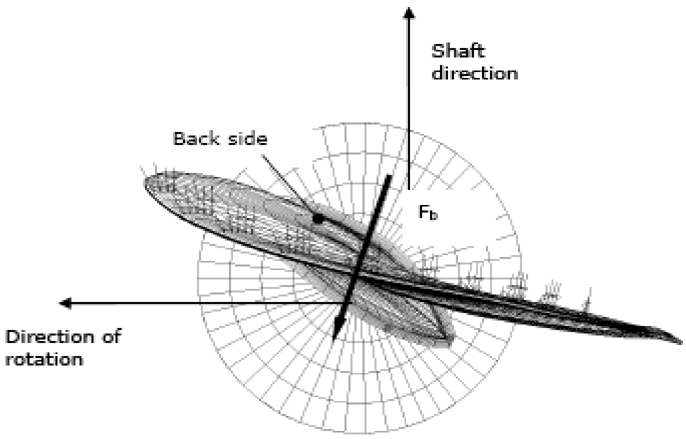


Figure 6-1.
Direction of the resultant backward blade force taken perpendicular to the chord line at radius 0.7R.
The ice contact pressure at the leading edge is indicated with small arrows.

6.3 Design ice conditions

In estimating the ice loads of the propeller for various ice classes, account was taken of different types of operation as shown in Table 6-2. For the estimation of design ice loads, a maxi- mum ice block size must be determined. The maximum design ice block entering the propeller is a rectangular ice block with the dimensions $H_{ice}\cdot 2H_{ice}\cdot 3H_{ice}$. The thickness of the ice block (H_{ice}) is given in Table 6-3.

Table 6-2: Types of operation for different ice classes.

Ice class	Operation of the ship
L1A	Operation in ice channels and in level ice The ship may proceed by ramming
L1, **L2**, **L3**	Operation in ice channels

Table 6-3: Thickness of design ice block.

	L1A	L1	L2	L3
Thickness of the design maximum ice block entering the propeller (H_{ice})	1.75 m	1.5 m	1.2 m	1.0 m

6.4 Materials

6.4.1 Materials exposed to sea water

The materials of components exposed to sea water, such as propeller blades, propeller hubs, and thruster body, shall have an elongation of no less than 15% in a test specimen, the gauge length of which is five times the diameter. A Charpy V impact test shall be carried out for materials other than bronze and austenitic steel. An average impact energy value of 20 J based on three tests must be obtained at minus 10°C. For nodular cast iron, average impact energy of 10 J at minus 10°C is required accordingly.

6.4.2 Materials exposed to sea water temperature

Materials exposed to sea water temperature shall be made of steel or another ductile material. An average impact energy value of 20 J, based on three tests, must be obtained at minus 10°C. This requirement applies to the propeller shaft, blade bolts, CP mechanisms, shaft bolts, strut-pod connecting bolts etc. It does not apply to surface-hardened components, such as bearings and gear teeth. The nodular cast iron of a ferrite structure type may be used for relevant parts other than bolts. The average impact energy for nodular cast iron shall be a minimum of 10 J at minus 10°C.

6.5 Design loads

The given loads are intended for component strength calculations only and are total loads, including ice-induced loads and hydrodynamic loads, during propeller/ice interaction. The presented maximum loads are based on a worst case scenario that occurs once during the service life of the ship. Thus, the load level for a higher number of loads is lower.

The values of the parameters in the formulae given in this section are provided in the units shown in the symbol list in 6.2.

If the highest point of the propeller is not at a depth of at least h_i below the water surface when the ship is in ballast condition, the propulsion system shall be designed according to ice class **L1** for ice classes **L2** and **L3**.

6.5.1 Design loads on propeller blades

F_b is the maximum force experienced during the lifetime of a ship that bends a propeller blade backwards when the propeller mills an ice block while rotating ahead. F_f is the maximum force experienced during the lifetime of a ship that bends a propeller blade forwards when the propeller mills an ice block while rotating ahead. F_b and F_f originate from different propeller/ice interaction phenomena, and do not occur simultaneously. Hence, they are to be applied to one blade separately.

6.5.1.1 Maximum backward blade force *Fb* for open propellers

$$F_b = 27(nD)^{0.7}(\frac{EAR}{Z})^{0.3}D^2 \text{ [KN], when } D \le D_{limit} \quad (6.1)$$

$$F_b = 23(nD)^{0.7}(\frac{EAR}{Z})^{0.3}DH_{ice}^{1.4} \text{ [KN], when } D > D_{limit} \quad (6.2)$$

where

$$D_{limit} = 0.85H_{ice}^{1.4} \quad (6.3)$$

n is the nominal rotational speed (at MCR in free running condition) of a CP propeller and 85% of the nominal rotational speed (at MCR in free running condition) of an FP propeller.

6.5.1.2 Maximum forward blade force *Ff* for open propellers

$$F_f = 250(\frac{EAR}{Z})D^2 \text{ [KN], when } D \le D_{limit} \quad (6.4)$$

$$F_f = 500(\frac{EAR}{Z})D\frac{1}{1-d/D}H_{ice} \text{ [KN], when } D > D_{limit} \quad (6.5)$$

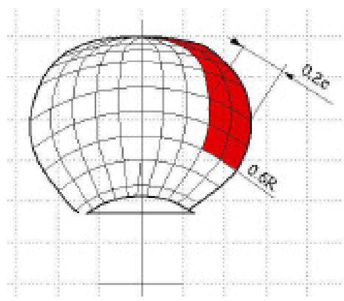
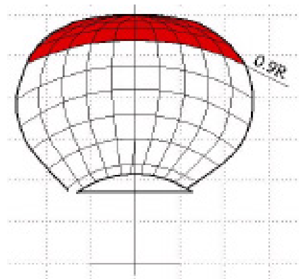
where

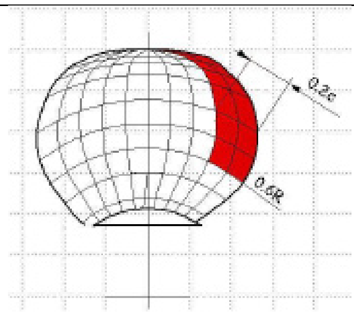
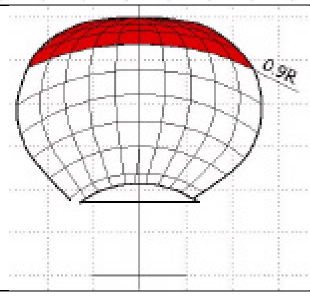
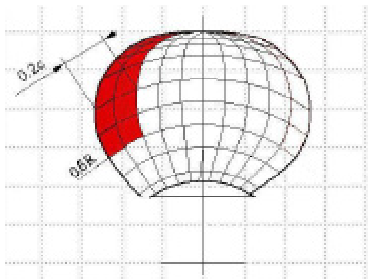
$$D_{limit} = \frac{2}{1-d/D}H_{ice} \text{ [m]} \quad (6.6)$$

6.5.1.3 Loaded area on the blade for open propellers

Load cases 1-4 must be covered, as given in Table 6-4 below, for CP and FP propellers. To obtain blade ice loads for a reversing propeller, load case 5 must also be covered for FP propellers.

Table 6-4: Load cases for open propellers

	Force	Loaded area	Right-handed propeller blade seen from behind
Load case 1	F_b	Uniform pressure applied on the blade back (suction side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length.	
Load case 2	50 % of F_b	Uniform pressure applied on the blade back (suction side) on the blade tip area outside 0.9R radius.	

	Force	Loaded area	Right-handed propeller blade seen from behind
Load case 3	F_f	Uniform pressure applied on the blade face (pressure side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length	
Load case 4	50% of F_f	Uniform pressure applied on the blade face (pressure side) of the blade tip area outside 0.9R radius	
Load case 5	60% of F_f or F_b, whichever is greater	Uniform pressure applied on the blade face (pressure side) to an area from 0.6R to the tip and from the trailing edge to 0.2 times the chord length	

6.5.1.4 Maximum backward blade ice force F_b for ducted propellers

$$F_b = 9.5(nD)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3} D^2 \text{ [kN], when } D \leq D_{limit} \quad (6.7)$$

$$F_b = 66(nD)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3} D^{0.6} H_{ice}^{1.4} \text{ [kN], when } D > D_{limit} \quad (6.8)$$

where:

$D_{limit} = 4H_{ice}$ [m]

n is the nominal rotational speed (at MCR in free running condition) of a CP propeller and 85% of the nominal rotational speed (at MCR in free running condition) of an FP propeller.

6.5.1.5 Maximum forward blade ice force F_f for ducted propellers

$$F_f = 250\left(\frac{EAR}{Z}\right) D^2 \text{ [kN], when } D \leq D_{limit} \quad (6.9)$$

$$F_f = 500\left(\frac{EAR}{Z}\right) D \frac{1}{1-d/D} H_{ice} \text{ [kN], when } D > D_{limit} \quad (6.10)$$

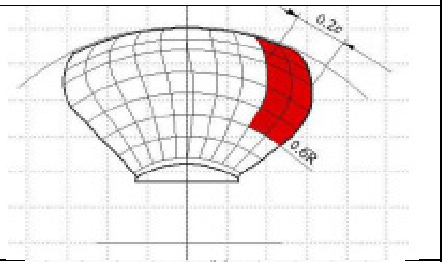
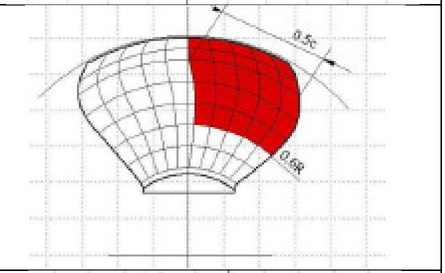
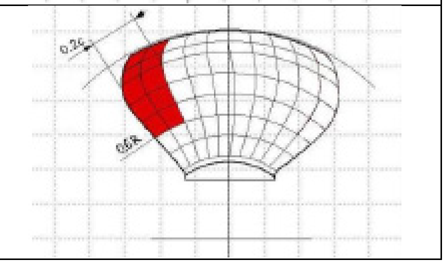
where:

$$D_{limit} = \frac{2}{1-d/D} H_{ice} \text{ [m]} \quad (6.11)$$

6.5.1.6 Loaded area on the blade for ducted propellers

Load cases 1 and 3 have to be covered as given in Table 6-5 for all propellers, and an additional load case (load case 5) for an FP propeller, to cover ice loads when the propeller is re-versed.

Table 6-5: Load cases for ducted propellers

	Force	Loaded area	Right handed propeller blade seen from behind
Load case 1	F_b	Uniform pressure applied on the blade back (suction side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length	
Load case 3	F_f	Uniform pressure applied on the blade face (pressure side) to an area from 0.6R to the tip and from the leading edge to 0.5 times the chord length	
Load case 5	60% of F_f or F_b, whichever is greater	Uniform pressure applied on the face (pressure side) to an area from 0.6R to the tip and from blade the trailing edge to 0.2 times the chord length.	

6.5.1.7 Maximum blade spindle torque Q_{smax} for open and ducted propellers

The spindle torque Q_{smax} around the axis of the blade fitting shall be determined both for the maximum backward blade force F_b and forward blade force F_f, which are applied as in Table 6-4 and Table 6-5. The larger of the obtained torques is used as the dimensioning torque. If the above method gives a value which is less than the default value given by the formula below, the default value shall be used.

$$Q_{smax} = 0.25 F c_{0.7} \text{ [kNm]} \tag{6.12}$$

where $c_{0.7}$ is the length of the blade section at 0.7R radius and F is either F_b or F_f, whichever has the greater absolute value.

6.5.1.8 Load distributions for blade loads

The Weibull-type distribution (probability that F_{ice} exceeds $(F_{ice})_{max}$), as given in Figure 6-2, is used for the fatigue design of the blade.

$$P\left(\frac{F_{ice}}{(F_{ice})_{\max}} \geq \frac{F}{(F_{ice})_{\max}}\right) = \exp\left(-\left(\frac{F}{(F_{ice})_{\max}}\right)^k lnN_{ice}\right) \tag{6.13}$$

where k is the shape parameter of the spectrum, N_{ice} is the number of load cycles in the spectrum, and F_{ice} is the random variable for ice loads on the blade, $0 \leq F_{ice} \leq (F_{ice})_{\max}$. The shape parameter k = 0.75 shall be used for the ice force distribution of an open propeller and the shape parameter k = 1.0 for that of a ducted propeller blade.

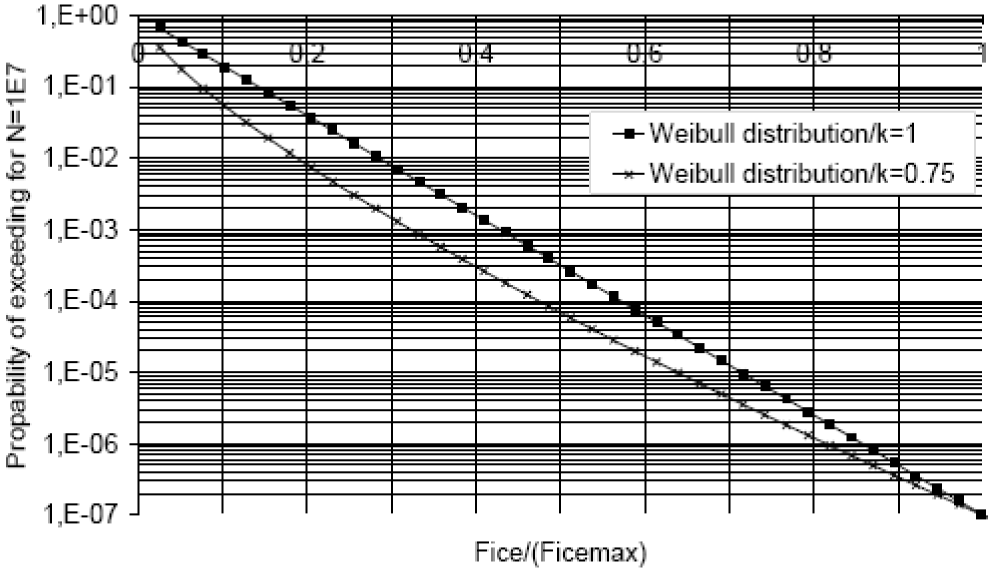


Figure 6-2.
The Weibull-type distribution (probability that F_{ice} exceeds $(F_{ice})_{max}$) used for fatigue design.

6.5.1.9 Number of ice loads

The number of load cycles per propeller blade in the load spectrum shall be determined according to the formula:

$$N_{ice} = k_1 k_2 k_3 N_{class} n_n \tag{6.14}$$

where the values for N_{class} are given in Table 6-6 and the propeller location factor k_1 in Table 6-7.

Table 6-6: Values of N_{class}

Ice Class	**L1A**	**L1**	**L2**	**L3**
impacts in life/n_n	$9 \cdot 10^6$	$6 \cdot 10^6$	$3.4 \cdot 10^6$	$2.1 \cdot 10^6$

Table 6-7: Values of the propeller location factor k_1

	Centre propeller Bow first operation	Wing propeller Bow first operation	Pulling propeller (wing and centre) Bow propeller or Stern first operation
k_1	1	2	3

The submersion factor k_2 is determined from the equation

$$\begin{aligned} k_2 &= 0.8 - f, \text{ when } f < 0 \\ &= 0.8 - 0.4f, \text{ when } 0 \le f \le 1 \\ &= 0.6 - 0.2f, \text{ when } 1 < f \le 2.5 \\ &= 0.1, \text{ when } f > 2.5 \end{aligned} \tag{6.15}$$

where the immersion function f is

$$f = \frac{h_0 - H_{ice}}{D/2} - 1 \tag{6.16}$$

where h_0 is the depth of the propeller centreline at the lower ice waterline (LIWL) of the ship.

The propulsion machinery type factor k_3 is 1 for fixed propulsors and 1.2 for azimuthing propulsors.

For components that are subject to loads resulting from propeller/ice interaction with all the propeller blades, the number of load cycles (N_{ice}) must be multiplied by the number of propeller blades (Z).

6.5.2 Axial design loads for open and ducted propellers

6.5.2.1 Maximum ice thrust on propeller T_f and T_b for open and ducted propellers

The maximum forward and backward ice thrusts are:

$$T_f = 1.1F_f \text{ [kN]} \tag{6.17}$$

$$T_b = 1.1F_b \text{ [kN]} \tag{6.18}$$

6.5.2.2 Design thrust along the propulsion shaft line for open and ducted propellers

The design thrust along the propeller shaft line must be calculated using the formulae below. The greater value of the forward and backward direction loads shall be taken as the design load for both directions. Factors 2.2 and 1.5 take account of the dynamic magnification resulting from axial vibration.

In a forward direction

$$T_r = T + 2.2T_f \text{ [kN]} \tag{6.19}$$

in a backward direction

$$T_r = 1.5T_b \text{ [kN].} \tag{6.20}$$

If the hydrodynamic bollard thrust, T, is not known, it must be taken as given in Table 6-8.

Table 6-8: Default values for hydrodynamic bollard thrust, *T*

Propeller type	T
CP propellers (open)	1.25 T_n
CP propellers (ducted)	1.1 T_n
FP propellers driven by turbine or electric motor	T_n
FP propellers driven by diesel engine (open)	0.85 T_n
FP propellers driven by diesel engine (ducted)	0.75 T_n

Here T_n is the nominal propeller thrust at MCR in the free running open water condition.

6.5.3 Torsional design loads

6.5.3.1 Design ice torque on propeller Q_{max} for open propellers

$Q_{\max}$ is the maximum torque on a propeller resulting from ice/propeller interaction during the service life of the ship.

$$Q_{\max} = 10.9\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}(nD)^{0.17}D^3 \text{ [kNm], when } D \le D_{limit.} \tag{6.21}$$

$$Q_{\max} = 20.7\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}(nD)^{0.17}D^{1.9}H_{ice}^{1.1} \text{ [kNm], when } D > D_{limit.} \tag{6.22}$$

where

$$D_{limit} = 1.8H_{ice} \text{ [m]} \tag{6.23}$$

n is the rotational propeller speed at MCR in bollard condition. If unknown, n must be attributed a value in accordance with Table 6-9.

Table 6-9: Default rotational propeller speed at MCR in bollard condition.

Propeller type	Rotational speed n
CP propellers	n_n
FP propellers driven by turbine or electric motor	n_n
FP propellers driven by diesel engine	0.85 n_n

Here n_n is the nominal rotational speed at MCR in the free running open water condition.

For CP propellers, the propeller pitch, $P_{0.7}$ shall correspond to MCR in bollard condition. If not known, $P_{0.7}$ shall have a value equal to $0.7 \cdot P_{0.7n}$, where $P_{0.7n}$ is the propeller pitch at MCR in free running condition.

6.5.3.2 Design ice torque on propeller $Qmax$ for ducted propellers

$Q_{\max}$ is the maximum torque on a propeller during the service life of the ship resulting from ice/propeller interaction.

$$Q_{\max} = 7.7\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}(nD)^{0.17}D^{3}\ [\text{kNm}],\ \text{when } D \le D_{limit} \tag{6.24}$$

$$Q_{\max} = 14.6\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}(nD)^{0.17}D^{1.9}H_{ice}^{1.1}\ [\text{kNm}],\ \text{when } D > D_{limit} \tag{6.25}$$

where

$$D_{limit}=1{,}8H_{ice}. \tag{6.26}$$

n is the rotational propeller speed at MCR in bollard condition. If not known, n shall have avalue according to Table 6-9.

For CP propellers, the propeller pitch, $P_{0.7}$ shall correspond to MCR in bollard condition. If not known, $P_{0.7}$ shall have a value equal to 0.7□$P_{0.7n}$, where $P_{0.7n}$ is the propeller pitch at MCR in free running condition.

6.5.3.3 Design torque for non-resonant shaft lines

If there is no relevant first blade order torsional resonance in the operational speed range or in the range 20% above and 20% below the maximum operating speed (bollard condition), the following estimation of the maximum torque can be used.

Directly coupled two stroke diesel engines without flexible coupling

$$Q_{peak}=Q_{emax}+Q_{vib}+Q_{max}\, I_e/I_t\ [\text{kNm}] \tag{6.27}$$

and other plants

$$Q_{peak}=Q_{emax}+Q_{max}\, I_e/I_t\ [\text{kNm}] \tag{6.28}$$

Where:

I_e is the equivalent mass moment of inertia of all parts on the engine side of the component under consideration and

I_t is the equivalent mass moment of inertia of the whole propulsion system.

All the torques and the inertia moments shall be reduced to the rotation speed of the component being examined.

If the maximum torque, Q_{emax}, is unknown, it shall be accorded the value given in Table 6-10.

Table 6-10: Default values for prime mover maximum torque

Propeller type	Q_{emax}
Propellers driven by electric motor	$^*Q_{motor}$
CP propellers not driven by electric motor	Q_n
FP propellers driven by turbine	Q_n
FP propellers driven by diesel engine	0.75 Q_n

*Q_{motor} is the electric motor peak torque.

6.5.3.4 Design torque for shaft lines having resonances

If there is first blade order torsional resonance in the operational speed range or in the range 20% above and 20% below the maximum operating speed (bollard condition), the design torque (Q_{peak}) of the shaft component shall be determined by means of torsional vibration analysis of the propulsion line. There are two alternative ways of performing the dynamic analysis.

Time domain calculation for estimated milling sequence excitation

Frequency domain calculation for blade orders sinusoidal excitation.

The frequency domain analysis is generally considered conservative compared to the time domain simulation, provided that there is a first blade order resonance in the considered speed range.

6.5.3.4.1 Time domain calculation of torsional response

Time domain calculations shall be calculated for the MCR condition, MCR bollard conditions and for blade order resonant rotational speeds so that the resonant vibration responses can be obtained.

The load sequence given in this chapter, for a case where a propeller is milling an ice block, shall be used for the strength evaluation of the propulsion line. The given load sequence is not intended for propulsion system stalling analyses.

The following load cases are intended to reflect the operational loads on the propulsion system, when the propeller interacts with ice, and the respective reaction of the complete system. The ice impact and system response causes loads in the individual shaft line components. The ice torque Q_{max} may be taken as a constant value in the complete speed range. When considerations at specific shaft speeds are performed, a relevant Q_{max} may be calculated using the relevant speed according to section 6.5.3.

Diesel engine plants without an elastic coupling shall be calculated at the least favourable phase angle for ice versus engine excitation, when calculated in the time domain. The engine firing pulses shall be included in the calculations and their standard steady state harmonics can be used.

If there is a blade order resonance just above the MCR speed, calculations shall cover rotation- al speeds up to 105% of the MCR speed.

The propeller ice torque excitation for shaft line transient dynamic analysis in the time domain is defined as a sequence of blade impacts which are of half sine shape. The excitation frequency shall follow the propeller rotational speed during the ice interaction sequence. The torque due to a single blade ice impact as a function of the propeller rotation angle is then defined using the formula:

$$Q(\phi) = C_q Q_{max} \sin(\phi(180/\alpha_i)) \tag{6.29}$$

when φ rotates from 0 to αi plus integer revolutions

$$Q(\phi) = 0, \tag{6.30}$$

when φ rotates from α_i to 360 plus integer revolutions,

where:

φ is the rotation angle from when the first impact occurs and parameters C_q and α_i are given in Table 6-11.

α_i is the duration of propeller blade/ice interaction expressed in terms of the propeller rotation angle. See Figure 6-3.

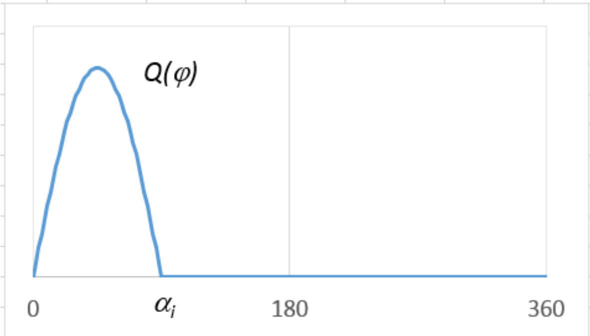


Figure 6-3.
Schematic ice torque due to a single blade ice impact as a function of the propeller rotation angle.

Table 6-11: Ice impact magnification and duration factors for different blade numbers.

Torque excitation	Propeller/ ice interaction	C_q	α_i [deg]			
			Z=3	Z=4	Z=5	Z=6
Excitation case 1	Single ice block	0.75	90	90	72	60
Excitation case 2	Single ice block	1.0	135	135	135	135
Excitation case 3	Two ice blocks (phase shift 360/(2·Z) deg.)	0.5	45	45	36	30
Excitation case 4	Single ice block	0.5	45	45	36	30

The total ice torque is obtained by summing the torque of single blades, while taking account of the phase shift 360 deg./Z, see Figure 6-4. At the beginning and end of the milling sequence (within the calculated duration) linear ramp functions shall be used to increase C_q to its maximum value within one propeller revolution and vice versa to decrease it to zero (see the examples of different Z numbers in Figure 6-4).

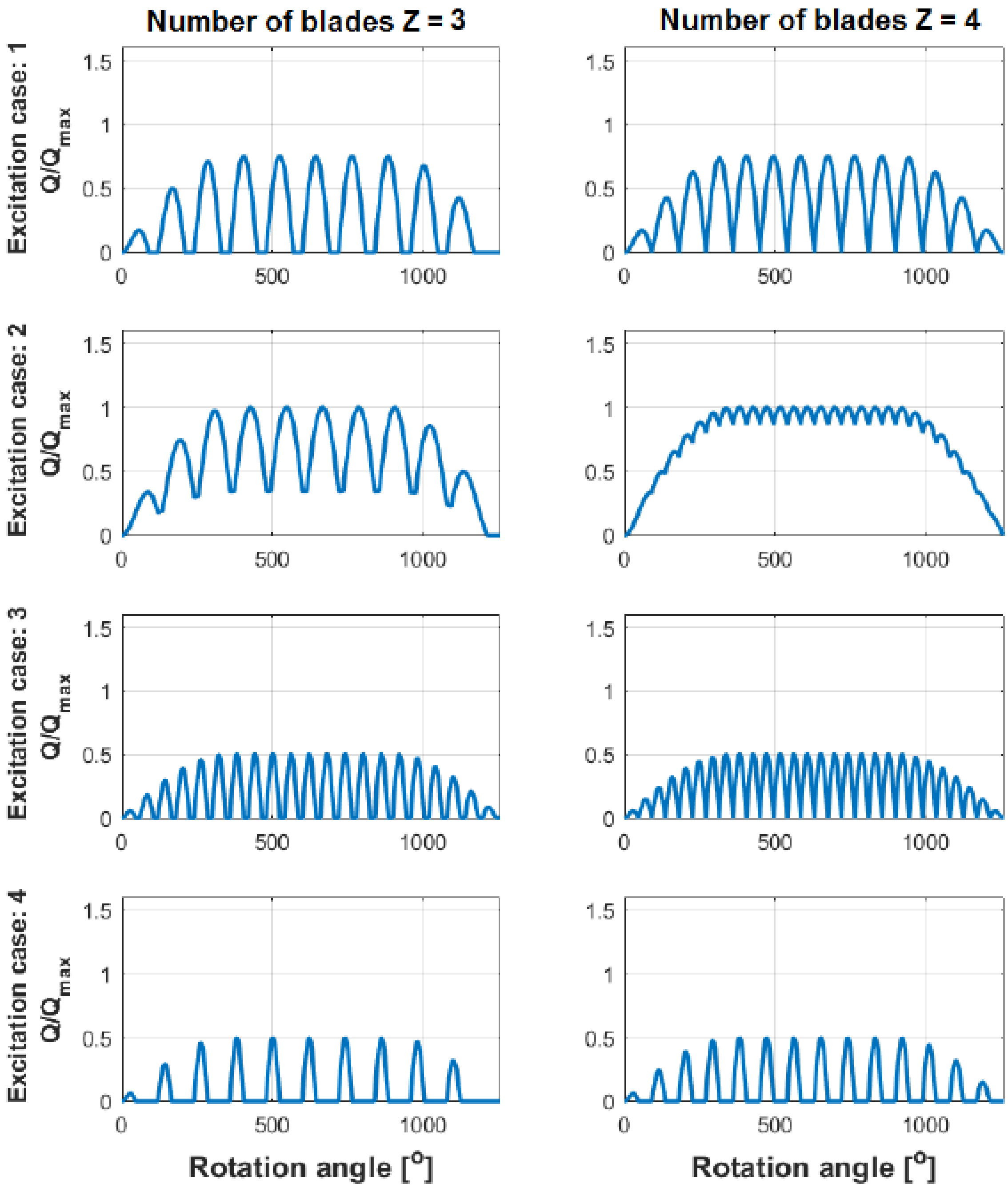
The number of propeller revolutions during a milling sequence shall be obtained from the formula:

$$N_Q = 2H_{ice} \tag{6.31}$$

The number of impacts is $Z \cdot N_Q$ for blade order excitation. An illustration of all excitation cases for different numbers of blades is given in Figure 6-4.

A dynamic simulation must be performed for all excitation cases at the operational rotational speed range. For a fixed pitch propeller propulsion plant, a dynamic simulation must also cover the bollard pull condition with a corresponding rotational speed assuming the maximum possible output of the engine.

If a speed drop occurs until the main engine is at a standstill, this indicates that the engine may not be sufficiently powered for the intended service task. For the consideration of loads, the maximum occurring torque during the speed drop process must be used.


Number of blades Z = 3
Number of blades Z = 4
Excitation case: 1
Excitation case: 2
Excitation case: 3
Excitation case: 4
Q/Q_max
1.5
1
0.5
0
500
1000
Rotation angle [°]

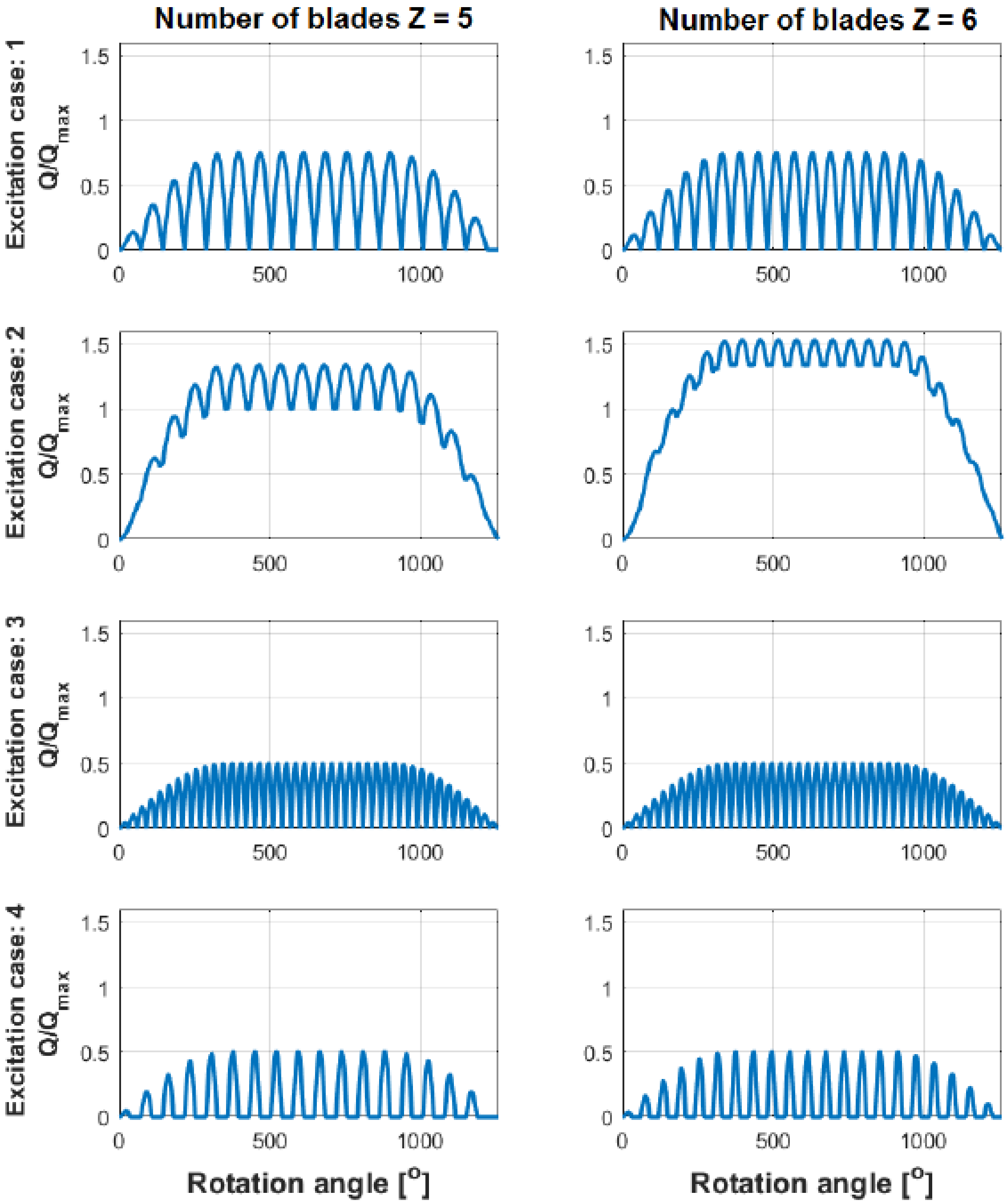


Figure 6-4.
The shape of the propeller ice torque excitation sequences for propellers with 3, 4, 5 or 6 blades.

For the time domain calculation, the simulated response torque typically includes the engine mean torque and the propeller mean torque. If this is not the case, the response torques must be obtained using the formula:

$$Q_{peak}=Q_{emax}+Q_{rtd,} \tag{6.32}$$

where Q_{rtd} is the maximum simulated torque obtained from the time domain analysis.

6.5.3.4.2 Frequency domain calculation of torsional response

For frequency domain calculations, blade order and twice-the-blade-order excitation may be used. The amplitudes for the blade order and twice-the-blade-order sinusoidal excitation have been derived based on the assumption that the time domain half sine impact sequences were continuous, and the Fourier series components for blade order and twice-the-blade-order components have been derived. The propeller ice torque is then:

$$Q_F(\varphi)=Q_{max}(C_{q0}+C_{q1}sin\ (ZE_{0\varphi}+\alpha_1)+C_{q2}sin\ (2ZE_0\varphi+\alpha_2)),\ [\text{kNm}] \qquad (6.33)$$

where

C_{q0} is mean torque parameter
C_{q1} is the first blade order excitation parameter
C_{q2} is the second blade order excitation parameter
α_1, α_2 are phase angles of the excitation component
φ is the angle of rotation
E_0 is the number of ice blocks in contact

The values of the parameters are given in Table 6-12.

Table 6-12: Coefficient values for frequency domain excitation calculation

Torque excitation	Z=3					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.375	0.36	-90	0	0	1
Excitation case 2	0.7	0.33	-90	0.05	-45	1
Excitation case 3	0.25	0.25	-90	0		2
Excitation case 4	0.2	0.25	0	0.05	-90	1
Torque excitation	Z=4					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.45	0.36	-90	0.06	-90	1
Excitation case 2	0.9375	0	-90	0.0625	-90	1
Excitation case 3	0.25	0.25	-90	0	0	2
Excitation case 4	0.2	0.25	0	0.05	-90	1
Torque excitation	Z=5					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.45	0.36	-90	0.06	-90	1
Excitation case 2	1.19	0.17	-90	0.02	-90	1
Excitation case 3	0.3	0.25	-90	0.048	-90	2
Excitation case 4	0.2	0.25	0	0.05	-90	1
Torque excitation	Z=6					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.45	0.36	-90	0.05	-90	1
Excitation case 2	1.435	0.1	-90	0	0	1
Excitation case 3	0.3	0.25	-90	0.048	-90	2
Excitation case 4	0.2	0.25	0	0.05	-90	1

The design torque for the frequency domain excitation case must be obtained using the formula:

$$Q_{peak} = Q_{emax} + Q_{vib} + (Q^{n}_{max} C_{q0}) I_e / I_t + Q_{rf1} + Q_{rf2}, \tag{6.34}$$

where

Q^{n}_{max} is the maximum propeller ice torque at the operation speed in consideration

C_{q0} is the mean static torque coefficient from Table 6-12

Q_{rf1} is the blade order torsional response from the frequency domain analysis

Q_{rf2} is the second blade order torsional response from the frequency domain analysis

If the prime mover maximum torque, Q_{emax}, is not known, it shall be taken as given in Table 6-10. All the torque values have to be scaled to the shaft revolutions for the component in question.

6.5.3.4.3 Guidance for torsional vibration calculation

The aim of time domain torsional vibration simulations is to estimate the extreme torsional load for the ship's lifespan. The simulation model can be taken from the normal lumped mass elastic torsional vibration model, including damping. For a time domain analysis, the model should include the ice excitation at the propeller, other relevant excitations and the mean torques provided by the prime mover and hydrodynamic mean torque in the propeller. The calculations should cover variation of phase between the ice excitation and prime mover excitation. This is extremely relevant to propulsion lines with directly driven combustion engines. Time domain calculations shall be calculated for the MCR condition, MCR bollard conditions and for resonant speed, so that the resonant vibration responses can be obtained.

For frequency domain calculations, the load should be estimated as a Fourier component analysis of the continuous sequence of half sine load sequences. First and second order blade components should be used for excitation.

The calculation should cover the entire relevant rpm range and the simulation of responses at torsional vibration resonances.

6.5.4 Blade failure load

6.5.4.1 Bending force, F_{ex}

The ultimate load resulting from blade failure as a result of plastic bending around the blade root shall be calculated using formula 6.35, or alternatively by means of an appropriate stress analysis, reflecting the non-linear plastic material behaviour of the actual blade. In such a case, the blade failure area may be outside the root section. The ultimate load is assumed to be acting on the blade at the 0.8R radius in the weakest direction of the blade.

A blade is regarded as having failed if the tip is bent into an offset position by more than 10% of propeller diameter D.

$$F_{ex} = \frac{300 c t^2 \sigma_{ref1}}{0.8D - 2r} \text{ [kN]} \tag{6.35}$$

where:

$\sigma_{ref1} = 0{,}6\sigma_{0,2} + 0{,}4\,\sigma_u$ [MPa]

σ_u (minimum ultimate tensile strength to be specified on the drawing) and

$\sigma_{0,2}$ (minimum yield or 0.2% proof strength to be specified on the drawing) are representative values for the blade material *c*, *t* and *r* are, respectively, the actual chord length, maximum thickness and radius of the cylindrical root section of the blade, which is the weakest section out-side the root fillet typically located at the point where the fillet terminates at the blade profile.

6.5.4.2 Spindle torque, Q_{sex}

The maximum spindle torque due to a blade failure load acting at 0.8*R* shall be determined. The force that causes blade failure typically reduces when moving from the propeller centre towards the leading and trailing edges. At a certain distance from the blade centre of rotation, the maximum spindle torque will occur. This maximum spindle torque shall be defined by an appropriate stress analysis or using the equation given below.

$$Q_{sex} = \max(C_{LE0.8};0{,}8C_{TE0.8})C_{spex}F_{ex} \text{ [kNm]} \tag{6.36}$$

where:

$$C_{spex} = C_{sp}C_{fex} = 0.7\left(1-\left(\frac{4EAR}{Z}\right)^3\right) \tag{6.37}$$

where:
C_{sp} is a non-dimensional parameter taking account of the spindle arm
C_{fex} is a non-dimensional parameter taking account of the reduction of the blade failure force at the location of the maximum spindle torque.
If C_{spex} is below 0.3, a value of 0.3 shall to be used for C_{spex}.
$C_{LE0.8}$ is the leading edge portion of the chord length at 0.8*R*
$C_{TE0.8}$ is the trailing edge portion of the chord length at 0.8*R*.

Figure 6-5 illustrates the spindle torque values due to blade failure loads across the entire chord length.

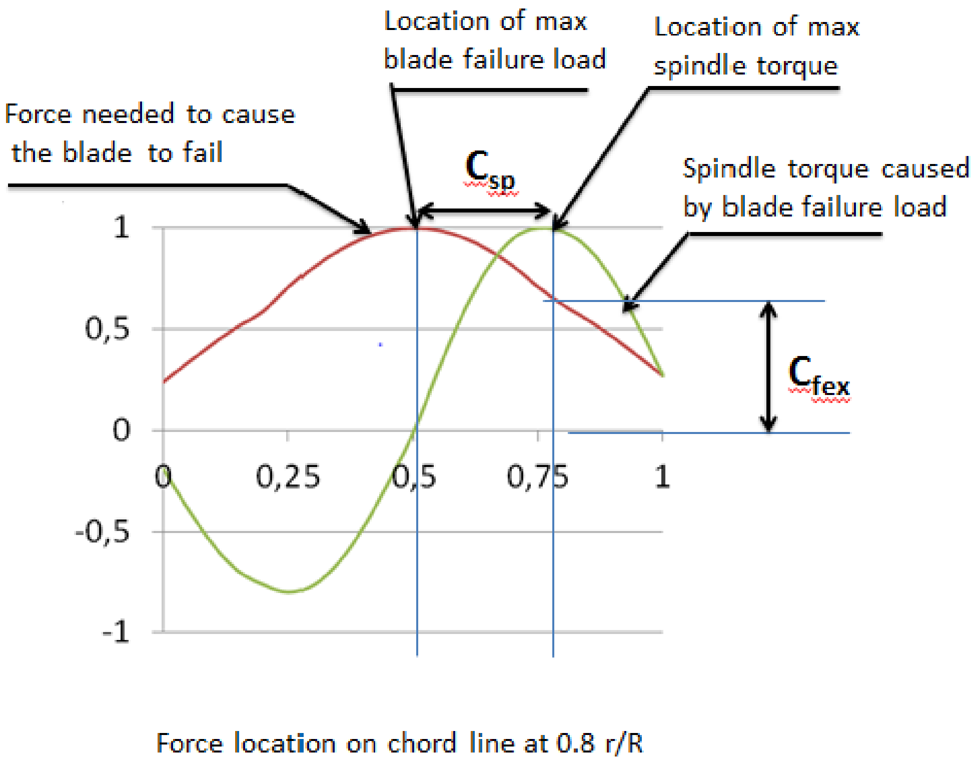


Figure 6-5.
Schematic figure showing a blade failure load and the related spindle torque when the force acts at a different location on the chord line at radius 0.8*R*

6.6 Design

6.6.1 Design principle

The strength of the propulsion line shall be designed according to the pyramid strength principle. This means that the loss of the propeller blade shall not cause any significant damage to other propeller shaft line components.

6.6.2 Propeller blade

6.6.2.1 Calculation of blade stresses

The blade stresses shall be calculated for the design loads given in section 6.5.1. Finite element analysis shall be used for stress analysis for the final approval of all propellers. The following simplified formulae can be used for estimating the blade stresses for all propellers at the root area (r/R < 0.5). Root area dimensions based on formula 6.38 can be accepted, even if the FEM analysis would show greater stresses at the root area.

$$\sigma_{st} = C_1 \frac{M_{BL}}{100ct^2} \text{ [MPa]} \tag{6.38}$$

where:

constant C_1 is the actual stress/stress obtained from the beam equation. If the actual value is not available, C_1 should have a value of 1.6.

$M_{BL} = (0.75 - r/R)RF$, for relative radius $r/R < 0.5$

F is the force F_b or F_f, whichever has greater absolute value.

6.6.2.2 Acceptability criterion

The following criterion for calculated blade stresses must be fulfilled:

$$\frac{\sigma_{ref2}}{\sigma_{st}} \geq 1.3 \tag{6.39}$$

where:

σ_{st} is the calculated stress for the design loads. If FEM analysis is used for estimating the stresses, von Mises stresses shall be used.

σ_{ref2} is the reference strength, defined as:

$\sigma_{ref\,2}$= 0,7σ_u or

σ_{ref2} = 0,6$\sigma_{0,2}$+0,4σ_u, whichever is lower.

6.6.2.3 Fatigue design of propeller blade

The fatigue design of the propeller blade is based on an estimated load distribution for the service life of the ship and the S-N curve for the blade material. An equivalent stress that produces the same fatigue damage as the expected load distribution shall be calculated and the acceptability criterion for fatigue should be fulfilled as given in section 6.6.2.4. The equivalent stress is normalised for 10^8 cycles.

For materials with a two-slope SN curve (Figure 6-6), fatigue calculations in accordance with this chapter are not required if the following criterion is fulfilled.

$$\sigma_{exp} \geq B_1 \sigma_{ref2}^{B_2} log(N_{ice})^{B_3} \tag{6.40}$$

where the B_1, B_2 and B_3 coefficients for open and ducted propellers are given in Table 6-13.

Table 6-13: Values of coefficients B_1, B_2 and B_3

	Open propeller	Ducted propeller
B_1	0.00328	0.00223
B_2	1.0076	1.0071
B_3	2.101	2.471

Note: above mentioned values of coefficients are applicable for ships contracted for construction on or after 1 January 2019.

For the calculation of equivalent stress, two types of SN curves are available.

Two slope SN curve (slopes 4.5 and 10), see Figure 6-6.

One slope SN curve (the slope can be chosen), see Figure 6-7.

The type of the SN-curve shall be selected to correspond with the material properties of the blade. If the SN-curve is unknown, the two slope SN curve shall be used.

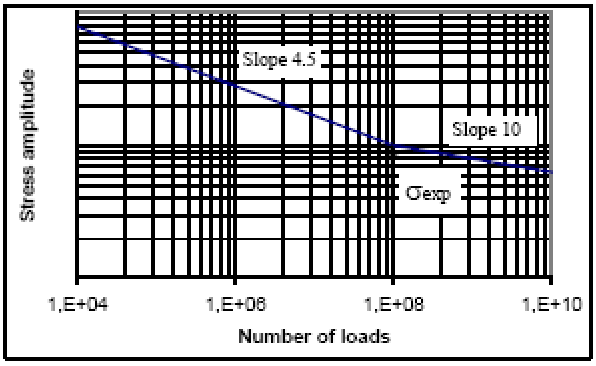


Figure 6-6.
Two-slope S-N curve.

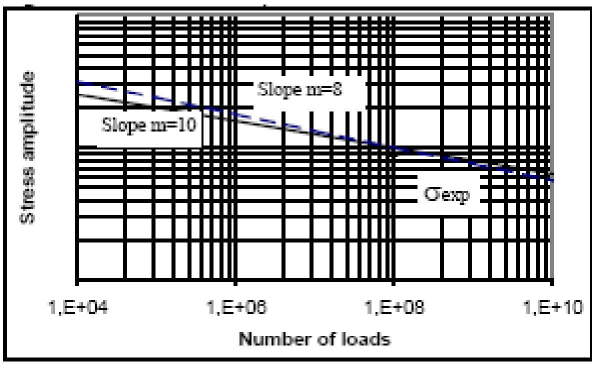


Figure 6-7.
Constant-slope S-N curve. Equivalent fatigue stress

The equivalent fatigue stress for 10^8 stress cycles, which produces the same fatigue damage as the load distribution for the service life of the ship, is:

$$\sigma_{fat}=\rho(\sigma_{ice})_{max}, \tag{6.41}$$

where:

$(\sigma_{ice})_{max} = 0,5((\sigma_{ice})_{fmax} - (\sigma_{ice})b_{max})$,

$(\sigma_{ice})_{max}$ is the mean value of the principal stress amplitudes resulting from design for-ward and backward blade forces at the location being studied.

$(\sigma_{ice})_{fmax}$ is the principal stress resulting from forward load.
$(\sigma_{ice})_{bmax}$ is the principal stress resulting from backward load.

In the calculation of $(\sigma_{ice})_{max}$, case 1 and case 3 (or case 2 and case 4) are considered a pair for $(\sigma_{ice})_{fmax}$, and $(\sigma_{ice})_{bmax}$ calculations. Case 5 is excluded from the fatigue analysis.

Calculation of parameter ρ for two-slope S-N curve

The parameter ρ relates the maximum ice load to the distribution of ice loads according to the regression formula:

$$\rho = C_1(\sigma_{ice})_{\max}^{C_2}\sigma_{fl}^{C_3}log(N_{ice})^{C_4} \tag{6.42}$$

where:

$$\sigma_{fl}=\gamma_{\epsilon1}\gamma_{\epsilon2}\gamma_v\gamma_m\sigma_{exp} \tag{6.43}$$

where:

$\gamma_{\epsilon1}$ is the reduction factor due to scatter (equal to one standard deviation)

$\gamma_{\epsilon2}$ is the reduction factor for test specimen size effect

γ_v is the reduction factor for variable amplitude loading

γ_m is the reduction factor for mean stress

σ_{exp} is the mean fatigue strength of the blade material at 10^8 cycles to failure in seawater.

The following values should be used for the reduction factors if actual values are unavailable:
$\gamma_\epsilon = \gamma_{\epsilon1}\gamma_{\epsilon2} = 0.67$, $\gamma_v = 0.75$, and $\gamma_m = 0.75$.

The coefficients C_1, C_2, C_3, and C_4 are given in Table 6-14. The applicable range of N_{ice} for calculating ρ is $5x10^6 \le N_{ice} \le 108$.

Table 6-14: Parameters for ρ determination.

	Open propeller	Ducted propeller
C_1	0.000747	0.000534
C_2	0.0645	0.0533
C_3	-0.0565	-0.0459
C_4	2.22	2.584

Calculation of parameterρ for constant-slope S-N curve

For materials with a constant-slope S-N curve – see Figure 6-7 – the factor ρ shall be calculated using the following formula:

$$\rho = \left(G\,\frac{N_{ice}}{N_R}\right)^{\frac{1}{m}}\left(ln(N_{ice})\right)^{-\frac{1}{k}} \tag{6.44}$$

where:

k shape parameter of the Weibull distribution;
$k = 1.0$ for ducted propellers;
$k = 0.75$ for open propellers.

N_R is the reference number of load cycles ($=10^8$).

The applicable range of N_{ice} for calculating ρ is $5x10^6 \le N_{ice} \le 10^8$.

Values for the parameter G are given in Table 6-15. Linear interpolation may be used to calculate the value of m/k ratios other than those given in Table 6-15.

Table 6-15: Value of the parameter G for different m/k ratios.

m/k	*G*	*m/k*	*G*	*m/k*	*G*
3	6	**6.5**	1871	**10**	$3.629{\cdot}10^6$
3.5	11.6	**7**	5040	**10.5**	$11.899{\cdot}10^6$
4	24	**7.5**	14 034	**11**	$39.917{\cdot}10^6$
4.5	52.3	**8**	40 320	**11.5**	$136.843{\cdot}10^6$
5	120	**8.5**	119 292	**12**	$479.002{\cdot}10^6$
5.5	287.9	**9**	362 880		
6	720	**9.5**	$1.133{\cdot}10^6$		

6.6.2.4 Acceptability criterion for fatigue

The equivalent fatigue stress at all locations on the blade must fulfil the following acceptability criterion:

$$\frac{\sigma_{fl}}{\sigma_{fat}} \geq 1.5 \tag{6.45}$$

where:

$$\sigma_{fl} = \gamma_{\epsilon 1}\gamma_{\epsilon 2}\gamma_v\gamma_m\sigma_{exp}$$

where:

$\gamma_{\epsilon 1}$ is the reduction factor due to scatter (equal to one standard deviation)
$\gamma_{\epsilon 2}$ is the reduction factor for test specimen size effect
γ_v is the reduction factor for variable amplitude loading
γ_m is the reduction factor for mean stress
σ_{exp} is the mean fatigue strength of the blade material at 10^8 cycles to failure in seawater.

The following values should be used for the reduction factors if actual values are unavailable:
$\gamma_\epsilon = \gamma_{\epsilon 1}\gamma_{\epsilon 2} = 0.67$, $\gamma_v = 0.75$, and $\gamma_m = 0.75$.

6.6.3 Propeller bossing and CP mechanism

The blade bolts, the CP mechanism, the propeller boss, and the fitting of the propeller to the propeller shaft shall be designed to withstand the maximum and fatigue design loads, as defined in section 6.5. The safety factor against yielding shall be greater than 1.3 and that against fatigue greater than 1.5. In addition, the safety factor for loads resulting from loss of the propeller blade through plastic bending, as defined in section 6.5.4, shall be greater than 1.0 against yielding.

6.6.4 Propulsion shaft line

The shafts and shafting components, such as the thrust and stern tube bearings, couplings, flanges and sealings, shall be designed to withstand the propeller/ice interaction loads as given in section 6.5. The safety factor must be at least 1.3 against yielding for extreme operational loads, 1.5 for fatigue loads and 1.0 against yielding for the blade failure load.

6.6.4.1 Shafts and shafting components

The ultimate load resulting from total blade failure, as defined in section 6.5.4 should not cause yielding in shafts and shaft components. The loading shall consist of the combined axial, bending, and torsion loads, wherever this is significant. The minimum safety factor against yielding must be 1.0 for bending and torsional stresses.

6.6.5 Azimuthing main propulsors

6.6.5.1 Design principle

In addition to the above requirements for propeller blade dimensioning, azimuthing thrusters must be designed for thruster body/ice interaction loads. Load formulae are given for estimating once in a lifetime extreme loads on the thruster body, based on the estimated ice condition and ship operational parameters. Two main ice load scenarios have been selected for defining the extreme ice loads. Examples of loads are illustrated in Figure 6-8. In addition, blade order thruster body vibration responses may be estimated for propeller excitation. The following load scenario types are considered:
Ice block impact on the thruster body or propeller hub
Thruster penetration into an ice ridge that has a thick consolidated layer
Vibratory response of the thruster at blade order frequency.

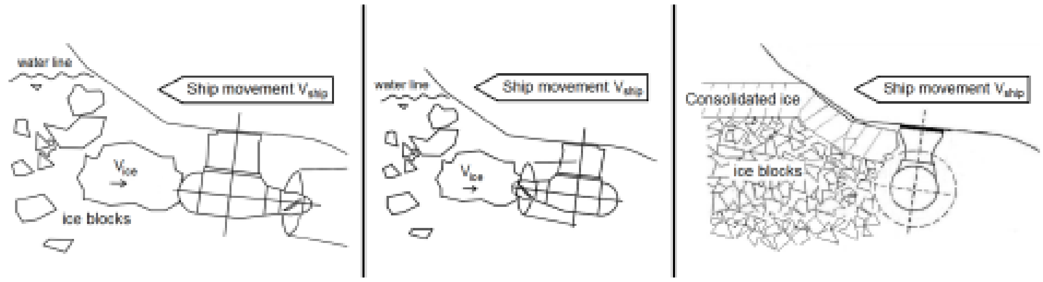


Figure 6-8.
Examples of load scenario types.

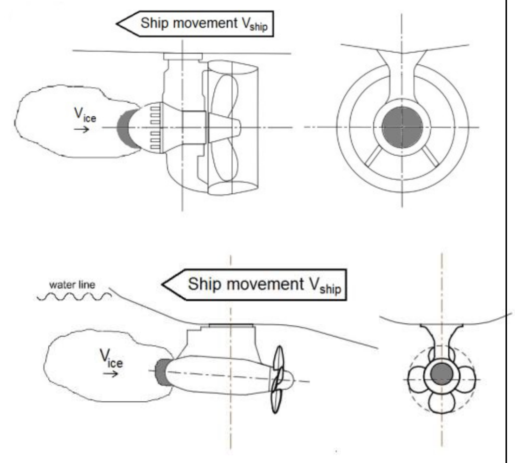
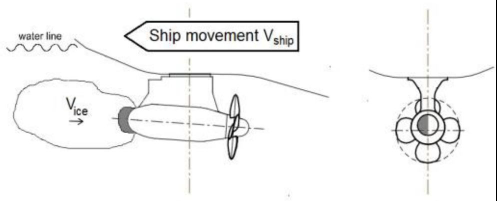
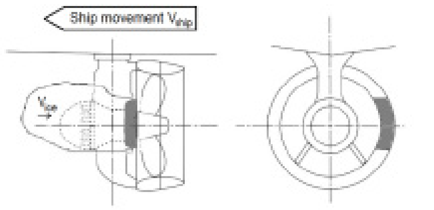
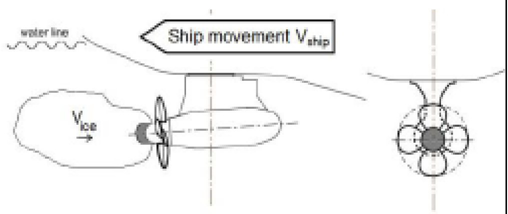
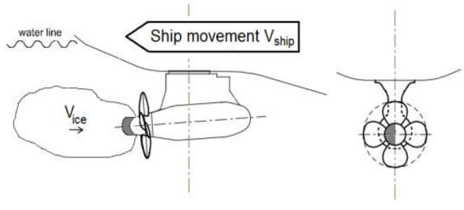
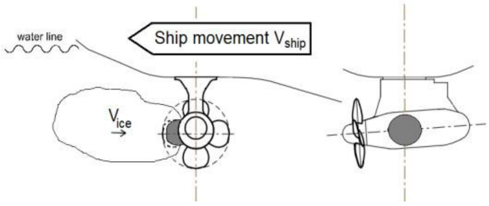
The steering mechanism, the fitting of the unit, and the body of the thruster shall be designed to withstand the plastic bending of a blade without damage. The loss of a blade must be taken into account for the propeller blade orientation causing the maximum load on the component being studied. Top-down blade orientation typically places the maximum bending loads on the thruster body.

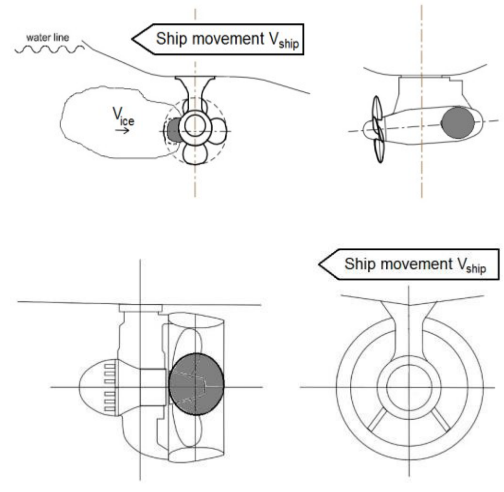
6.6.5.2 Extreme ice impact loads

When the ship is operated in ice conditions, ice blocks formed in channel side walls or from the ridge consolidated layer may impact on the thruster body and the propeller hub. Exposure to ice impact is very much dependent on the ship size and ship hull design, as well as the location of the thruster. The contact force will grow in terms of thruster/ice contact until the ice block reaches the ship speed.

The thruster must withstand the loads occurring when the design ice block defined in Table 6-3 impacts on the thruster body when the ship is sailing at a typical ice operating speed. Load cases for impact loads are given in Table 6-16. The contact geometry is estimated to be hemispherical in shape. If the actual contact geometry differs from the shape of the hemisphere, a sphere radius must be estimated so that the growth of the contact area as a function of penetration of ice corresponds as closely as possible to the actual geometrical shape penetration.

Table 6-16: Load cases for azimuthing thruster ice impact loads

	Force	Loaded area	
Load case T1a Symmetric longi- tudinal ice impact on thruster	F_{ti}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T1b Non-symmetric longitudinal ice impact on thruster	*50% of* F_{ti}	Uniform distributed load or uniform pressure, which are applied on the other half of the impact area.	
Load case T1c Non-symmetric longitudinal ice impact on nozzle	F_{ti}	Uniform distributed load or uniform pressure, which are applied on the impact area. Contact area is equal to the nozzle thickness (H_{nz})*the contact height (H_{ice}).	
Load case T2a Symmetric longi- tudinal ice impact on propeller hub	F_{ti}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T2b Non-symmetric longitudinal ice impact on propel- ler hub	*50% of* F_{ti}	Uniform distributed load or uniform pressure, which are applied on the other half of the impact area.	
Load case T3a Symmetric lateral ice impact on thruster body	F_{ti}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	

	Force	Loaded area	
Load case T3b Non-symmetric lateral ice impact on thruster body or nozzle	F_{ti}	Uniform distributed load or uniform pressure, which are applied on the impact area. Nozzle contact radius R to be taken from the nozzle length (Lnz).	

The ice impact contact load must be calculated using formula 6.46. The related parameter values are given in Table 6-17. The design operation speed in ice can be derived from Tables 6-18 and 6-19, or the ship in question's actual design operation speed in ice can be used. The longitudinal impact speed in Tables 6-18 and 6-19 refers to the impact in the thruster's main operational direction. For the pulling propeller configuration, the longitudinal impact speed is used for load case T2, impact on hub; and for the pushing propeller unit, the longitudinal impact speed is used for load case T1, impact on thruster end cap. For the opposite direction, the impact speed for transversal impact is applied.

$$F_{ti} = C_{DMI} 34.5 R_c^{0.5} (m_{ice} v_s^2)^{0.333} \text{ [kN]} \qquad (6.46)$$

where:

R_c is the impacting part sphere radius, see Figure 6-9 [m]
m_{ice} is the ice block mass [kg]
v_s is the ship speed at the time of contact [m/s]
C_{DMI} is the dynamic magnification factor for impact loads.
C_{DMI} shall be taken from Table 6-17 if unknown.

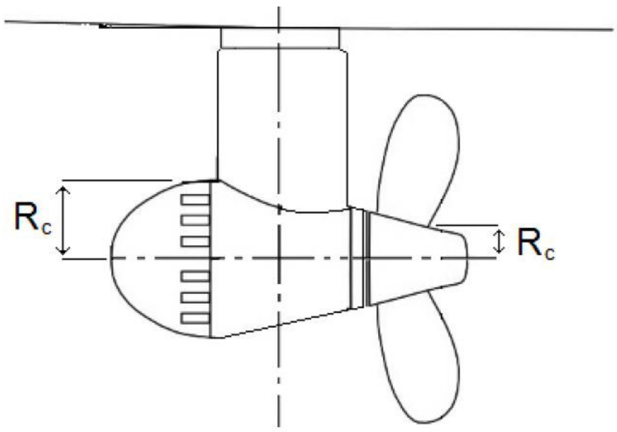


Figure 6-9.
Dimensions used for R_c

For impacts on non-hemispherical areas, such as the impact on the nozzle, the equivalent impact sphere radius must be estimated using the equation below.

$$R_{ceq} = \sqrt{\frac{A}{\pi}} \text{ [m]} \quad (6.47)$$

If the 2*R_{ceq} is greater than the ice block thickness, the radius is set to half of the ice block thickness. For the impact on the thruster side, the pod body diameter can be used as a basis for determining the radius. For the impact on the propeller hub, the hub diameter can be used as a basis for the radius.

Table 6-17: Parameter values for ice dimensions and dynamic magnification

	L1A	L1	L2	L3
Thickness of the design ice block impacting thruster (2/3 of H_{ice})	1.17 m	1.0 m	0.8 m	0.67 m
Extreme ice block mass ($mice$)	8670 kg	5460 kg	2800 kg	1600 kg
C_{DMI} (if not known)	1.3	1.2	1.1	1

Table 6-18: Impact speeds for aft centerline thruster

Aft centreline thruster				
Longitudinal impact in main operational direction	6 m/s	5 m/s	5 m/s	5 m/s
Longitudinal impact in reversing direction (pushing unit propeller hub or pulling unit cover end cap impact)	4 m/s	3 m/s	3 m/s	3 m/s
Transversal impact in bow first operation	3 m/s	2 m/s	2 m/s	2 m/s
Transversal impact in stern first operation (double acting ship)	4 m/s	3 m/s	3 m/s	3 m/s

Table 6-19: Impact speeds for aft wing, bow centerline and bow wing thrusters

Aft wing, bow centreline and bow wing thruster				
Longitudinal impact in main operational direction	6 m/s	5 m/s	5 m/s	5 m/s
Longitudinal impact in reversing direction (pushing unit propeller hub or pulling unit cover end cap impact)	4 m/s	3 m/s	3 m/s	3 m/s
Transversal impact	4 m/s	3 m/s	3 m/s	3 m/s

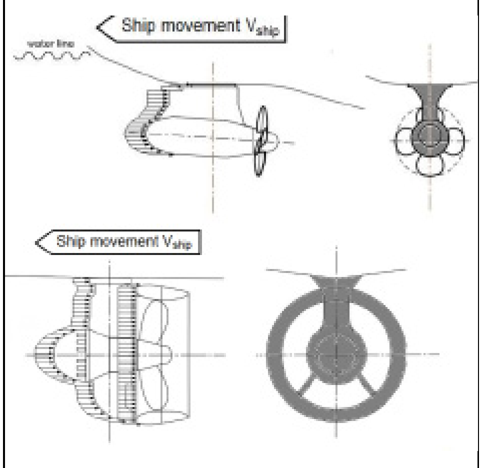
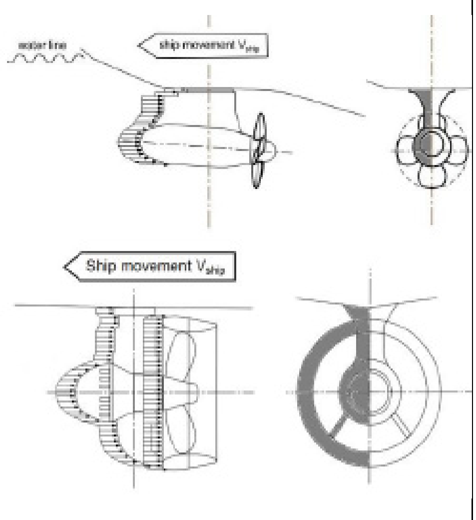
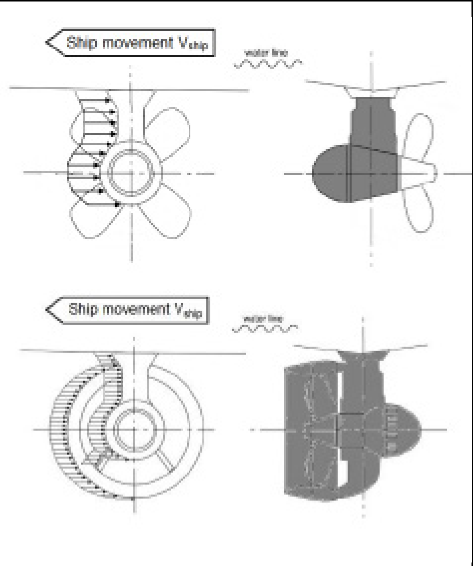
6.6.5.3 Extreme ice loads on thruster hull when penetrating an ice ridge

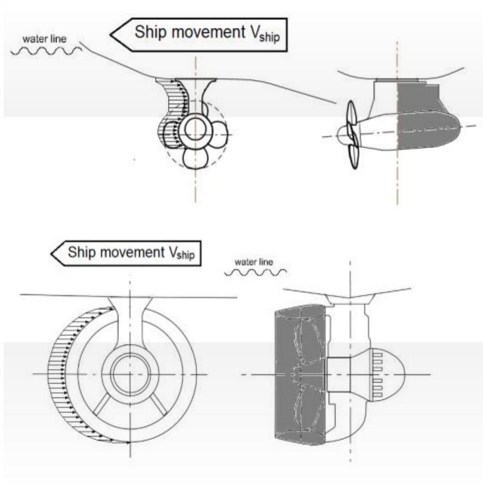
In icy conditions, ships typically operate in ice channels. When passing other ships, ships may be subject to loads caused by their thrusters penetrating ice channel walls. There is usually a consolidated layer at the ice surface, below which the ice blocks are loose. In addition, the thruster may penetrate ice ridges when backing. Such a situation is likely in the case of ships with notation **L1A** in particular, because they may operate independently in difficult ice conditions. However, the thrusters in ships with lower ice classes may also have to withstand such a situation, but at a remarkably lower ship speed.

In this load scenario, the ship is penetrating a ridge in thruster first mode with an initial speed. This situation occurs when a ship with a thruster at the bow moves forward, or a ship with a thruster astern moves in backing mode. The maximum load during such an event is considered the extreme load. An event of this kind typically lasts several seconds, due to which the dynamic magnification is considered negligible and is not taken into account.

The load magnitude must be estimated for the load cases shown in Table 6-20, using equation 6.48. The parameter values for calculations are given in Table 6-21 and Table 6-22. The loads must be applied as uniform distributed load or uniform pressure over the thruster surface. The design operation speed in ice can be derived from Table 6-21 or Table 6-22. Alternatively, the actual design operation speed in ice of the ship in question can be used.

Table 6-20: Load cases for ridge ice loads

Load case T4a Symmetric longitudinal ridge penetration loads	F_{tr}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T4b Non-symmetric longitudinal ridge penetration loads	50% of F_{tr}	Uniform distributed load or uniform pressure, which are applied on the other half of the contact area.	
Load case T5a Symmetric lateral ridge penetration loads for ducted azimuthing unit and pushing open propeller unit	F_{tr}	Uniform distributed load or uniform pressure, which are applied symmetrically on the contact area.	

Load case T5b Non-symmetric lateral ridge penetration loads for all azimuthing units	50% of F_{tr}	Uniform distributed load or uniform pressure, which are applied on the other half of the contact area.	

$$F_{tr} = 32 v_s^{0.66} H_r^{0.9} A_t^{0.74} [\text{kN}] \tag{6.48}$$

where v_s is ship speed [m/s]

H_r is design ridge thickness (the thickness of the consolidated layer is 18% of the total ridge thickness) [m]

A_t is the projected area of the thruster [m^2].

When calculating the contact area for thruster-ridge interaction, the loaded area in the vertical direction is limited to the ice ridge thickness, as shown in Figure 6-10**.**

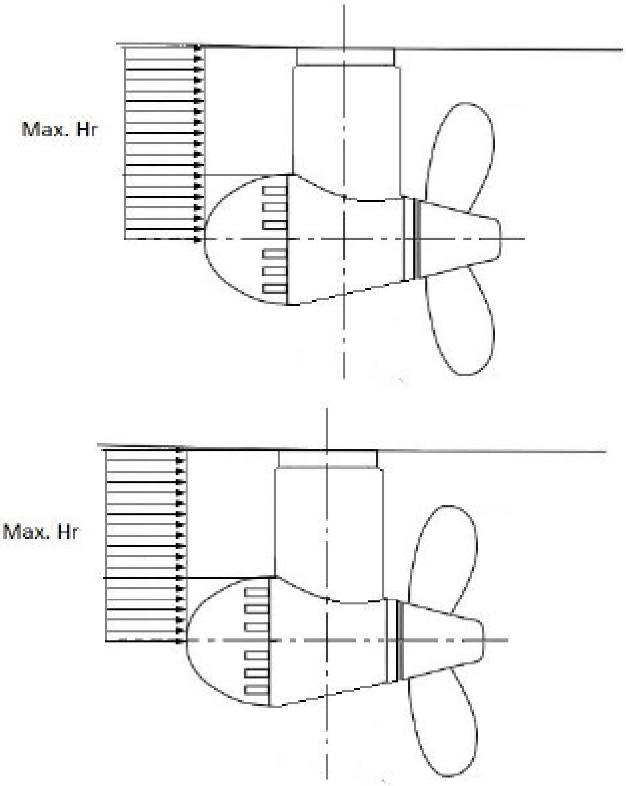


Figure 6-10.
Schematic figure showing the reduction of the contact area by the maximum ridge thickness

Table 6-21:
Parameters for calculating maximum loads when the thruster penetrates an ice ridge. Aft thrusters. Bow first operation

	L1A	L1	L2	L3
Thickness of the design ridge consolidated layer	1.5 m	1.5 m	1.2 m	1.0 m
Total thickness of the design ridge, H_r	8 m	8 m	6.5 m	5 m
Initial ridge penetration speed (longitudinal loads)	4 m/s	2 m/s	2 m/s	2 m/s
Initial ridge penetration speed (transversal loads)	2 m/s	1 m/s	1 m/s	1 m/s

Table 6-22:
Parameters for calculating maximum loads when the thruster penetrates an ice ridge. Thruster first mode such as double acting ships.

	L1A	L1	L2	L3
Thickness of the design ridge consolidated layer	1.5 m	1.5 m	1.2 m	1.0 m
Total thickness of the design ridge, H_r	8 m	8 m	6.5 m	5 m
Initial ridge penetration speed (longitudinal loads)	6 m/s	4 m/s	4 m/s	4 m/s
Initial ridge penetration speed (transversal loads)	3 m/s	2 m/s	2 m/s	2 m/s

6.6.5.4 Acceptability criterion for static loads

The stresses on the thruster must be calculated for the extreme once-in-a-lifetime loads described in section 6.6.5. The nominal von Mises stresses on the thruster body must have a safety margin of 1.3 against the yielding strength of the material. At areas of local stress concentrations, stresses must have a safety margin of 1.0 against yielding. The slewing bearing, bolt connections and other components must be able to maintain operability without incurring damage that requires repair when subject to the loads given in sections 6.6.5.2 and 6.6.5.3 multiplied by a safety factor of 1.3.

6.6.5.5 Thruster body global vibration

Evaluating the global vibratory behavior of the thruster body is important, if the first blade order excitations are in the same frequency range with the thruster global modes of vibration, which occur when the propeller rotational speeds are in the high power range of the propulsion line. This evaluation is mandatory and it must be shown that there is either no global first blade order resonance at high operational propeller speeds (above 50% of maximum power) or that the structure is designed to withstand vibratory loads during resonance above 50% of maximum power.

When estimating thruster global natural frequencies in the longitudinal and transverse direction, the damping and added mass due to water must be taken into account. In addition to this, the effect of ship attachment stiffness must be modelled.

6.7 Alternative design procedure

6.7.1 Scope

As an alternative to sections 6.5 and 6.6, a comprehensive design study may be performed to the satisfaction of the PRS. The study must be based on the ice conditions given for different ice classes in section 6.3. It must include both fatigue and maximum load design calculations and fulfil the pyramid strength principle, as given in section 6.6.1.

6.7.2 Loading

Loads on the propeller blade and propulsion system shall be based on an acceptable estimation of hydrodynamic and ice loads.

6.7.3 Design levels

The analysis must confirm that all components transmitting random (occasional) forces, excluding propeller blade, are not subjected to stress levels in excess of the yield stress of the component material, with a reasonable safety margin.

Cumulative fatigue damage calculations must give a reasonable safety factor. Due account must be taken of material properties, stress raisers, and fatigue enhancements.

A vibration analysis must be performed and demonstrate that the overall dynamic system is free of the harmful torsional resonances resulting from propeller/ice interaction.

7 MISCELLANEOUS MACHINERY REQUIREMENTS

7.1 Starting arrangements

The capacity of the air receivers must be sufficient to provide no less than 12 consecutive starts of the propulsion engine without reloading, if it has to be reversed for moving astern, or 6 consecutive starts if the propulsion engine does not have to be reversed for moving astern.

If the air receivers serve any purposes other than starting the propulsion engine, they must have additional capacity sufficient for such purposes.

The capacity of the air compressors must be sufficient for charging the air receivers from atmospheric to full pressure in one (1) hour, except for a ship with ice class **L1A**, if its propulsion engine has to be reversed for going astern, in which case the compressor must be able to charge the receivers in half an hour.

7.2 Sea inlet and cooling water systems

The cooling water system shall be designed to secure the supply of cooling water when navigating in ice.

For this purpose, at least one cooling water inlet chest shall be arranged as follows:

The sea inlet shall be situated near the centreline of the ship and well aft, if possible.

Guidance for designing the volume of the chest shall be around one cubic metre for every 750 kW in engine output of the ship, including the output of auxiliary engines necessary for the operation of the ship.

The chest shall be sufficiently high to allow ice to accumulate above the inlet pipe.

A pipe for discharge cooling water, allowing full capacity discharge, shall be connected to the chest.

The open area of the strainer plates shall be no less than four (4) times the inlet pipe sectional area.

If there are difficulties in meeting the requirements of paragraphs 2 and 3 above, two smaller chests may be arranged for alternating the intake and discharge of cooling water. Otherwise, the arrangements and situation shall be as above.

Heating coils may be installed in the upper part of the sea chest.

Arrangements for using ballast water for cooling purposes may be useful as a reserve in terms of ballast, but cannot be accepted as a substitute for an inlet chest as described above.

Annex I

Parameters and calculated minimum engine power for sample ships

For checking the results of calculated powering requirements, Table I-1 presents input data for a number of sample ships.

Table I-1:
Parameters and calculated minimum engine power of sample ships

Sample ship No.										
		1	2	3	4	5	6	7	8	9
PRS Ice class		**L1A**	**L1**	**L2**	**L3**	**L1A**	**L1A**	**L1**	**L1**	**L2**
α	[degree]	24	24	24	24	24	24	36	20	24
φ1	[degree]	90	90	90	90	30	90	30	30	90
φ2	[degree]	30	30	30	30	30	30	30	30	30
L	[m]	150	150	150	150	150	150	150	150	150
B	[m]	25	25	25	25	25	22	25	25	25
T	[m]	9	9	9	9	9	9	9	9	9
``+L_{BOW}	[m]	45	45	45	45	45	45	45	45	45
L_{PAR}	[m]	70	70	70	70	70	70	70	70	70
A_{wf}	[m^2]	500	500	500	500	500	500	500	500	500
D_P	[m]	5	5	5	5	5	5	5	5	5
Prop. No / type		1/CP	1/CP	1/CP	1/CP	1/CP	1/CP	1/CP	1/CP	1/FP
New ships (section 3.2.2)	[kW]	7840	4941	3478	2253	6799	6406	5343	5017	3872
Existing ships (section 3.2.4)	[kW]	9192	6614			8466	7645	6614	6614	

Annex II

Required engine output for a ship of ice class L2 or L3, the keel of which has been laid or which was at a similar stage of construction before 1 September 2003

The engine output shall be no less than that determined by the formula below and in no case less than 740 kW for ice classes L2 and L3.

$$P = f_1 f_2 f_3 (f_4 \Delta + P_0) \text{ [kW]},$$

where:

f_1 = 1.0 for a fixed pitch propeller
= 0.9 for a controllable pitch propeller
$f_2 = \varphi 1/200 + 0.675$ but not more than 1.1,

where:

φ_1 is the rake of the stam at centerline [degrees] (see Figure 3-1).

The product $f_1 f_2$ shall not be less than 0.85.
f_2 = 1.1 for a bulbous bow
$f_3 = 1.2B/\Delta^{1/3}$ but no less than 1.0
f_4 and P_0 are as follows:

PRS Ice class	L2	L3	L2	L3
Displacement	$\Delta < 30\,000$		$\Delta \geq 30\,000$	
f_4	0.22	0.18	0.13	0.11
P_0	370	0	3070	2100

$\Delta$is displacement [t] of the ship at the maximum ice class draught according to 2.1. This need be no greater than 80.000 t.

Annex III

Ice class draught marking

Subject to section 2.2, the ship's sides must be provided with a warning triangle and a draught mark at the maximum permissible ice class draught amidships (see Figure III-1). The purpose of the warning triangle is to provide information on the draught limitation of the vessel, when sailing in ice, for the masters of icebreakers and for inspection personnel in ports.

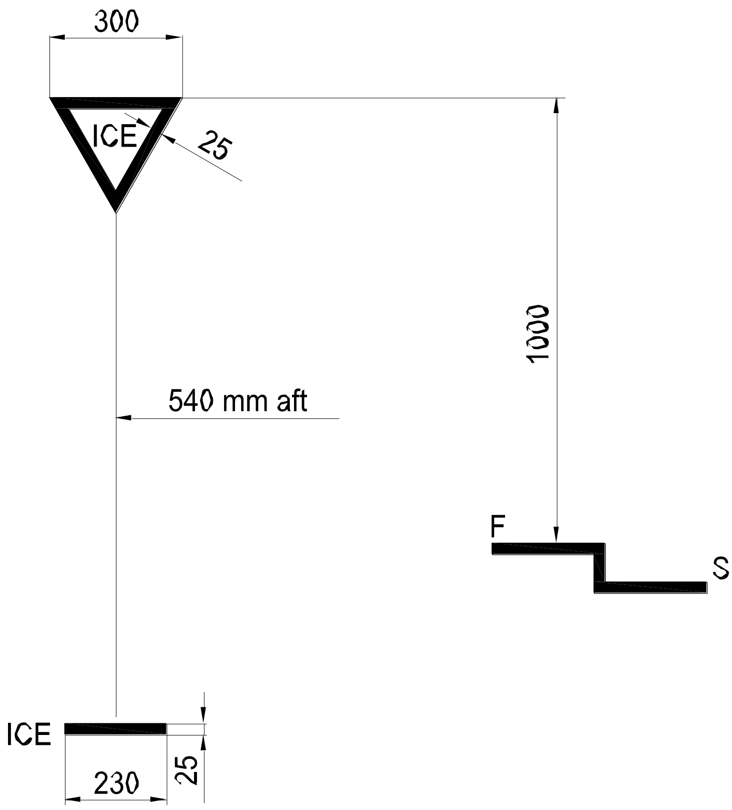


Figure III-1.
Ice Class Draught Marking

Notes to Figure III-1

The upper edge of the warning triangle must be located vertically above the "ICE" mark, 1.000 mm higher than the Summer Load Line in fresh water but in no case higher than the deck line. The sides of the triangle must be 300 mm in length.

The ice class draught mark must be located 540 mm abaft of the centre of the load line ring or 540 mm abaft of the vertical line of the timber load line mark, if applicable.

The marks and figures must be cut out of 5 - 8 mm plate and then welded to the ship's side. The marks and figures must be painted in a red or yellow reflecting colour in order to make the marks and figures plainly visible, even in ice conditions.

The dimensions of all letters must be the same as those used in the load line mark.

SECTION 2

GUIDELINES FOR THE APPLICATION OF PRS ICE CLASS REGULATIONS, 2017

PRS Note:

Guidelines contain implemented changes to the November 2017 version:

- New section 3.3 "Information needed for confirming a Finnish-Swedish ice class"
- New text added to section 4, "The purpose and scope of the rules" to clarify the aims and assumptions of the rules
- New subsections in section 10, "Propulsion machinery", and subsequent renumbering of subsections. Subsections 10.1, 10.3, 10.4 and 10.6 are new. Previous subsections 10.1, 10.2, 10.3, 10.4 and 10.5 have been renumbered as 10.2, 10.5, 10.7, 10.8 and 10.9 respectively

1 GOAL OF THE GUIDELINES

The purpose of these Guidelines is to provide classification societies, ship designers and shipyards with background information on the ideas behind the rules, to provide a harmonised interpretation for the implementation of certain parts of the rules, and to provide guidance on certain aspects of the winterisation of ships, which are not covered by the rules.

2 THE STATUS OF THE GUIDELINES

In general, PRS accepts class approval, based on these Guidelines, for the design of vessels. The approval is required for the engine power of a vessel if the engine power is determined using model tests or by means other than the formulae given in regulation 3.2.5 of Section I. Instructions for the application of a letter of compliance are given in Appendix 1. Model tests performed for vessels contracted for construction on or after 1 January 2012 should be completed according to these Guidelines.

These Guidelines replace all Guidelines previously issued by PRS.

3 IMPLEMENTATION OF PRS REGULATIONS AND ITS HARMONIZATION WITH FINNISH-SWEDISH ICE CLASS RULES (FSICR) IN FINLAND AND SWEDEN

The Finnish and Swedish administrations provide icebreaker assistance to ships bound for ports in these two countries during the winter season. Depending on the ice conditions, restrictions are enforced with regard to the size and ice class of ships entitled to icebreaker assistance. Winter traffic restrictions on ships are set in order to ensure smooth winter navigation and the safety of navigation in ice. Assistance for ships with inadequate engine output or ice strengthening would be both difficult and time-consuming. It would also be unsafe to expose such vessels to ice loads and ice pressure.

The traffic restrictions are modified during the winter period, depending on the ice conditions. A typical maximum traffic restriction on ships bound for Finnish ports in the eastern Gulf of Finland is at least ice class IA (**L1**) and a minimum deadweight of 2.000 TDW. A typical strictest traffic restriction for the ports in the northern Bay of Bothnia is ice class IA (**L1**) and a minimum deadweight of 4,000 TDW. On the other hand, a lower minimum ice class is required for ships bound for ports on the south-western coast of Finland, where the ice conditions are less difficult. A typical minimum requirement is ice class IC **(L3**) and a deadweight of 3.000 TDW.

3.1 Compliance of PRS Ice Class Certificate with FSICR issued by TRAFI

Since 1 January 2006, Ice Class Certificates have no longer been issued in Finnish ports by the inspectors of the Finnish Transport Safety Agency. The compliance of ship Ice Class with FSICR will be determined on the basis of PRS Classification Certificate.

3.2 Information Needed for Confirming PRS Ice Class

Polish authorities determine PRS ice class of a vessel based on the PRS Classification Certificate of the vessel. As icebreaker assistance is given free of charge to vessels fulfilling the ice class requirements for a given time and port, it is important that the authorities have sufficient information about the ice class of the vessel. The minimum and maximum ice class draughts fore, amidships and aft are needed for the icebreaker crews to ascertain that vessels are loaded safely for traffic in ice. The minimum power requirement is important to ascertain that the vessel has sufficient capability for operating in ice.

A vessel may be denied icebreaker assistance if the required information for determining PRS ice class is not available. If this information can be found on the PRS Classification Certificate or on an annex to the PRS Classification Certificate, it will ensure that PRS ice class of the vessel can be confirmed without delay.

Vessels of PRS ice class IA Super and IA, which have had their keel laying on 1 September 2003 or before, shall fulfil the machinery power requirements of the current Regulations in order to retain their PRS ice class from 1 January of the year on which 20 years have elapsed since their commissioning. The new machinery power requirement will be confirmed for each ship separately PRS. If it is stated on the PRS Classification Certificate or its annex that the power requirement is calculated according to the current Regulations, no further confirmation is needed.

3.3 Information needed for confirming a Finnish-Swedish ice class

Both Finnish and Swedish authorities determine the compliance with Finnish-Swedish ice class of a vessel based on the PRS Classification Certificate of the vessel. As icebreaker assistance is given free of charge to vessels fulfilling the ice class requirements for a given time and port, it is important that the authorities have sufficient information about the ice class of the vessel. The minimum and maximum ice class draughts fore, amidships and aft are needed for the icebreaker crews to ascertain that vessels are loaded safely for traffic in ice. The minimum power requirement is important to ascertain that the vessel has sufficient capability for operating in ice.

A vessel may be denied icebreaker assistance if the required information for determining a Finnish-Swedish ice class is not available. If this information can be found on the PRS Classification Certificate or on an annex to the Classification Certificate, it will ensure that the compliance with Finnish-Swedish ice class of the vessel can be confirmed without delay.

Vessels of PRS Ice class compliant with Finnish-Swedish ice class IA Super and IA, which have had their keel laying on 1 September 2003 or before, shall fulfil the machinery power requirements of the current PRS Rules in order to retain their Finnish-Swedish ice class from 1 January of the year on which 20 years have elapsed since their commissioning. The new machinery power requirement will be confirmed for each ship separately by the Finnish and Swedish authorities. If it is stated on the PRS Classification Certificate or its annex that the power requirement is calculated according to the current PRS Rules, no further confirmation of compliance with FSICR is needed.

4 THE PURPOSE AND SCOPE OF THE REGULATIONS

The PRS Ice Class Regulations are primarily intended for the design of merchant ships trading in the Northern Baltic in winter. The regulations primarily address matters directly relevant to the capability of a ship to advance in ice. The regulations for minimum engine output (Chapter 3 of the Regulations -Section I) can be considered regulations of an operational type. Ships are required to have a certain speed in a brash ice channel, in order to ensure the smooth progress of traffic in ice conditions. The regulations for strengthening the hull, rudder, propellers, shafts and gears (Chapters 4 to 6 of the Regulations – Section I) are clearly related to the safety of navigation in ice. In principle, all parts of the hull and the propulsion machinery exposed to ice loads must be ice-strengthened.

The Regulations implicitly assume that ice classed ships are at least about 2,000 TDW as this is the minimum deadweight for ships given ice breaker assistance in the case a deadweight limit is set. Smaller ships can be designed according to the rules but some of the requirements may not be practical for very small vessels. The rules are meant for the design of cargo and passenger vessels and other types of vessels such as tugs and icebreaking supply vessels are not explicitly taken into account.

Finnish and Swedish ice classes are determined solely for determining which ships are eligible for ice-breaker assistance to Finnish and Swedish ports and to determine the fairway dues for ships calling to Finnish ports.

4.1 Design Philosophy

The PRS Ice Class Regulations are intended for the design of merchant ships operating in first-year ice conditions during part of the year. In most cases, compromises have to be made when ships are designed both for open water and ice conditions. The basic philosophy underlying the regulations is to require a certain minimum engine power for ships in any ice class, for operational reasons. However, no general requirements have been set for the hull form. The structural strength of the hull and the propulsion machinery should be able to withstand ice loads, with a minimum safety margin. For economic reasons, excessive ice strengthening is avoided.

The PRS Ice Class Regulations set the minimum requirements for engine power and ice strengthening for ships based on the assumption that icebreaker assistance is available when required. Special consideration should be given to ships designed for independent navigation in ice, or for ships designed for navigation in sea areas other than the Baltic Sea.

The design points for hull and propulsion machinery, as well as for the ice performance (propulsion power), are all different. This reflects the fact that different ice conditions in different ship operations form the critical design situations. The design points are as follows:

Item	Design point in FSICR	Description of the design point
Hull	Impact with level ice of thickness h_0	The ship can encounter thick level ice in ridges where the consolidated layer can be 80% thicker than the level ice thickness. Channel edges can also be very thick.
Propulsion machinery	Impact with large ice floes of thickness H_{ice}	Propellers encounter only broken ice and the design scenario involves an impact with these floes. Large ice floes can be encountered among level ice floes, for example in old channels.
Propulsion power	Ship must make at least 5 knots in a brash ice channel of thickness H_M	Ships must be able to follow icebreakers at a reasonable speed and proceed independently in old brash ice channels at reasonable speeds.

4.1.1 The Engine Power Regulations

The regulations for minimum engine output are based on long term experience of Finnish and Swedish icebreaker assistance in the northern Baltic Sea area. The number of icebreakers is limited, and they must be able to assist all ships entering or leaving winter ports. Thus, the minimum engine power requirement is “a matter of definition” to be decided by the Maritime Authorities depending on the number of icebreakers, the number of ships in need of assistance, the ice conditions, and the maximum waiting time for icebreaker assistance. In Finland, the maximum average waiting time for icebreaker assistance is defined as about four hours.

The principle underlying the winter navigation system is that all ships meeting the traffic restrictions are given icebreaker assistance. An ice-classed ship is assisted by an icebreaker when the ship is stuck in ice or is in need of assistance, because her speed has substantially decreased. Normally, the ship is assisted to (or from) the fairway entrance after which the ship should be able to sail into port on its own (or sail out of the port on its own), although the icebreaker often has to escort smaller ships, in particular, into the port. Most of the fairways leading to Finnish coastal ports are routed through the archipelago area. In archipelago areas, the ice cover is stationary. The engine power requirements included in the rules have been developed for navigation in brash ice channels in archipelago areas, at a minimum speed of 5 knots. Thus, it should not be assumed that mere compliance with these regulations guarantees a certain degree of capability to advance in ice without icebreaker assistance, or to withstand heavy ice compression in the open sea, where the ice field may move due to high wind speeds. It should be also noted that the ice-going capacity of small ships may be somewhat lower than that of larger ships in the same ice class. This observation, which is based on the experience of icebreaker operators, may be partly attributed to the beneficial effect of greater inertia during ice going.

4.1.2 Hull Structural Design

The regulations for the structural design of hulls (Chapter 4 of the Regulations – Section I) deal with the local strength of the hull (plating, frames, stringers and web frames). The ice loads given in the Regulations have been determined based on measurements taken on ships that sail in the Baltic Sea in winter. The regulations do not take account of a situation where a ship is stuck in compressive and/or moving ice and large ice forces are acting on the parallel midbody. It is assumed that icebreaker assistance is available in such cases, leaving no time for a serious compressive situation to develop. However, in the experience of the Administrations, vessels strengthened to ice classes **L1** and **L1A** are rarely damaged in compressive ice situations. Ice damage on the midbody of ships in ice class IC has been observed in recent years.

Recent observations of ice damage on ice-strengthened vessels indicate that most of the damage on hulls occurs at an early stage of the winter season. These ships are probably operated at high speed on the open sea when the ice coverage is less than 100%. Damage to the hull may therefore occur when the vessel hits an ice floe at high speed.

4.1.3 Propeller, Shafts and Gears

The "pyramid strength" principle, i.e. the hierarchical strength principle has been adopted for the design of propulsion systems. This means that the propeller blades are the weakest element in the propulsion train and the strength increases towards the main engine or propulsion motor.

Recent observations of ice damage on ice-strengthened vessels indicate that most damage to propellers occurs at a later stage of the winter season than damage occurring on the hull. Obviously, thick ice blocks place the largest loads on propellers.

4.1.4 Application of the Regulations on the Design of Ships for Other Sea Areas

If the PRS Ice Class Regulations are applied to the design of ships for other sea areas, the following issues should be taken into consideration:

The PRS Ice Class Regulations have been developed for first year ice conditions with a maximum level ice thickness of 1.0 m, an ice bending strength (cantilever beam test) of about 500 kPa and a maximum compressive strength of sea ice of around 5 MPa.

Consideration should be taken of ice compression in the sea area.

There is no ocean swell in the Baltic Sea. The vertical extension of the ice belt in the bow area may therefore not be adequate, if the vessel is operated in an area with high swell and floating ice.

5 GENERAL (CHAPTER 1 OF THE REGULATIONS – SECTION I)

5.1 Ice Classes

Ships in ice class **L1A** are intended for year round operation in the Baltic Sea area. This means that the Administrations do not set traffic restrictions for this ice class. Ships in ice class **L1** are intended for year-round operation in the Baltic Sea area, and are escorted if necessary.

Ships in ice class **L2** or IC may have limited access to Finnish and Swedish ports for part of the year, depending on the ice conditions.

Ships belonging to ice classes II and III are not strengthened for navigation in ice. Traffic restrictions based on ice class, deadweight and possibly power are given according to ice conditions. In Finland, the fairway dues depend on the ice class of the vessel, for which reason "ice classes" II and III are used.

6 ICE CLASS DRAUGHT (CHAPTER 2 OF THE REGULATIONS – SECTION I)

The UIWL and LIWL waterlines may be broken lines, as these are the envelopes of all permitted load situations. The forward design draught should never be less than the draught amidships. The same draught used for calculating the minimum engine power of the ship (see paragraph 3.2.2 of the Regulations), should be used in determining the vertical extension of ice strengthening (see 4.3.1 and 4.4.1 of the Regulations).

It is recommended that, at the design stage, some reserve is allowed for the ice class draughts UIWL and LIWL. If this is done, the engine power of the vessel, as well as the vertical extension of the ice belt, will continue to fulfil the rule requirements in the future, if the UIWL draught of the ship is increased or the LIWL draught is decreased when the ship is in operation.

It has been observed that ships in light load condition require more icebreaker assistance than in their fully loaded condition. Thus, ships should always be operated so that the waterline is between UIWL and LIWL, preferably closer to UIWL. Consideration should also be given to operating with the deepest possible propeller submergence.

7 ENGINE OUTPUT (CHAPTER 3 OF THE REGULATIONS – SECTION I)

7.1 Definitions (Section 3.2.1 of the Regulations)

The length of the bow (LBOW) should be measured between the forward border of the side where the waterlines are parallel to the centreline and the fore perpendicular at UIWL. The same perpendicular should also be used when calculating the length of the bow at LIWL.

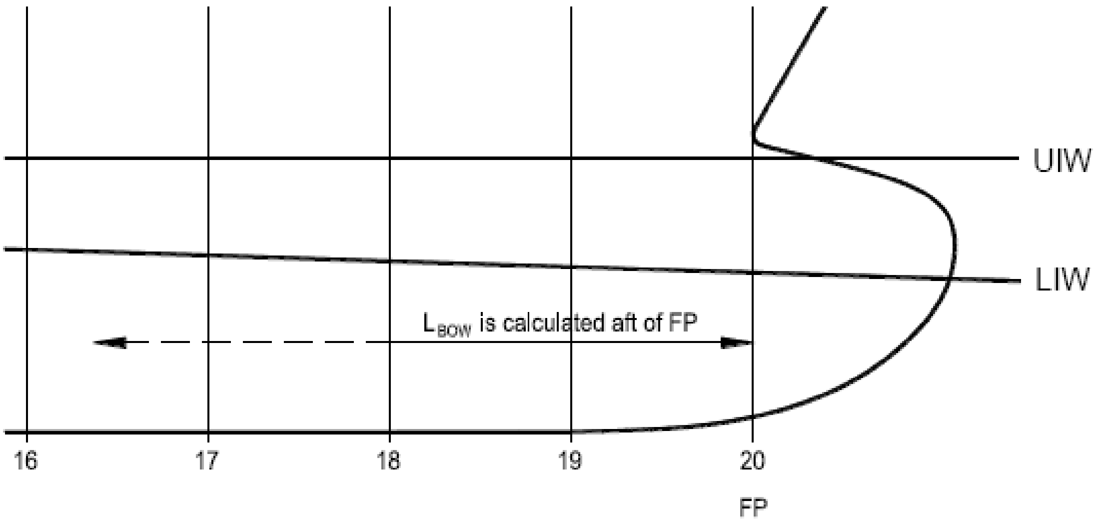


Figure 1.
Measurement of the length of the bow.

The length of the parallel midship (LPAR) should be measured from the aft perpendicular if the section of the side where the waterlines are parallel to the centreline extends aft of the aft perpendicular.

No negative values of the rake of the bow at B/4 (φ_2) should be used in the calculations. If the rake of the bow has a negative value, as presented in Figure 2 below, 90 degrees should be used in the calculations.

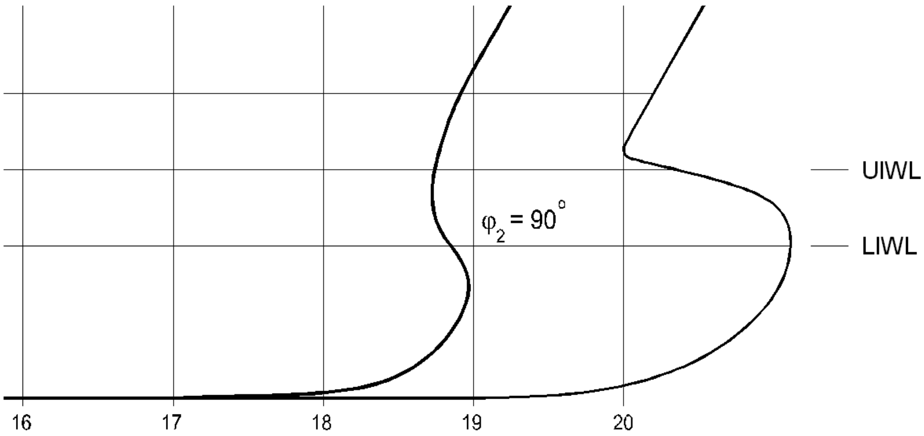


Figure 2.
Determination of the angle φ_2

7.2 Existing Ships of Ice Class L2 or IC (Section 3.2.3 of the regulations)

To be entitled to retain ice class L2 or L3, a ship, the keel of which has been laid or which has been at a similar stage of construction on or after 1 November 1986, but before 1 September 2003, should comply with the requirements of Chapter 3 of the ice class regulations of 1985, as amended. If the owner of the ship so requests, the required minimum engine power can be determined in accordance with the ice class regulations of 2017.

To be entitled to retain ice class L2 or L3, a ship, the keel of which has been laid or which has been at a similar stage of construction before 1 November 1986, should comply with regulation 3 of the ice class regulations of 1971 (Board of Navigation Rules for Assigning Ships Separate Ice-Due Classes, issued on 6 April1971), as amended. If the owner of the ship so requests, the required minimum engine power can be determined in accordance with the ice class regulations of 1985 or 2017.

7.3 On the Selection of the Propulsion System

The following machinery systems are used in ice-going ships:
– Diesel – electric propulsion system;
– Medium speed diesel and gearbox;
– Low-speed diesel with direct shaft.
– The propulsors may include:
– Controllable pitch or fixed pitch propellers;
– Contra-rotating and tandem propellers in the azimuths
– Podded or azimuthing propulsors.

A diesel-electric (or steam/gas turbine-electric) propulsion system is very common in icebreakers, but not in merchant vessels. It provides very efficient propulsion characteristics at slow speed and excellent manoeuvring characteristics, but due to its high cost it is very seldom used in merchant vessels. The capability for fast load and direction changes, the ability to maintain RPM and good reversing capability are the characteristics of good propulsion systems in ice.

A propulsion system with a medium-speed engine, a gearbox and a controllable pitch (CP) propeller is the most common propulsion system used in merchant vessels with an ice class. This provides reasonable propulsion characteristics at slow speed, as well as reasonable manoeuvring characteristics.

A direct driven diesel engine with a fixed pitch propeller provides poor propeller thrust at a low ship speed. It is recommended that a controllable pitch propeller be installed on ships with a direct driven diesel engine propulsion system.

7.4 Other Methods of Determining K_e or R_{ch}

According to section 3.2.5 *"for an individual ship, in lieu of the K_e or R_{CH} values defined in 3.2.2 and 3.2.3, the use of K_e or R_{CH} values based on more exact calculations or values based on model tests may be approved"*. Guidelines on these issues are given in the following paragraphs.

If R_{CH} is determined using the rule formulae, then K_e can be determined by using direct calculations or the rule formulae. However, if R_{CH} is determined using model tests then propeller thrust should be calculated by direct calculations using the actual propeller data, instead of using the rule formulae. The reason for this is the need to ensure that the propulsion system is able to produce the required thrust to overcome the channel resistance. It should be noted that the total resistance in ice, R_{iTOT}, is the sum of open water resistance R_{OW} and ice resistance R_{CH} i.e. $R_{iTOT} = R_{OW} + R_{CH}$ where the ice resistance R_{CH} is given in the ice class regulations, eq. (3.2).

7.4.1 Other Methods of Determining K_e

One of the drawbacks of the Rule equations for power is that open water resistance is included in a very approximate fashion (the net thrust i.e. the thrust available for overcoming the ice resistance at 5 knots is assumed to be $0.8T_B$). This is not correct, as open water and ice resistance are unrelated in the case of the various ice thicknesses related to each ice class. A brief study of the open water and channel resistance of typical ships in different ice classes has led to the proposal of a regression formula whose results are given in Table 2.

The calculation of the thrust of nozzle propellers is not dealt with in the text of the current regulations. Please refer to Appendix 2 for guidelines on the calculation of the propeller thrust for nozzle propellers and open propellers.

The thrust of the propellers can also be determined at full scale by bollard pull tests. Please refer to Appendix 3 for guidelines on bollard pull tests for determining the thrust of the propeller(s). A summary is presented in the table below.

Table 1
Summary of Appendices 2 & 3. "Actual value" means the value obtained from tests or calculations, as applicable

Thrust > F_{actor} * R_{CH}

	Calculation of Thrust	Bollard Pull Test	Bollard Tow Test 1	Bollard Tow Test 2
Speed (knots)	5	0	5	5
R_{CH}	rule formula	rule formula	rule formula	rule formula
R_{OW}	Table 2 or actual value*)	Table 2 or actual value	-	-

	Calculation of Thrust	Bollard Pull Test	Bollard Tow Test 1	Bollard Tow Test 2
Thrust reduction factor	0.15 or actual value	-	-	-
J-factor	-	0.20 or actual value	-	-
Measuring accuracy	-	±2 %	±2 %	Winch
Factor	Table 2	1.05 times Table 2 values	1.00	1.10

**) The actual value can be calculated using the frictional resistance coefficient* $C_f = 0.075\cdot[lg(Re/100)]^{-2}$ *increased by 50 % to account for the residual resistance coefficient; Re is the Reynolds numer.*

Table 2
Factors referred to in Table 1

Ice class	Direct shaft	Azimuthing thruster
L1A	1.28	1.25
L1	1.32	1.29
L2	1.37	1.34
IC	1.45	1.42

7.4.2 Other Methods of Determining R_{CH}

The resistance of the vessel in a brash ice channel can be determined by model testing in an ice tank. For guidelines on ice model testing and model test reporting, please refer to Appendix 4.

8 HULL STRUCTURAL DESIGN (CHAPTER 4 OF THE REGULATIONS – SECTION I)

8.1 Frame Connections

Frame connections must transfer the loads stemming from the secondary members to primary structural members in the structural hierarchy. The maximum load transferred by the stringers in the transverse framing system to the web frames is as follows (where p is the rule ice pressure and h the ice load height)

$$F = 1.8\, p\cdot h\cdot l$$

where l is the web frame spacing S in the transverse framing system; and by transverse or longitudinal frames to stringers (or deck strips) or web frames, respectively,

$$F = p\cdot h\cdot l$$

where l is the frame spacing s in a transverse framing system or longitudinal frame span l in a longitudinal framing system. These equations are based on the ice loads the frames are assumed to carry and on the fact that a safety factor of 1.8 is assumed in the stringer design. The connection must be designed with a capacity sufficient to carry at least this load, without exceeding the yield or buckling capacity of the structure. Particular attention should be paid to adequate stiffening of the connection between deep web frames and longitudinal stiffeners with large spacing. Examples of the appropriate connections are given in Figures 3a and 3b. The frame connection for a web frame, stringer, deck or deck strip should use a lug as shown in Fig. 3c. In higher ice classes, the distance of the lug from the shell plating (d in the Fig. 3c) should be zero i.e. the lug should also be attached to the shell. In such a case, the requirements of the PRS should be followed.

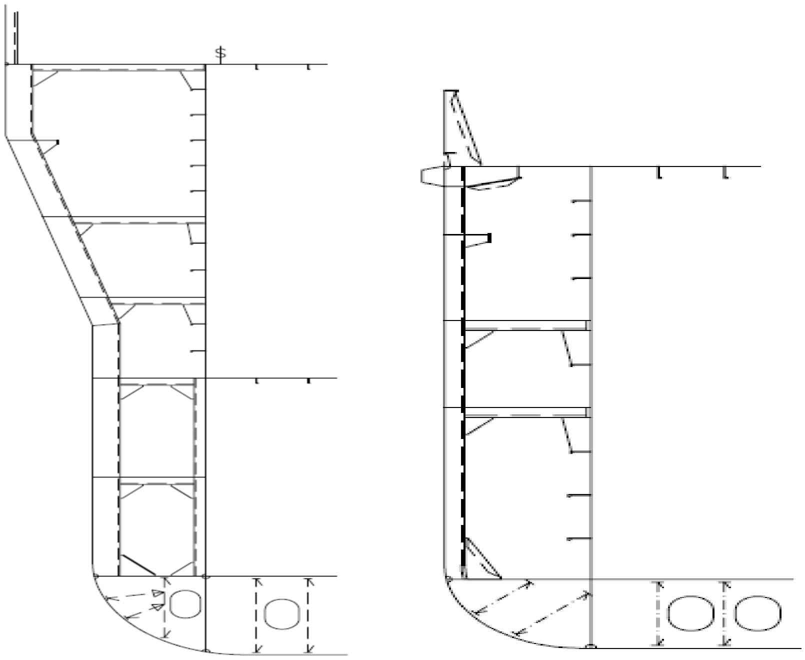
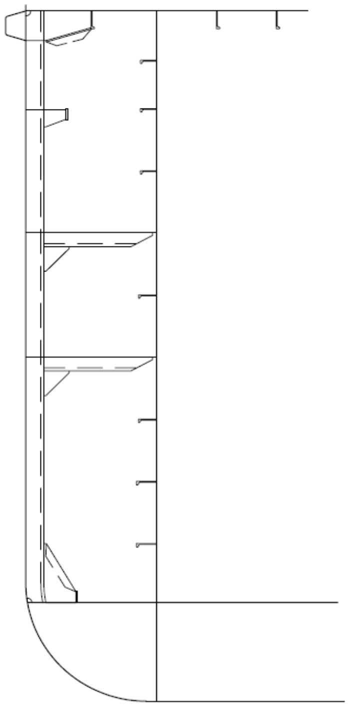
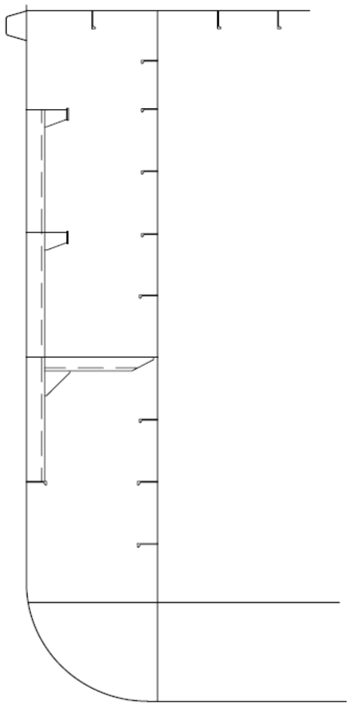


Figure 3a.
Examples of frame connections





ICE FRAME ICE FRAME

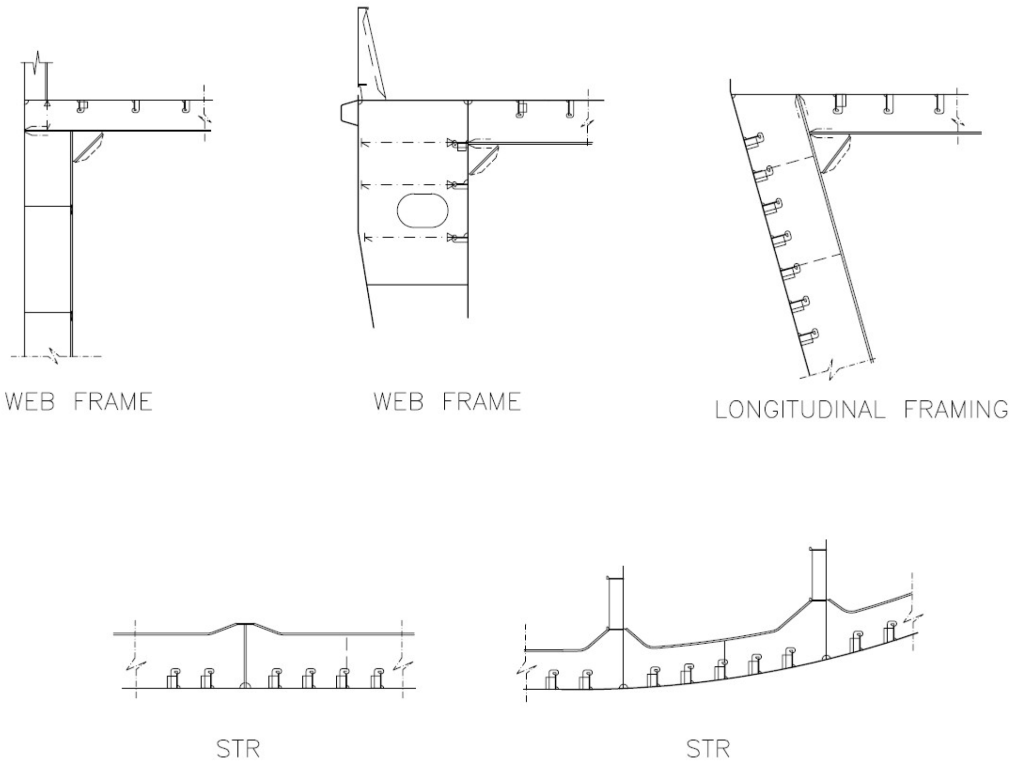


Figure 3b.
Examples of frame connections

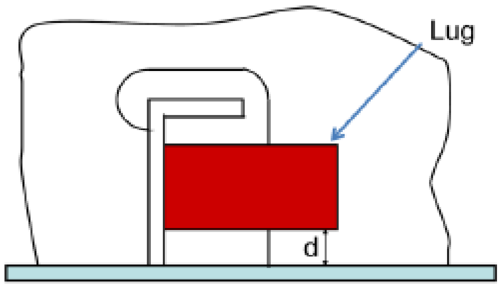


Fig. 3c.
Construction of a frame connection using a lug

8.2 Vertical Extension of Ice Strengthening of Framing (Section 4.4.1 of the Regulations)

It is assumed that only the ice belt area (Area 1 in Figure 4), as defined in paragraph 4.3.1, will be directly exposed to ice contact and pressure. For this reason, the vertical extension of the ice strengthening of the longitudinal frames should be extended up to and including the next frame up from the upper edge of the ice belt (frame 3 in Figure 4). Additionally, the frame spacing of the longitudinal frames just above and below the edge of the ice belt should be the same as the frame spacing in the ice belt (the spacing between frames 2 and 3 should be the same as between frames 1 and 2 in Figure 4). If, however, the first frame in the area above the ice belt (frame 3 in area 2 in Figure 4) is closer than about $s/2$ to the edge of the ice belt, the same frame spacing as in the ice belt should be used above the edge of the ice belt i.e. in the spacing between frames 3 and 4 (where s is the frame spacing in the ice belt).

8.3 Inclined or Unsymmetrical Frame Profiles (Section 4.4.4.2 in the Regulations)

Section 4.4.4.2 refers to the need for supporting frames that are 'not normal' or 'unsymmetrical' against tripping, but defines neither 'normal' nor 'symmetric'. For design purposes, the frame inclination, and the combined effects of the frame inclination and asymmetry on the principal axis of the frame, must be separately evaluated. Accordingly, if either the angle of the frame inclination or the principal axis of the frame (without attached plating) deviates more than 15° from normal to the plating, support against tripping is required. Please note that if the PRS has its own standard for these limits, this must be followed.

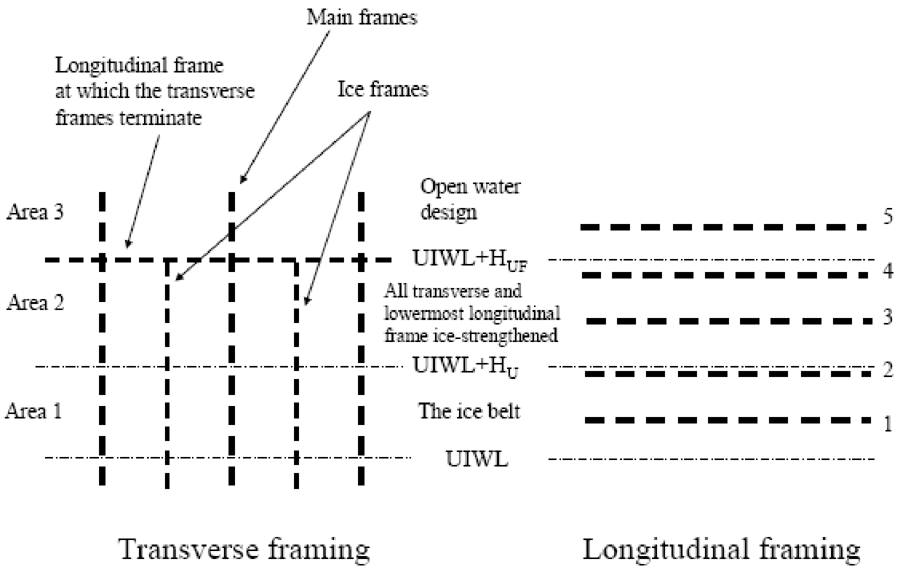


Figure 4.
The different ice strengthening areas at and above the UIWL. The distances H_U and H_{UF} are specified in tables in 4.3.1 and 4.4.1, respectively (columns 'above UIWL'); these distances vary depending on the ice class.

8.4 Section Modulus and Shear Area

Instead of using the formulae given in section 4.6.2 of the regulations for the section modulus and shear area of web frames, a direct stress calculation may be performed to determine these.

In each case, the point of application of the concentrated load should be chosen in relation to the arrangement of stringers and longitudinal frames so as to obtain the maximum shear and bending moments. The allowable stresses are as follows:

Shear stress: $\tau = \sigma_y/\sqrt{3}$
Bending stress: $\sigma_b = \sigma_y$

Equivalent stress: $\sigma_c = \sqrt{\sigma_b^2 + 3\tau^2} = \sigma_y$

8.5 Arrangements for Towing

The towing method normally used in the Baltic by icebreakers is notch towing. Notch towing is often the most efficient way of assisting ships of moderate size (with a displacement not exceeding 30.000 tons) in ice. If the bulb or ice knife makes a ship unsuitable for notch towing, in heavy ice conditions this kind of ship may have to wait for the ice compression to diminish before the ship can be escorted without notch towing. During towage, the towed vessel acts like a large rudder for the icebreaker and this causes difficulties, particularly if the merchant vessel is loaded or the bow does not fit well with the notch.

The towing arrangement usually involves a thick wire which is split into two slightly thinner wires, shown in Figure 5. Two fairleads must be fitted symmetrically off the centreline with one bollard each. The distance of the bollards from the centreline is approximately 3m. The bollards must be aligned with the fairleads, allowing the towlines to be fastened straight onto them. A typical towing arrangement is shown in Figure 5. The additional installation of a centreline fairlead is recommended, since this is still useful for many open water operations and some operations in ice.

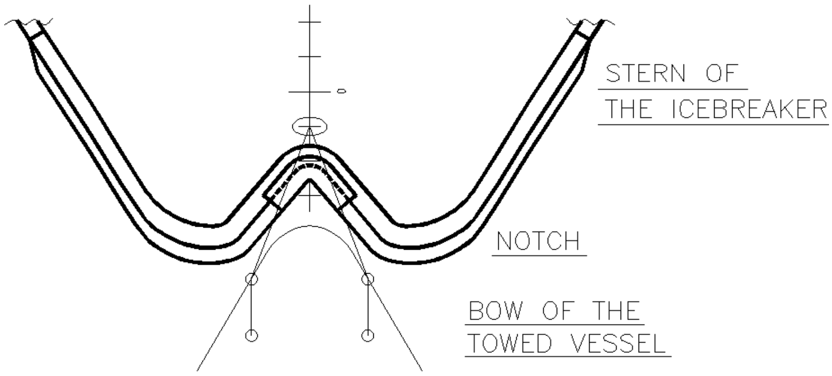


Figure 5.
The typical towing arrangement.

A bollard or other means for securing a towline, structurally designed to withstand the breaking force of the towline of the ship, must also be fitted. Operational experience indicates that the bollards can never be too strong and should be properly integrated into the steel structure. As a guideline for bollard design, it should be required that they withstand at least the maximum icebreaker winch force, which is usually 100 – 150 t. The maximum possible force on the bollards is given by the breaking load of the most commonly used cable, a 62 mm cable. This has a breaking load of about 200 t.

The ship bow should be suitable for notch towing. Such suitability involves the proper shape of the bow waterline at the height of the icebreaker notch. This height is around 2.5 m. If the bow shape is too blunt, it will not fit well into the icebreaker notch. For guidance, the notch shape of IB Otso and Kontio, together with the notch of MSV Botnica, are presented below in Figure 6.

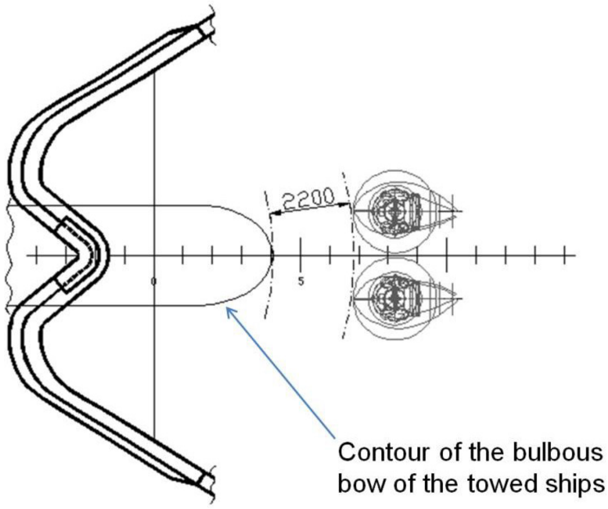


Figure 6a.
A sketch of the notch of IB Otso and Kontio, also showing the bulbous bow of the towed ship.

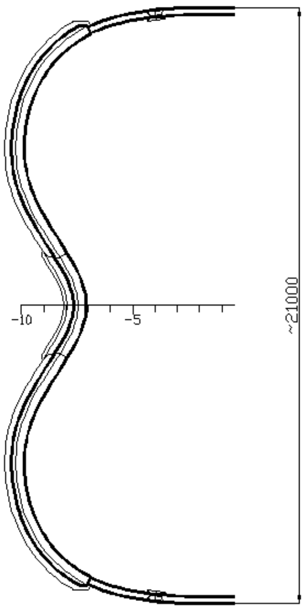


Figure 6b.
The notch of MSV Botnica.

Ships with a bulb protruding forward of the forward perpendicular are, however, difficult to tow in a notch. If the towed ship has a bulb, suitability for notch towing also depends on the profile of the bow. The owners should check the appropriate Finnish or Swedish guidelines for winter navigation, to see whether the bulb in question allows notch towing. The bulb will not fit into the notch if the bow is too high, (see Figure 7). If the bow is too high in ballast condition, the ship could be trimmed to lower the bow. When the ship is loaded, the bulb will be low and may then make contact with the icebreaker propellers or rudders. It is recommended that the bulb should not extend more than 2.5 m forward of the forward perpendicular, (see Figure 7). This recommendation should be checked alongside the details of icebreakers operating in the operational area. For guidance, the stern profiles of two icebreakers are presented in Figure 8.

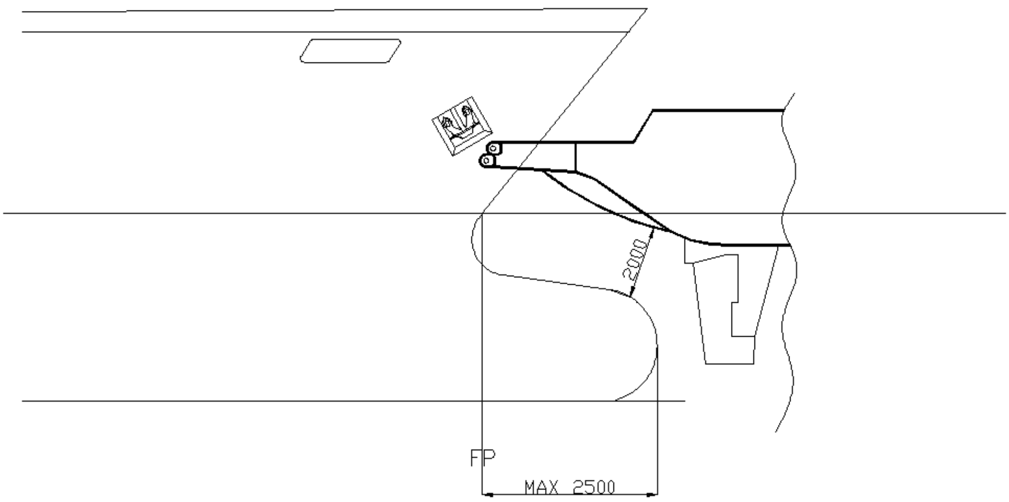


Figure 7a.
The extension of the bulb forward of the forward perpendicular with a suitable loading condition.

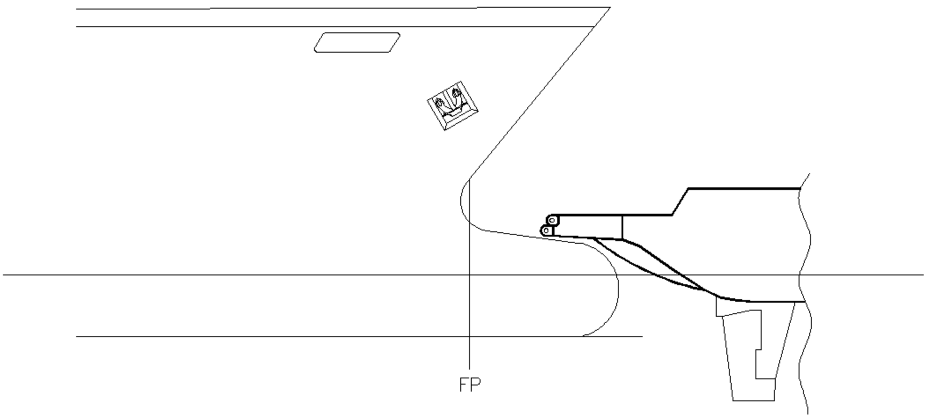


Figure 7b.
Problems arising when towing a vessel in ballast with an unsuitable loading condition.

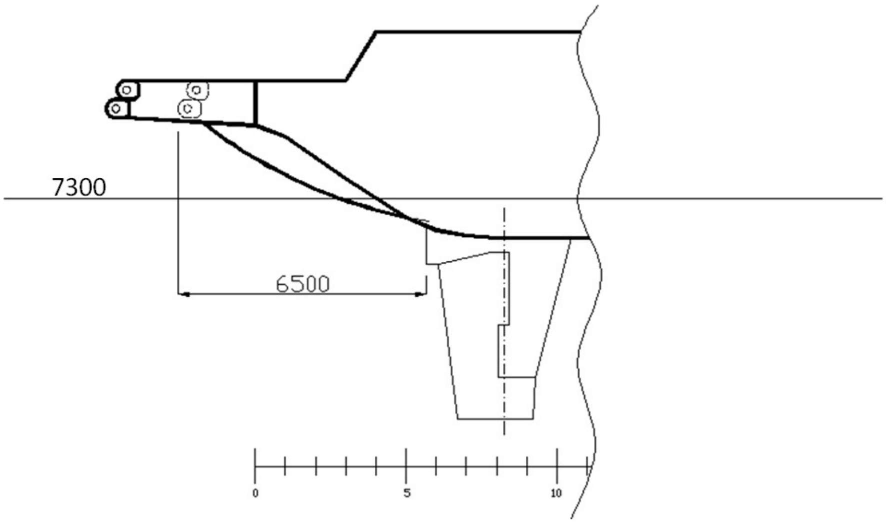


Figure 8a.
A sketch of the stern profile of IB Otso and Kontio.

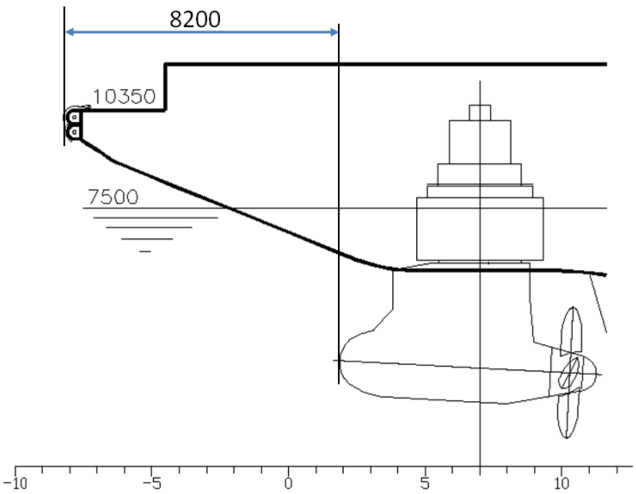


Figure 8b.
The stern profile of MSV Botnica.

Some merchant ships have an ice knife fitted above the bulb, (see Figure 9). This ice knife is just a vertical plate which presents a sharp edge against the notch at certain draughts. As these ice knives destroy the fendering at the icebreaker notch, their use is discouraged if efficient icebreaker assistance is to be provided.

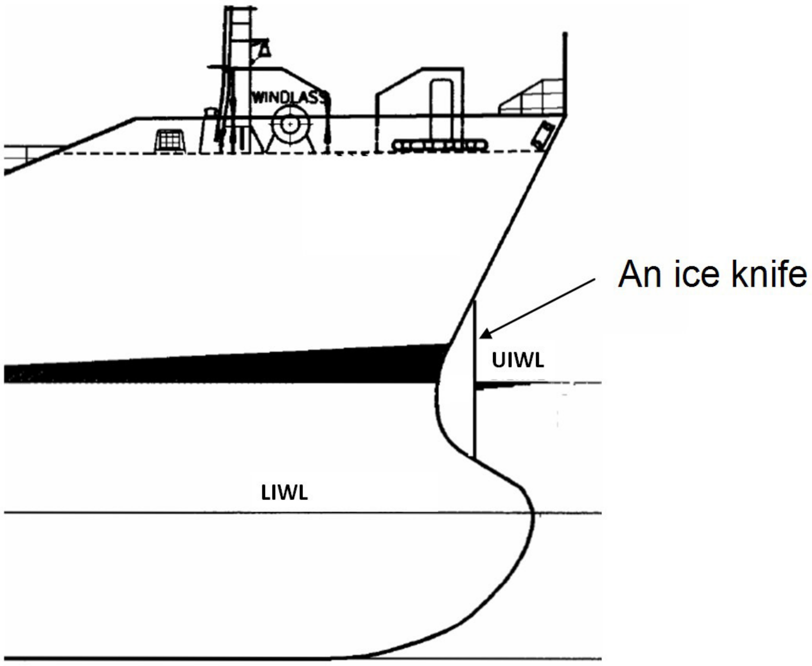


Figure 9.
An ice knife is fitted on some ships.

8.6 Reduction in Corrosion Allowance (Section 4.3.2 of the Regulations)

The corrosion allowance should be set at 2mm. A 1mm reduction in corrosion allowance can be considered if a recognised abrasion resistant coating is applied. Recognition of an abrasion resistant ice coating is generally based on satisfactory service experience and laboratory tests. As the actual performance of a coating cannot be accurately assessed in laboratory, service experience is particularly important to the assessment of such products. Manufacturers should therefore submit sufficient dry docking reports of ships to which the coating has previously been applied and which have operated in ice conditions, in addition to laboratory test results. The laboratory tests should be carried out using a recognised coating system as a reference.

The surface preparation and coating application are as important as selecting the correct coating and should strictly follow the manufacturer's instructions. In general, the steel surfaces should be abrasive blasted to Sa2½ (ISO 8501-1) or Sa3, with a surface roughness of at least 75μm. If repair painting is applied, similar requirements must be followed – and the old coating should be roughened and the salinity (chloride contamination level) of the surfaces should be checked and must be less than 5μg/cm^2.

When considering laboratory testing, the following testing procedure could be followed:
– Resistance to abrasion (Taber abraser test)
– Impact resistance
– Adhesion strength
– Extensibility (flexibility) e.g. according to ASTM D4145
– In addition, the following corrosion tests could be considered:
– Cyclic corrosion test or salt spray test

– Water immersion test
– Cathodic disbondment test.

The test results should be compared with those from a product already recognised by the PRS. A measure of an abrasion resistance is given by the Taber abrasion test (ASTM D4060), where the rate of abrasion was 160mg/1,000 rounds, using a 1kg weight and CS17 disks.

The acceptance of a 1mm corrosion allowance is subject to adequate documentation submitted to Finnish or Swedish authorities or classification societies.

8.7 Propeller Clearance

An extremely narrow clearance between the propeller blade tip and the stern frame or the bottom of the level ice sheet should be avoided, as a small clearance will place very high loads on the propeller blade tip. In the first case, the loads are caused by ice floes being forced between the stern frame and the propeller and, in the second case, in situations where there is a risk that the propeller will hit large floes, especially when going astern. The stern frame clearance should be at least 0.5 m and the ice clearance should be positive when the level ice thickness is taken as stated in the regulations, (see the table in section 4.2.1.).

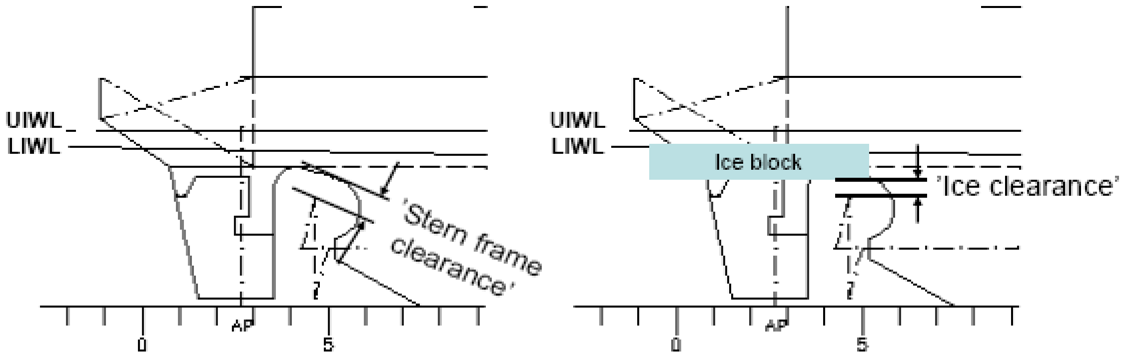


Figure 10.
The clearance between the stern frame and the propeller (left) and the ice sheet and the propeller when the ship is at LIWL (right).

8.8 Transom STERN

A wide transom stern extending below the UIWL will seriously impede the capability of the ship to go astern in ice, because the ice will be crushed against the transom. The capability to go astern in ice is most important for higher ice class ships. For this reason, a transom stern should not be extended below the UIWL if this can be avoided. If it is unavoidable, the part of the transom below the UIWL should be kept as narrow as possible, in order to limit the area of the stern against which ice is crushed. The part of the transom stern situated within the ice belt should be strengthened at least to the level of the midship region, because the loading of the midbody mainly arises due to crushing, as the side at the midbody tends to be vertical.

8.9 BILGE KEELS

Bilge keels are often damaged or ripped off in ice, (see Figure 11). The reason for this is that ice floes roughly follow the buttock lines when the ship is proceeding in ice. The connection of bilge keels to the hull should be designed to minimise the risk of damage to the hull if the bilge keel is damaged. A construction often described as an 'A-type' bilge keels is recommended due to its strength. An example of this kind of construction is shown in Figure 12. To limit the damage which occurs when a bilge keel is partly damaged, it is recommended that bilge keels are split into several, shorter independent lengths. The forward and aft parts of the bilge keels should also be pointed towards the oncoming ice when going forward or astern, respectively.

Figure 11.
Damage caused by ice on the bilge keel of a ship. Please note that this is an example of damage caused by ice, not an example of good or bad design.

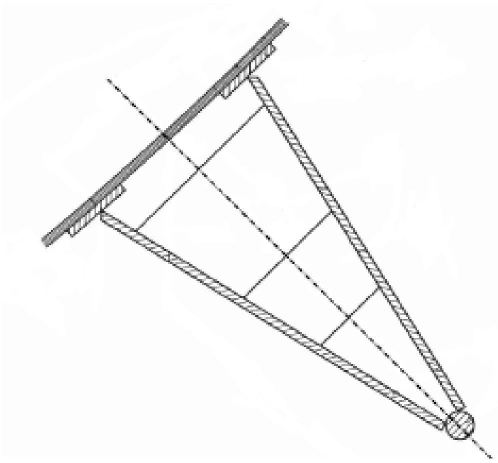


Figure 12.
An example of an A-type bilge keel construction.

9 RUDDER AND STEERING ARRANGEMENTS (CHAPTER 5 OF THE REGULATIONS – SECTION I)

9.1 Ice Knife

When going astern, level ice will be broken by the stern and the ice floes will be forced under the ship. The function of the ice knife is to push ice floes approaching the rudder downwards, so that the rudder is not subject to head-on impacts with ice floes and large forces that deviate the rudder out the amidships position occur less frequently. Attention should be paid to the strength and shape of the ice knife with regard to its function. A properly shaped ice knife is shown in Figure 13: the lowest part of the ice knife should be below water for all draughts. However, if it is not intended that the ship will go astern in ice at some draughts, a smaller ice knife could be used. An ice knife is recommended for all ships with an ice class of **L1A** or **L1**.

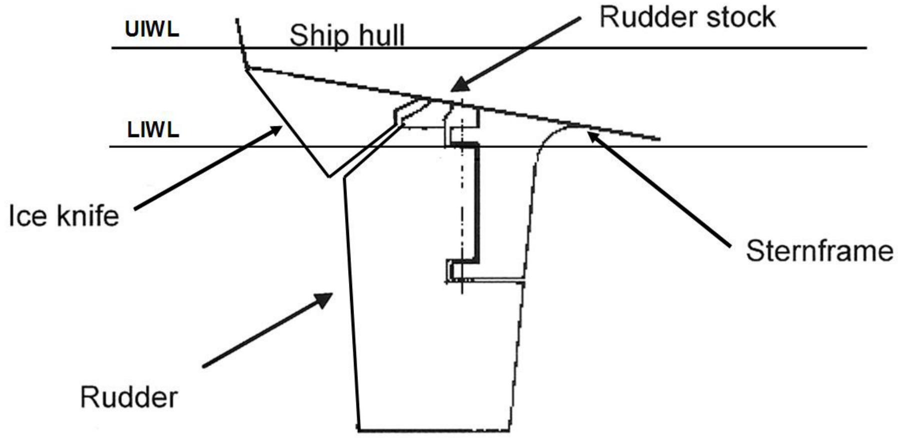


Figure 13.
An example of an adequate ice knife design.

If the vessel has a flap-type rudder, special attention should be paid to the design of the rudder in combination with the ice knife, as the flap mechanism is more vulnerable to ice forces.

9.2 Rudder Turning Mechanism

When going astern, a large turning moment will be applied to the rudder, especially if it is allowed to deviate from the amidships position. In order to avoid a situation where the rudder is forced sideways, the operators should pay attention to keeping the rudder amidships when going astern. At the same time, rudder stoppers should be installed in order to avoid excessive movement of the rudder(s).

When the rudder is turned sideways, a great deal of pressure will act on the rudder turning mechanism. The relief valves for hydraulic pressure in the turning mechanism must therefore be effective. The components of the steering gear should be dimensioned to withstand loads corresponding to the required diameter of the rudder stock.

9.3 Bow Thrusters

In general, bow thrusters are not used in ice, because ice floes can damage the thruster blades. Of course, thrusters can be specifically designed for ice loading, as some manufacturers have done. Ice floes can become jammed in the tunnel entrance, making operation of the thrusters impossible. Sometimes, a grid is recommended at the tunnel entrance in order to prevent ice floes from entering the tunnel. On the other hand, this can diminish the thruster’s performance when used in open water. Some classification societies may have their own recommendations for the grillage design of bow thrusters and note should be taken of these.

10 PROPULSION MACHINERY (CHAPTER 6 OF THE REGULATIONS)

10.1 Time domain calculation of torsional response (section 6.5.3.4.1 of the PRS Regulations)

Current PRS Regulations (2017) state that when calculating the torsional excitation for propeller shaft, the ice torque increases to maximum torque value during a 360 degree period. This same 360 degree ramp is also used to decrease the excitation back to zero level. In earlier rules this ramp was 270 degrees.

For lowest ice class, L3, the full ice milling sequence is 720 degrees. With 360 degree ramp this means that this sequence consist of only ramp to increase the ice torque (0 -> 360) and a ramp to decrease it (360 -> 720). As the ice torque is modelled as sinusoidal, it follows that the maximum intended ice torque value is not always achieved.

Depending on the number of propeller blades and the excitation case, this leads to 4-14 % lower value of the maximum torque than intended. One possible remedy to this situation is using 270-degree ramp for ice class L3. The FSICR gives the minimum level for design to comply with the requirements of a Finnish-Swedish ice class and Classification Societies can have stricter requirements. For meeting the requirements of Finnish-Swedish ice classes, adherence to the text of FSICR is however sufficient.

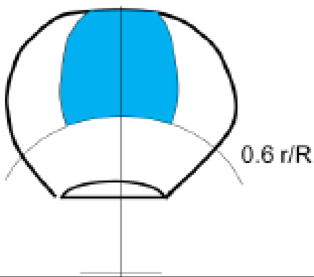
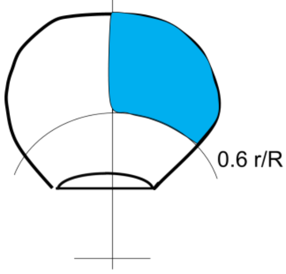
10.2 Calculation of Blade Failure Load and the Related Spindle Torque with Elastic-Plastic Fem (Section 6.5.4.1 and 6.5.4.2 of the PRS Regulations)

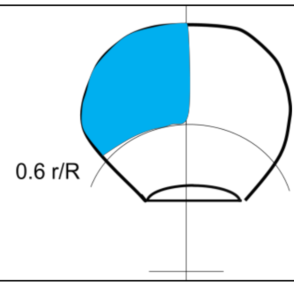
The ultimate load resulting from blade failure, as a result of plastic bending around the blade root, is given in the ice class regulations. As an alternative to the simple equation methodology, an elastic plastic FEM can be used to calculate the blade failure load and blade failure spindle torque.

The blade plastic failure should be calculated using load case FEX1. The pressure acts perpendicular to the blade surface on the bent blade. The pressure should be increased gradually until the shaft bending moment begins to drop. The maximum value of the shaft bending moment is then regarded as the design shaft bending moment.

The blade plastic failure should be calculated using load cases FEX2 and FEX3, with elastic-plastic FEM. The ice pressure acts perpendicular to the blade surface on the bent blade. The pressure should be increased gradually until the blade spindle torque starts to drop. The maximum value of the shaft spindle torques for load cases FEX2 and FEX3 is then regarded as the design spindle torque.

Table 3
Load cases for elastic plastic FEM analysis of blade failure load

Load case	Description	Location of contact pressure
Load case Fex1	Constant pressure in the area above 0.6 r/R. Chord wise until 50% towards leading edge and 50% towards trailing edge.	
Load case Fex2	Constant pressure in the area above 0.6 r/R but only on the leading edge half of the propeller blade.	

Load case	Description	Location of contact pressure
Load case Fex3	Constant pressure in the area above 0.6 r/R but only on the trailing edge half of the propeller blade.	

10.3 Safety Valves Fitted to Azimuthing Main Propulsor Units (Sections 6.6.5.1 and 6.6.5.2 of the PRS Regulations)

The dimensioning steering torque of azimuthing main propulsors may be limited according to the rules of the classification society to take into account the effect of torque limiting devices such as mechanical torque limiters or hydraulic safety valves in the steering mechanism.

10.4 Acceptability criterion for static loads (section 6.6.5.4 of the PRS Regulations)

The safety factor of 1.3 is not meant to be applied to the detail design of bolted connections, for example the pre-stresses in bolts. Rather, classification societies' rules should be applied for these. The loads calculated in sections 6.6.5.2 and 6.6.5.3 are multiplied by the safety factor and then applied to the thruster body. With these loads the components must be able to maintain operability without incurring damage that requires repair.

10.5 Extreme Ice Impact Loads on Azimuthing Main Propulsors (Section 6.6.5.2 of the PRS Regulations)

The extreme ice impact load calculation given in the regulations is based on a moderately sized ice block and common ship operational speed. The same load level can also occur in the case of large ice blocks and lower speeds. For reference in relation to an ice block of this size, the propeller load reference ice thickness H_{ice} can be used as follows, to correspond to the load levels given in the regulations.

Table 0-1
Parameter values for ice dimensions and dynamic magnification

	L1A	L1	L2	L3
Thickness of the design ice block impacting thruster (2/3 of H_{ice})	1.17 m	1.0 m	0.8 m	0.67 m
Extreme ice block mass (m_{ice})	8670kg	5460 kg	2800 kg	1600 kg
C_{DMI} (if not known)	1.3	1.2	1.1	1

Table 0-2
Impact speeds for aft centerline thruster

Aft centreline thruster				
Longitudinalimpactinmain operational direction	6 m/s	5 m/s	5 m/s	5 m/s
Longitudinal impact in reversing direction (pushing unit propeller hub or pulling unit cover end cap impact)	4 m/s	3 m/s	3 m/s	3 m/s
Transversalimpactbowfirst operation	3 m/s	2 m/s	2 m/s	2 m/s
Transversalimpactsternfirst operation (double acting ship)	4 m/s	3 m/s	3 m/s	3 m/s

Table 0-3
Impact speeds for aft wing, bow centreline and bow wing thrusters

Aft wing, bow centreline and bow wing thruster				
Longitudinalimpactinmain operational direction	6 m/s	5 m/s	5 m/s	5 m/s
Longitudinal impact in reversing direction (pushing unit propeller hub or pulling unit cover end cap impact)	4 m/s	3 m/s	3 m/s	3 m/s
Transversal impact	4 m/s	3 m/s	3 m/s	m/s

10.6 Extreme ice ridge loads on azimuthing main propulsors (section 6.6.5.3 of the PRS Regulations)

The thickness of the consolidated layer of the design ridges are given as information only in Table 6-21 and Table 6-22. The thickness of the consolidated ridge has been taken into account in the formulation of formula 6.48 but it is not used as an input parameter in the calculations.

10.7 Fatigue Design and Acceptability Criterion for Fatigue for Azimuthing Main Propulsors

The thruster body and other components are subject to alternating loads. The load distribution should be estimated in order to evaluate the fatigue strength. If there is no proven method of estimating the load distribution, a Weibull type load distribution with a shape factor of 1 can be used to estimate the fatigue load distribution. The maximum load for the distribution used should be the extreme load given in sections 6.6.5.2 and 6.6.5.3.

The number of ice loads is as follows:

$$N_t = Z \cdot N_{ice} \cdot C_{nice} \quad [\,] \tag{01}$$

where:

C_{nice} is a factor taking account of the thruster body loads on top of the propeller induced loads

If C_{nice} is unknown, a value of 1.2 should be used.

The Miner damage sum should be below 1, with a safety margin of 1.5 in the load level.

10.8 Simplified Methodology for Estimation of Once in a Lifetime Vibratory Loads on the Azimuthing Thruster (Section 6.6.5.5 of the PRS Regulations)

Experience of full-scale performance shows that the blade order vibration of the thruster global modes causes significant vibration if resonance occurs at the rotational speed when using the propulsion line at high power. In such circumstances, there is high risk of damage to the bearings, the thruster gear and in the structures. A simplified methodology is given below for evaluating vibration amplitude during resonance. The response can then be used to evaluate the risk of damage. Resonant vibration should be avoided at high power revolutions.

The methodology consists of the following steps.

Estimation of the global natural frequencies of the thrusters in the longitudinal and transverse direction and the propeller revolutions at blade order resonances.

Estimation of propeller blade order excitation due to propeller-ice interaction.

Estimation of the vibratory response based on the estimated dynamic magnification factor.

10.9 Estimation of the natural frequencies and modes

An azimuthing thruster tends to have lateral and longitudinal vibration modes as described in Figure 14.

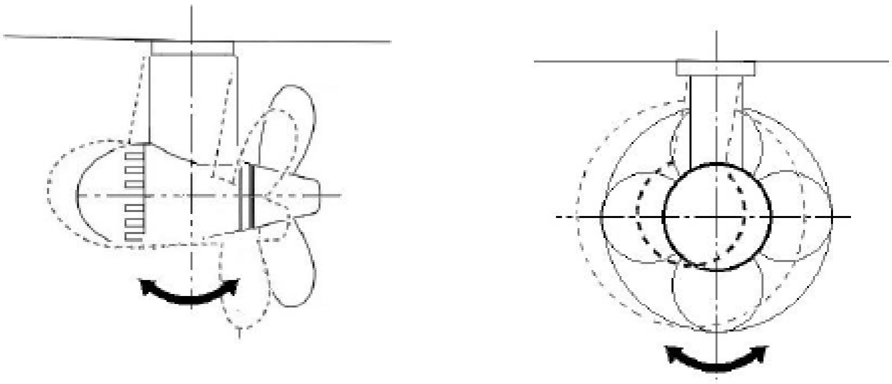


Figure 14.
Schematic figure of typical longitudinal (left) and transversal (right) natural modes of the thruster.

In the estimation of thruster global natural frequencies in the longitudinal and transverse direction, account should be taken of the damping and added mass due to water. In addition, the effect of the ship hull attachment stiffness should be modelled.

10.10 Estimation of blade order excitation on propeller due to ice

The dynamic excitation loads are derived from the propeller ice loads presented in the ice class regulations. The propeller-induced load sequences are estimated to be a continuous series of half sine shape impacts. The blade order component of that series is applied as sinusoidal excitation of the propeller. The excitations are illustrated in Figure 15 and the formulae used for defining the excitation amplitude are given below.

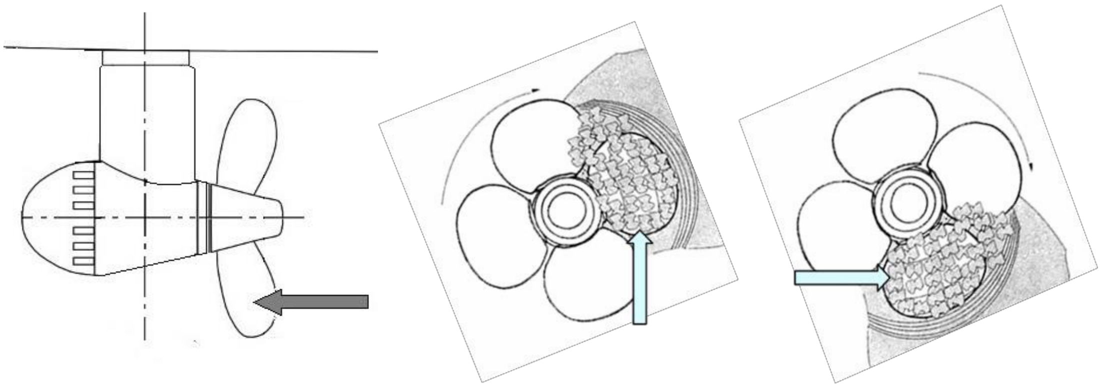


Figure 15.
Location of the excitation on the blade.

Axial blade order excitation amplitude $F_{bla} = F_{\mathrm{b}}$ or F_{f} whichever is greater
Vertical blade order excitation amplitude $F_{blv} = 0.75 * Q^*_{\mathrm{max}} / (0.7 * R)$
Lateral blade order excitation amplitude $F_{blh} = 0.75 * Q^*_{\mathrm{max}} / (0.7 * R)$

where:

F_{b} and F_{f} are the maximum backward and forward blade forces calculated using the ice class rule equations.

Q^*_{max} is the maximum ice torque calculated at the relevant rotational speed, using the ice class rule equation.

R is the propeller radius.

If the thrusters have blade order vibratory resonance at the operational revolution range, the extreme loads at the resonance can be estimated using FEM, or with the simplified method described below.

The sinusoidal extreme excitation is for any direction:

$$F_{bl}(\varphi)=F_{bl}\cdot C_{q1}\cdot\sin(Z*\varphi+\alpha_1)\ [\mathrm{kN}]$$

where:

F_{bl} is F_{bla}, F_{blv} or F_{blh} depending on the direction of the vibration

C_{q1} is first blade order Fourier component

φ is the angle of rotation

α_1 is the first order phase angle of excitation component

Z is the number of blades

Table 4
Blade order Fourier coefficients for sinusoidal excitation of the propeller

	C_{q1}	α_1
Z = 3	0.375	-90
Z > 3	0.36	-90

10.11 Estimation of the Response at Resonance

In the case of continuous sinusoidal excitation, the dynamic magnification factor is dependent on the damping. Damping during thruster vibration is not very well known when propeller/ice interaction occurs. However, estimates for open water condition may be used.

The response force at a resonance taking account of dynamic magnification for transverse, longitudinal and vertical vibration is then.

$$F_{bl\,resp}(\varphi)=C_{DM}\cdot F_{bl}(\varphi)$$

where:

F_{bl} is the excitation force defined earlier

C_{DM} is the dynamic magnification factor for thruster body vibration

The dynamic magnification factor C_{DM} may have different values for vibration in different directions. The typical value for open water is 10 – 20.

10.12 Application Point of the Response Force

For the strength evaluation of the thruster, the response force should be placed at the location of the thruster vibratory mass, typically at propeller shaft level, see Figure 16.

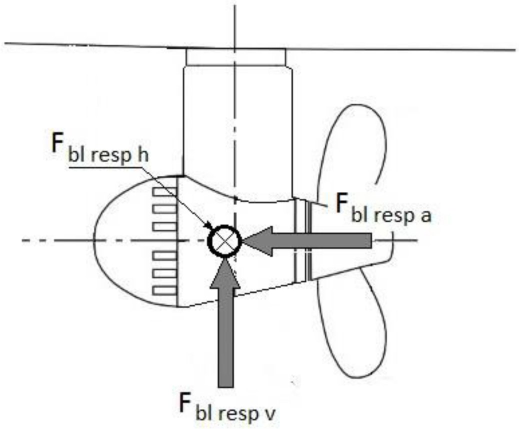


Figure 16.
Schematic figure showing the location of the response force for longitudinal, transverse and vertical vibration.

10.13 Local Ice Pressure on Azimuthing Thruster

The local strength of the thruster body tends to be sufficient when the thruster is designed for global loads. However, local ice pressure design can follow the principles for rudder design presented in chapter 5 of the ice class regulations. It is assumed that the thruster will encounter bow region pressures, since thrusters are usually strongly exposed to ice contact.

11 MISCELLANEOUS MACHINERY REQUIREMENTS (CHAPTER 7 OF THE REGULATIONS)

11.1 Sea Inlet and Cooling Water Systems

The principle behind section 7.2 of PRS Regulations involves ensuring the safe operation of machinery in ice conditions. According to item 4 "*a pipe for discharge cooling water, allowing full capacity discharge, shall be connected to the chest".* This promotes the melting of ice pieces and slush that may have entered the sea chest. Reference is also made to IMO MSC/Circ. 504 "*Guidance of Design and Construction of Sea Inlets under Slush Ice Conditions"*.

If the vessel is designed to operate in southern latitudes, with a very high cooling system capacity due to high sea water temperatures, it may be appropriate to design the capacity of the cooling water re-circulating line in accordance with the actual required cooling water capacity of the machinery in ice conditions, when the sea water temperature is much lower. The amount of water entering the sea chest through the recirculation line should enable the full capacity discharge of sea water required for the cooling of the machinery when sailing in ice.

Box coolers have functioned well in ice and are thus also an acceptable technical solution for ensuring the supply of cooling water when navigating in ice.

12 GENERAL SUITABILITY FOR WINTER CONDITIONS

When designing a ship for winter navigation in the Northern Baltic, account should also be taken of certain issues other than those mentioned in the regulations. The low ambient temperature should be borne in mind in particular.

12.1 Low Ambient Temperature

In the northern Baltic Sea area, the air temperature is below 0°C for much of the winter and may occasionally fall to around -30°C, and for short periods of time temperatures as low as - 40°C can be encountered. This should be taken into account when designing structures, equipment and arrangements essential to the safety and operation of the ship. Matters to be borne in mind include e.g. the functioning of hydraulic systems, the danger of water piping and tanks freezing, the start-up of emergency diesel engines, the strength of materials at low temperature, etc.

The following temperatures are given for reference in the Baltic Sea area:
– Ambient temperature: -30°C
– Sea water temperature: -2°C

Equipment and material exposed to the weather should be capable of withstanding and remaining operable at the design temperature for long periods. (Note: There have been no reported cases of brittle fracturing when material grades designed for normal worldwide service are used for winter navigation in Baltic Sea Areas). The propulsion and auxiliary machinery should be capable of full operation in ambient conditions, as required in winter conditions. For example, the engine suction air should be sufficiently heated before entering the engine, or other alternative solutions, such as a specially adapted waste-gate, should be considered.

Appendix 1

INSTRUCTIONS FOR THE APPLICATION OF A LETTER OF COMPLIANCE

If the required engine power of the vessel has been determined by model tests, a letter of compliance issued by the PRS is required. Such a letter of compliance should be drawn up for the individual ship in question. For this purpose, the following information should be forwarded to the Administration for each individual ship:

- The name of the vessel, if known
- The call sign of the vessel, if known
- The IMO number of the vessel
- The main dimensions of the vessel
- A copy of the final lines drawing of the vessel
- The main engine type and the total engine output the propulsion machinery can continuously deliver to the propeller(s) of the vessel
- Reference to the model test report
- The resistance of the vessel and available net thrust of the propulsion machinery in a brash ice channel, as defined in section 3.2.5 of the PRS Ice Class Regulations, 2017.

Appendix 2

GUIDELINES FOR THE CALCULATION OF PROPELLER THRUST FOR OPEN AND NOZZLE PROPELLERS

It has been suggested that an alternative power requirement for nozzled propellers should be accepted instead of the one given in PRS ice class regulations, based on a better propeller thrust than for an average propeller. Naturally, the vessel must fulfil the basic requirement of 5 knots in a specified brash ice channel (the thickness of which varies with the ice class), but the power used to produce the thrust must be optimised. Two more direct ways of calculating or determining thrust are examined below – using propellers in nozzles and the direct determination of propeller thrust.

The basic assumption of the regulations is that the bollard pull T_B of the vessel can be determined as:

$$T_B = K_g(P \cdot D_p)^{2/3} \tag{1}$$

where K_g is the efficiency factor of bollard pull, being 0.78 for a single CPP and 0.98 for a twin CPP, P is the ship power and D_P is the propeller diameter. As the requirement in the regulations is a speed of 5 knots, the concept of net thrust is used in the following calculations. The net thrust T_{NET} takes account of the open water resistance R_{OW} and the change in propeller thrust T at speed v_1 (the K_T curve decreases as J i.e., the speed, increases). The force balance in ice at speed v_1 is (R_{CH} is the rule channel resistance):

$$(1 - t_1)T(v_1) = R_{OW}(v_1) + R_{CH}(v_1) \tag{2}$$

where t_1 is the thrust deduction factor at speed v_1. This basic equation gives the definition of net thrust as:

$$T_{NET} = (1 - t_1)T(v_1) - R_{OW}(v_1) \tag{3}$$

This definition can be expressed with the bollard pull value and the propulsion coefficients, assuming that the propeller absorbs full power at both velocity points as:

$$T_{NET} = \frac{1-t_1}{1-t_0} \cdot \frac{K_T(J_1)}{K_T(0)} \cdot \left(\frac{n_1}{n_0}\right)^2 T_B - R_{OW}(v_1) = \frac{1-t_1}{1-t_0} \cdot \frac{K_T(J_1)}{K_T(0)} \cdot \left[\frac{K_Q(0)}{K_Q(v_1)}\right]^{\frac{2}{3}} \cdot T_B - R_{OW}(v_1) \tag{4}$$

where J_1 is the advance coefficient and n_1 the RPM at speed v_1, and t_0 is the thrust deduction factor at the bollard condition and n_0 the propeller RPM at the bollard condition. The RPM's can be determined using the torque coefficient, from the equations (ρ is the density of water):

$$P = K_Q(0) \cdot 2\pi\rho \cdot n_0^3 \cdot D_P^5$$
$$P = K_Q(J_1) \cdot 2\pi\rho \cdot n_1^3 \cdot D_P^5 \tag{5}$$

In this case, the crucial assumption is that the propeller absorbs full power at the bollard condition and at a speed of 5 knots. This assumption is adequate for diesel-electric drives and CP propellers, but it is not adequate for slow speed engines with a FP propeller and separate, more detailed calculations must be performed.

The basic requirement in the regulations is that, at a speed of 5 knots

$$T_{NET} = R_{CH} \tag{6}$$

from which the power can be calculated. As both the T_B and the RPMs include power, this solution is somewhat complicated. As two points on the T_{NET}-curve are known ($T_{NET} = T_B$ when v = 0 and T_{NET} = 0 when $v = v_{OW}$, open water speed), the situation can be simplified, if a parabolic curve is fitted between these points as follows:

$$T_{NET} = \left(1 - \frac{1}{3}\frac{v}{v_{OW}} - \frac{2}{3}\left(\frac{v}{v_{OW}}\right)^2\right) \cdot T_B \tag{7}$$

Eq. (4) shows that, to determine the net thrust precisely, the open water resistance is needed in addition to the propeller thrust. Based on (3), an estimate can be made of the thrust needed at a speed of 5 knots, if the typical values for open water resistance and the thrust deduction coefficient are used. For a certain, limited number of ships these have been estimated to be 0.12 $\cdot H_M^{-1.3}$ % of R_{CH} ($[H_M] = m$) and t = 0.15, respectively (for ice class **L1A**, the H_M is taken to be 1.3 m and for azimuthing thrusters t = 0.13). These lead to the requirement that the thrust at a speed of 5 knots must be factor$\cdot$ R_{CH}, where the factor is given in Table 2 of Chapter 7.4.1. This value is not a general figure, which means that generalisations of this kind cannot be made in principle. The actual and verified values for R_{OW} and t can be used in any situation.

Another question is posed by propellers in nozzles. At low speeds, nozzle propellers create higher thrust than open propellers of a corresponding size. As a rule of thumb, this extra thrust is given as 30% of the corresponding open propeller thrust. These facts can be fed into an equation if the net thrust, using e.g. (7), is first denoted as

$$T_{NET} = K_V \cdot T_B\,. \tag{8}$$

Then, the extra thrust is taken into account using the factor K_N, and the bollard pull of the nozzled propeller is (T_B as in (1))

$$T_{B,N} = K_N \cdot T_B \text{ and} \tag{9}$$

$$T_{NET} = K_V\, T_{B,N} = K_v K_N T_B \tag{10}$$

By starting with the basic equation (6), we now get

$$\begin{aligned} R_{CH} &= T_{NET} \\ &= K_v\,(v_1) \cdot T_{B,\,N} \\ &= K_v\,(v_1) \cdot K_N \cdot T_B \\ &= K_v\,(v_1) \cdot K_N \cdot K_g \cdot (P \cdot D_P)^{2/3} \end{aligned} \tag{11}$$

In the rule, K_V is assumed to be 0.8. Thus, the power requirement for a nozzle propeller is

$$P_N = \frac{1}{D_P}\left(\frac{R_{CH}}{K_v K_g K_N}\right)^{3/2} = \frac{K_e}{D_P}\left(\frac{R_{CH}}{K_N}\right)^{3/2} = \frac{1}{K_N^{3/2}} \cdot P_{OPEN} \tag{12}$$

This equation shows that, in theory, if the open water propeller has a diameter which is $K_N^{3/2}$ times larger than (i.e. about 1.48 times) the nozzle propeller's diameter, the thrusts are the same. Or, to put it slightly differently, the power of the nozzle propulsion can be around 70% of the corresponding open propulsion, and the performances are the same.

Appendix 3

GUIDELINES FOR BOLLARD PULL TESTS FOR DETERMINING THE THRUST OF THE PROPELLER(S)

The R_{CH} is defined as the channel resistance at a speed of 5 knots in a broken channel of a certain thickness. The propeller thrust must be greater than this channel resistance plus open water resistance.

Regulation 3.2.5 allows for alternative measures, in order to comply with the above requirement. A bollard pull test can thus be accepted as proof that the powering requirement has been fulfilled.

1 BOLLARD PULL TEST 1

By definition, this test is performed at zero speed. To achieve the correct test result, several factors must be considered, e.g. water depth, towline length etc. When conducting these tests, a bollard pull testing procedure by PRS, or the *Bollard Pull Trial Code* by Steerprop, should be followed.

The bollard pull should be measured by a calibrated 'load cell' with a deviation within a measuring range of less than ±2 %.

The measured bollard pull should be no less than that given in Tables 1 and 2. Account must also be taken of open water resistance R_{OW} and the factor t. The actual and verified values for R_{OW} and t can always be used.

2 BOLLARD TOW TEST 2

In practice, this type of test is probably the most convenient one. The vessel is connected to a tug and the two vessels perform a 'tug of war' pull, moving at a minimum speed of 5 knots in the direction of the test ship.

The force should be measured on the tug either by using:

An independent (external) 'load cell' with a deviation within the measuring range of less than ±2 %. The measured tow pull should be no less than $1.0 \cdot R_{CH}$

The integrated 'load cell' on the towing winch. The measured tow pull should be no less than $1.1 \cdot R_{CH}$.

Appendix 4

GUIDELINES FOR THE VERIFICATION OF A SHIP'S PERFORMANCE FOR ICE CLASSES THROUGH MODEL TESTS

In the PRS Ice Class Regulations, powering requirements refer to a certain required level of a ship's performance. The ship's performance is set as the ship's capacity to proceed at a constant speed of 5 knots in old brash ice channels of a certain thickness. For ice class **L1A**, it is also assumed that there is a 10 cm thick consolidated layer of ice on top of the channel. When verifying performance through model tests, the following points 2 to 7 should be checked:

1 Definition of the Design Point and Notation

The design point to be checked by the model tests is that the vessel can proceed at five knots in the brash ice channel specified for each ice class. This definition can be used in the propulsion tests, if the propulsion thrust to be obtained at full scale is modelled. If, however, resistance tests are conducted, then the total resistance in ice, R_{iTOT}, is measured. In the ice class regulations, it is assumed that the superposition assumption is valid. This states that the pure ice resistance, R_i, and open water resistance, R_{OW}, can be superimposed as

$$R_{iTOT} = R_i + R_{OW}.$$

Here, the pure ice resistance is either the channel resistance (ice classes **L1**, **L2** or **L3**) or the channel resistance plus level ice resistance (ice class L1A). Now, the ice class requirement is

$$T \cdot (1 - t) \geq R_{iTOT} = R_i + R_{OW},$$

where T is the thrust that the propeller develops at 5 knots and t is the thrust deduction factor at 5 knots (and in principle at the overload condition – the open water thrust deduction factor can, however, be used).

2 The Model Testing Procedure

The rule requirement is that the ship achieves at least 5 knots in a channel defined separately for each ice class. The rule resistance in the specified channel is given in the regulations. The aim of the model tests is to determine the channel resistance and the total resistance in the channel i.e. the channel and open water resistances, and then show that there is enough net propulsion thrust (i.e. taking account of the thrust deduction factor) available to overcome this resistance.

The results of the model tests should show the channel resistance, open water resistance at the same speed and the net thrust of the proposed propulsion arrangement at full scale at the specified speed. A propulsion test using stock propellers showing a self-propulsion point at a certain speed is insufficient. This is also reflected in the reporting requirements.

3 The Geometry of the Ice Channel

The rule-based channels have been given a thickness with respect to their mid part (H_M = 1m for **L1**, 0.8 m for **L2** and 0.6 m for **L3**), and their profile thickens towards the edges by a gradient of 2°, see Figure 1. This profile is based on channel measurements in the fairways of northern ports in the Gulf of Bothnia.

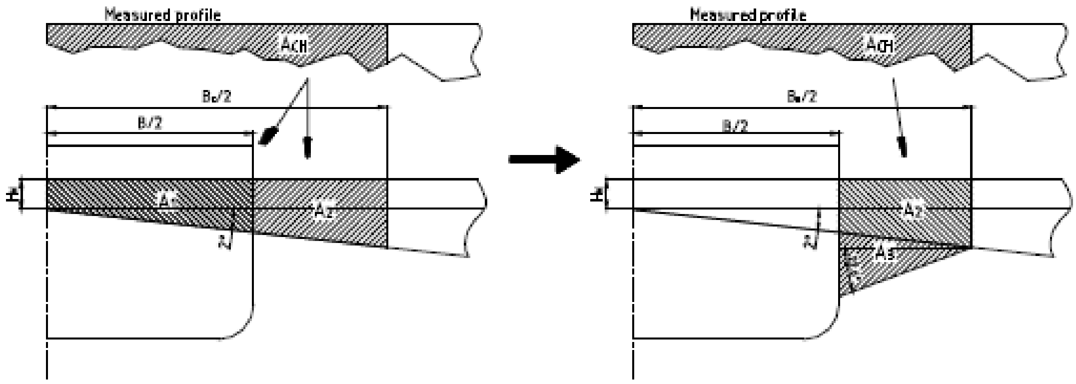


Figure 1.
The geometry of a "real" brash ice channel profile and the corresponding profile of a rule channel before the ship's passage, and the assumed behaviour of the ice during the passage. The relations between the cross-sectional areas are:

$$A_{CH} = A_1 + A_2 = A_2 + A_3.$$

However, it is difficult to achieve a channel profile in model test conditions resembling the ones referred to in the regulations. An average channel thickness, Hav, may thus be used, which is affected by the breadth of the ship, as follows:

$$H_{av} = H_M + 14.0 \cdot 10^{-3} B \quad (1)$$

where B is the beam of the vessel.

The width of the ice channel should be 2 x B, with level ice at the sides. The thickness profile of the ice channel should be measured at a breadth of 1.6 x B.

The channel profile should be measured at sufficiently small intervals (of around 10 … 20 cm) in order to ensure that the cross sectional area of the ice channel is accurately determined. In the longitudinal direction, the step of cross sectional profiling should be a maximum of 2 m.

The channel should be 100% covered with ice, so that there are around two layers of ice fragments on top of each other.

4 The Friction Coefficient

At full scale, the friction coefficient between ice floes and the hull, μ, ranges from 0.05 for new ships to 0.15 for corroded hull surfaces. In ice model tests, a friction factor of 0.05 – 0.1 is usually applied to the model.

If a friction coefficient of less than 0.1 is used in the model tests to determine the ice channel resistance R_{CH}, the engine power and the propeller thrust should be selected so that the vessel is able to sail at a speed of 5 knots with a friction coefficient of 0.1 at full scale. Correction of the resistance to account for a different friction coefficient can be done using the following formula:

$$R_{CH(with\ \mu_{target})} = \left[\left(0.6 + 4\mu_{target}\right)/\left(0.6 + 4\mu_{actual}\right)\right] R_{CH(with \mu_{actual})}$$

where μ_{actual} is the actual friction coefficient measured in the tests and $\mu_{target} = 0.1$.

5 Model Tests For Ice Class L1A

The preparation of a consolidated ice layer for ice class **L1A** is difficult, since it often becomes very inhomogeneous (the fragments are very small in a natural setting, but are often larger in the test channels) or too intact (resembling natural ice). For this reason, these tests could be carried out by superposing the level ice resistance and the channel resistance.

6 Determination of the Propulsion Power at Full Scale

When R_{CH} has been determined using model tests, and this resistance is used to verify compliance with the speed requirements for the applicable ice class, the actual propeller thrust at full scale should be applied using the actual propeller and engine data, instead of using the rule formulae, in order to ensure that the propulsion system is able to produce the thrust required to overcome the channel resistance. This is particularly important for low-speed, direct-drive diesel engines with an FPP. If it is observed that the ice is interacting extensively with the propeller during the tests, the resulting losses in propulsion must be taken into account in the calculation of the full scale power and speed of the ship. For the verification of the model scale bollard pull, the corresponding full-scale bollard pull must be given for the propeller to be used on the ship.

7 Model Test Reporting

The model test report should include the information given in the Annex to these Guidelines.

Annex to Appendix 4

REQUIRED INFORMATION IN A MODEL TEST REPORT

The following information should be included in a model test report submitted in order to have the engine power accepted in accordance with section 3.2.5 of the PRS Ice Class regulations equivalent to Finish-Swedish Ice Class Rules, 2017.

1. General description of the ice model basin and the model ice

2. Ship model
   - 2.1 Main particulars of the ship, including displacement and deadweight
   - 2.2 Main particulars of the model
   - 2.3 Description of the ship geometry, with hull lines drawing
   - 2.4 Model scale

3. Propulsion
   - 3.1 Description of the ship propulsion system, including the net thrust and bollard pull curves
   - 3.2 Description of the model propellers
   - 3.3 The bollard pull versus the RPM curve of the ship model

4. Test program and procedures
   - 4.1 Model test program
   - 4.2 Hull friction coefficient measurement procedure
   - 4.3 Description of the measurement system for propulsion values
   - 4.4 Description of the measurement system in resistance and/or propulsion tests
   - 4.5 Analysis procedures

5. Model ice
   - 5.1 Data on the parent level ice thickness
   - 5.2 Parent level ice strength (bending strength and also, preferably, compressive strength)
   - 5.3 Description of the method for producing the channel
   - 5.4 Measurement of the channel profile at sufficiently small intervals (intervals of around 10 … 20 cm) to allow the accurate determination of the cross sectional area of the channel. In a longitudinal direction, the cross sectional profiling interval should be 2 m at most. The methods used in this measurement should be described.
   - 5.5 From each cross section, an average channel thickness should becomputed based on a channel width which is the breadth of the ship and 1.6 times the ship's breadth.
   - 5.6 Description of the porosity of the brash ice. Photographs from above thec hannel, to provide a picture of the brash ice coverage along the entire length of the channel.
   - 5.7 For the ice class L1A, it is assumed that a consolidated layer 10 cm in thickness (full scale) is lying on top of the brash ice. If this layer is modelled, the modelling procedure should be described, including the manner in which it was produced and how its thickness and strength were measured

6. Test results
   - 6.1 Measurements of the hull coefficient for friction with ice
   - 6.2 The time histories of the model speed, propeller thrust, torque and RPM derived from each test. Indication of the part of the time history based on which the final values were calculated.
   - 6.3 Description of the behaviour of the brash ice in the channel. A measurement of the cohesion and internal friction angle or some other parameters describing the strength

of the brash ice should be performed, or an earlier result for these quantities should be produced in a similar manner based on a brash ice channel.

6.4 Photographs of the channel made by the vessel immediately after the tests, from above.

6.5 The deduced (from time histories referred to in section 6.2 above) and calculated model propulsion, total model resistance and ice resistance values

6.6 Full-scale resistance and engine power prediction, including a description of the extrapolation method. An estimate must be given of the accuracy of the result obtained by extrapolation.

7. Other information

7.1 Estimate of the resistance of the model in open water.

7.2 Calculation of the required engine power according to the PRS Regulations, 2010, with input data.

PART II

PRS Polar Class Regulations

I1 PRS POLAR CLASS DESCRIPTIONS AND APPLICATION

I1.1 Application

I1.1.1 The PRS requirements for Polar Class ships apply to ships constructed of steel and intended for independent navigation in ice-infested polar waters.

I1.1.2 Ships that comply with the requirements can be considered for a Polar Class notation as listed in Table 1. The requirements of Polar Class are in addition to the open water requirements of the PRS. If the hull and machinery are constructed such as to comply with the requirements of different Polar Classes, then both the hull and machinery are to be assigned the lower of these classes in the Certificate of Classification. Compliance of the hull or machinery with the requirements of a higher Polar Class is also to be indicated in the Certificate of Classification or equivalent.

I1.1.3 Ships which are assigned a Polar Class notation and complying with the relevant requirements of PRS Rules may be given the additional notation "Icebreaker". "Icebreaker" refers to any ship having an operational profile that includes escort or ice management functions, having powering and dimensions that allow it to undertake aggressive operations in ice-covered waters.

I1.1.4 For ships which are assigned a Polar Class notation, the hull form and propulsion power are to be such that the ship can operate independently and at continuous speed in a representative ice condition, as defined in Table 1 for the corresponding Polar Class. For ships and ship-shaped units which are intentionally not designed to operate independently in ice, such operational intent or limitations are to be explicitly stated in the Certificate of Classification or equivalent.

I1.1.5 For ships which are assigned a Polar Class notation **PC1** through **PC5**, bows with vertical sides, and bulbous bows are generally to be avoided. Bow angles should in general be within the range specified in I2.3.1 (v).

I1.1.6 For ships which are assigned a Polar Class notation **PC6** and **PC7**, and are designed with a bow with vertical sides or bulbous bows, operational limitations (restricted from intentional ramming) in design conditions are to be stated in the Certificate of Classification or equivalent.

I1.2 Polar Classes

I1.2.1 The Polar Class (PRS PC) notations and descriptions are given in Table 1. It is the responsibility of the Owner to select an appropriate Polar Class. The descriptions in Table 1 are intended to guide owners, designers and administrations in selecting an appropriate Polar Class to match the requirements for the ship with its intended voyage or service.

I1.2.2 The PRS Polar Class notation is used throughout the requirements for Polar Class ships to convey the differences between classes with respect to operational capability and strength.

Table 1 – PRS Polar Class descriptions

Polar Class	Ice descriptions (based on WMO Sea Ice Nomenclature)
PC1	Year-round operation in all polar waters
PC2	Year-round operation in moderate multi-year ice conditions
PC3	Year-round operation in second-year ice which may include multi-year ice inclusions.
PC4	Year-round operation in thick first-year ice which may include old ice inclusions
PC5	Year-round operation in medium first-year ice which may include old ice inclusions
PC6	Summer/autumn operation in medium first-year ice which may include old ice inclusions
PC7	Summer/autumn operation in thin first-year ice which may include old ice inclusions

I1.3 Upper and Lower Ice Waterlines

I1.3.1 The upper and lower ice waterlines upon which the design of the ship has been based is to be indicated in the Certificate of Classification. The upper ice waterline (UIWL) is to be defined by the maximum draughts fore, amidships and aft. The lower ice waterline (LIWL) is to be defined by the minimum draughts fore, amidships and aft.

I1.3.2 The lower ice waterline is to be determined with due regard to the ship's ice-going capability in the ballast loading conditions). The propeller is to be fully submerged at the lower ice waterline.

I2 STRUCTURAL REQUIREMENTS FOR PRS POLAR CLASS SHIPS

I2.1 General

I2.1.1 These requirements apply to PRS Polar Class ships according to I1.1.

I2.1.1.1 Definitions

I2.1.1.2 The length L_{UI} is the distance, in m, measured horizontally from the fore side of the stem at the intersection with the upper ice waterline (UIWL) to the after side of the rudder post, or the centre of the rudder stock if there is no rudder post. L_{UI} is not to be less than 96%, and need not be greater than 97%, of the extreme length of the upper ice waterline (UIWL) measured horizontally from the fore side of the stem. In ships with unusual stern and bow arrangement the length L_{UI} will be specially considered.

I2.1.1.3 The ship displacement D_{UI} is the displacement, in kt, of the ship corresponding to the upper ice waterline (UIWL). Where multiple waterlines are used for determining the UIWL, the displacement is to be determined from the waterline corresponding to the greatest displacement.

I2.2 Hull areas

I2.2.1 The hull of Polar Class ships is divided into areas reflecting the magnitude of the loads that are expected to act upon them. In the longitudinal direction, there are four regions: Bow, Bow Intermediate, Midbody and Stern. The Bow Intermediate, Midbody and Stern regions are further divided in the vertical direction into the Bottom, Lower and Icebelt regions. The extent of each hull area is illustrated in Figure 1.

I2.2.2 The upper ice waterline (UIWL) and lower ice waterline (LIWL) are as defined in UR I1.3.

I2.2.3 Figure 1 notwithstanding, at no time is the boundary between the Bow and Bow Intermediate regions to be forward of the intersection point of the line of the stem and the ship baseline.

I2.2.4 Figure 1 notwithstanding, the aft boundary of the Bow region need not be more than 0.45 L_{UI} aft of the fore side of the stem at the intersection with the upper ice waterline (UIWL).

I2.2.5 The boundary between the bottom and lower regions is to be taken at the point where the shell is inclined 7° from horizontal.

I2.2.6 If a ship is intended to operate astern in ice regions, the aft section of the ship is to be designed using the Bow and Bow Intermediate hull area requirements.

I2.2.7 Figure 1 notwithstanding, if the ship is assigned the additional notation “Icebreaker”, the forward boundary of the stern region is to be at least 0.04 L_{UI} forward of the section where the parallel ship side at the upper ice waterline (UIWL) ends.

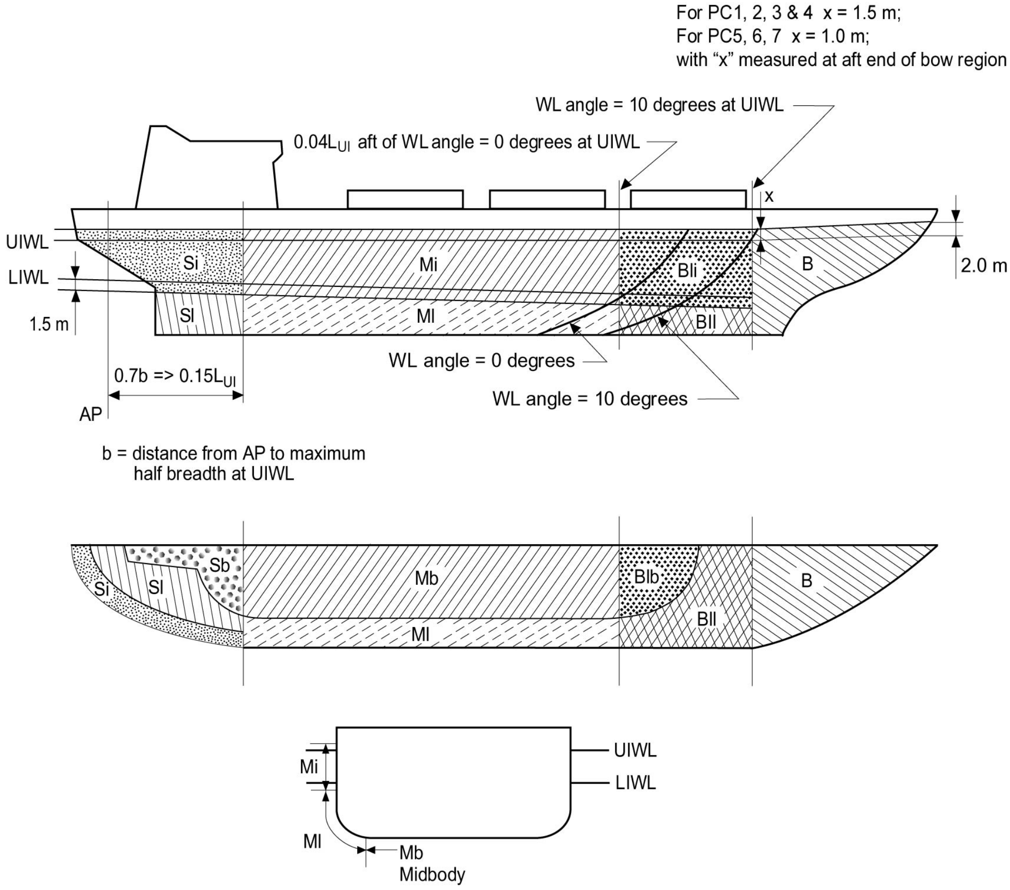


Figure 1 – Hull area extents

I2.3 Design ice loads

I2.3.1 General

(i) A glancing impact on the bow is the design scenario for determining the scantlings required to resist ice loads.

(ii) The design ice load is characterized by an average pressure (P_{avg}) uniformly distributed over a rectangular load patch of height (b) and width (w).

(iii) Within the Bow area of all Polar Class ships, and within the Bow Intermediate Icebelt area of Polar Class **PC6** and **PC7**, the ice load parameters are functions of the actual bow shape. To determine the ice load parameters (P_{avg}, b and w), it is required to calculate the following ice load characteristics for sub-regions of the bow area; shape coefficient (fa_i), total glancing impact force (F_i), line load (Q_i) and pressure (P_i).

(iv) In other ice-strengthened areas, the ice load parameters (P_{avg}, b_{NonBow} and w_{NonBow}) are determined independently of the hull shape and based on a fixed load patch aspect ratio, AR = 3.6.

(v) Design ice forces calculated according to I2.3.2.1 (iii) are applicable for bow forms where the buttock angle γ at the stem is positive and less than 80 deg, and the normal frame angle β' at the centre of the foremost sub-region, as defined in I2.3.2.1 (i), is greater than 10 deg.

(vi) Design ice forces calculated according to I2.3.2.1 (iv) are applicable for ships which are assigned the Polar Class **PC6** or **PC7** and have a bow form with vertical sides. This includes bows where the normal frame angles β' at the considered sub-regions, as defined in I2.3.2.1 (i), are between 0 and 10 deg.

(vii) For ships which are assigned the Polar Class **PC6** or **PC7**, and equipped with bulbous bows, the design ice forces on the bow are to be determined according to I2.3.2.1 (iv). In addition, the design forces are not to be taken less than those given in I2.3.2.1 (iii), assuming fa = 0.6 and AR = 1.3.

(viii) For ships with bow forms other than those defined in (v) to (vii), design forces are to be specially considered by the PRS.

(ix) Ship structures that are not directly subjected to ice loads may still experience inertial loads of stowed cargo and equipment resulting from ship/ice interaction. These inertial loads, based on accelerations determined by the PRS, are to be considered in the design of these structures.

I2.3.2 Glancing impact load characteristics

The parameters defining the glancing impact load characteristics are reflected in the Class Factors listed in Table 1 and Table 2.

Table 1 – Class factors to be used in I2.3.2.1 (iii)

Polar Class	Crushing failure Class Factor (CF_C)	Flexural failure Class Factor (CF_F)	Load patch dimensions Class Factor (CF_D)	Displacement Class Factor (CF_{DIS})	Longitudinal strength Class Factor (CF_L)
PC1	17.69	68.60	2.01	250	7.46
PC2	9.89	46.80	1.75	210	5.46
PC3	6.06	21.17	1.53	180	4.17
PC4	4.50	13.48	1.42	130	3.15
PC5	3.10	9.00	1.31	70	2.50
PC6	2.40	5.49	1.17	40	2.37
PC7	1.80	4.06	1.11	22	1.81

Table 2 – Class factors to be used in I2.3.2.1 (iv)

Polar Class	Crushing failure Class Factor (CF_{CV})	Line load Class Factor (CF_{QV})	Pressure Class Factor (CF_{PV})
PC6	3.43	2.82	0.65
PC7	2.60	2.33	0.65

I2.3.2.1 Bow area

(i) In the Bow area, the force (F), line load (Q), pressure (P) and load patch aspect ratio (AR) associated with the glancing impact load scenario are functions of the hull angles measured at the upper ice waterline (UIWL). The influence of the hull angles is captured through calculation of a bow shape coefficient (fa). The hull angles are defined in Figure 2.

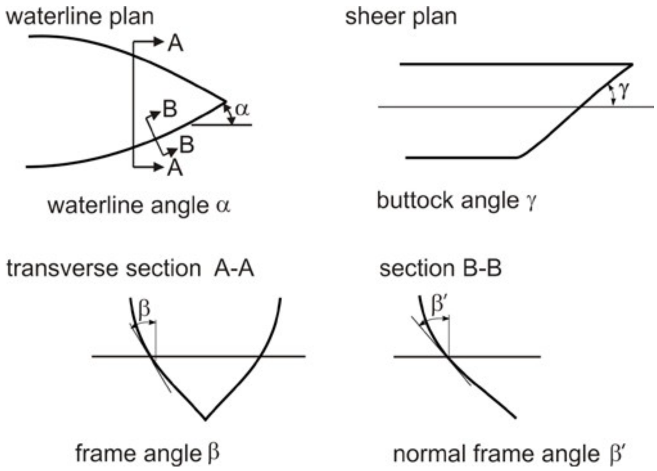


Figure 2 – Definition of hull angles

Note: β' = normal frame angle at upper ice waterline [deg]
α = upper ice waterline angle [deg]
γ = buttock angle at upper ice waterline (angle of buttock line measured from horizontal) [deg]
$\tan(\beta) = \tan(\alpha)/\tan(\gamma)$
$\tan(\beta') = \tan(\beta)\cdot\cos(\alpha)$

(ii) The waterline length of the bow region is generally to be divided into 4 sub-regions of equal length. The force (F), line load (Q), pressure (P) and load patch aspect ratio (AR) are to be calculated with respect to the mid-length position of each sub-region (each maximum of F, Q and P is to be used in the calculation of the ice load parameters P_{avg}, b and w).

(iii) The Bow area load characteristics for bow forms defined in I2.3.1 (v) are determined as follows:

a) Shape coefficient, fa_i, is to be taken as

fa_i = minimum ($fa_{i,1}$; $fa_{i,2}$; $fa_{i,3}$)

where:

$fa_{i,1} = (0.097 - 0.68 \cdot (x/L - 0.15)^2) \cdot \alpha_i / (\beta'_i)^{0.5}$

$fa_{i,2} = 1.2 \cdot CF_F / (\sin(\beta'_i) \cdot CF_C \cdot D^{0.64})$

$fa_{i,3} = 0.60$

b) Force, F_i

$F_i = fa_i \cdot CF_C \cdot D^{0.64}$ [MN]

c) Load patch aspect ratio, AR_i:

$AR_i = 7.46 \cdot \sin(\beta'_i) \geq 1.3$

d) Line load, Qi:

$Q_i = F_i^{0.61} \cdot CF_D / AR_i^{0.35}$ [MN/m]

e) Pressure, P_i:

$P_i = F_i^{0.22} \cdot CF_D^2 \cdot AR_i^{0.3}$ [MPa]

where

i = sub-region considered

L_{UI} = length as defined in I2.1.2.1 [m]

x = distance from the fore side of the stem at the intersection with the upper ice waterline (UIWL) to station under consideration [m]
α = waterline angle [deg], see Figure 2
β′ = normal frame angle [deg], see Figure 2
D_{UI} = displacement as defined in I2.1.2.2, not to be taken less than 5 [kt]
CF_C = Crushing failure Class Factor from Table 1
CF_F = Flexural failure Class Factor from Table 1
CF_D = Load patch dimensions Class Factor from Table 1

(iv) The Bow area load characteristics for bow forms defined in I2.3.1 (vi) are determined as follows:

a) Shape coefficient, fa_i, is to be taken as

$fa_i = \alpha_i/30$

b) Force, F_i:

$F_i = fa_i \cdot CF_{CV} \cdot D^{0.47}$ [MN]

c) Line load, Q_i:

$Q_i = F_i^{0.22} \cdot CF_{QV}$ [MN/m]

d) Pressure, Pi:

$P_i = F^{0.56} \cdot CF_{PV}$ [MPa]

where
i = sub-region considered
α = waterline angle [deg], see Figure 2
D_{UI} = displacement as defined in I2.1.2.2, not to be taken less than 5 [kt]
CF_{CV} = Crushing failure Class Factor from Table 2
CF_{QV} = Line load Class Factor from Table 2
CF_{PV} = Pressure Class Factor from Table 2.

I2.3.2.2 Hull areas other than the bow

(i) In the hull areas other than the bow, the force (F_{NonBow}) and line load (Q_{NonBow}) used in the determination of the load patch dimensions (b_{NonBow}, w_{NonBow}) and design pressure (P_{avg}) are determined as follows:

a) Force, F_{NonBow}:

$$F_{NonBow} = 0.36 \cdot CF_C \cdot DF \text{ [MN]}$$

b) Line Load, Q_{NonBow}:

$$Q_{NonBow} = 0.639 \cdot F_{NonBow}^{0.61} \cdot CF_D \text{ [MN/m]}$$

where
CF_C = Crushing failure Class Factor from Table 1
DF = ship displacement factor
= $D^{0.64}$ if $D \le CF_{DIS}$
= $CF_{DIS}^{0.64} + 0.10 \cdot (D - CF_{DIS})$ if $D > CF_{DIS}$
D_{UI} = displacement as defined in I2.1.2.2, not to be taken less than 10
[kt] CF_{DIS} = Displacement Class Factor from Table 1
CF_D = Load patch dimensions Class Factor from Table 1.

I2.3.3 Design load patch

(i) In the Bow area, and the Bow Intermediate Icebelt area for ships with class notation **PC6** and **PC7**, the design load patch has dimensions of width, w_{Bow}, and height, b_{Bow}, defined as follows:

$w_{Bow} = F_{Bow}/Q_{Bow}$ [m]

$b_{Bow} = Q_{Bow}/P_{Bow}$ [m]

where:

F_{Bow} = maximum force F_i in the Bow area [MN]
Q_{Bow} = maximum line load Q_i in the Bow area [MN/m]
P_{Bow} = maximum pressure P_i in the Bow area [MPa]

(ii) In hull areas other than those covered by I2.3.3 (i), the design load patch has dimensions of width, w_{NonBow}, and height, b_{NonBow}, defined as follows:

$w_{NonBow} = F_{NonBow}/Q_{NonBow}$ [m]

$b_{NonBow} = w_{NonBow}/3.6$ [m]

where:

F_{NonBow} = force as defined in I2.3.2.2 (i) (a) [MN]

Q_{NonBow} = line load as defined in I2.3.2.2 (i) (b) [MN/m].

I2.3.4 Pressure within the design load patch

(i) The average pressure, P_{avg}, within a design load patch is determined as follows:

$$P_{avg} = F / (b \cdot w) \text{ [MPa]}$$

where:
$F = F_{Bow}$ or F_{NonBow} as appropriate for the hull area under consideration [MN]
$b = b_{Bow}$ or b_{NonBow} as appropriate for the hull area under consideration [m]
$w = w_{Bow}$ or w_{NonBow} as appropriate for the hull area under consideration [m]

(ii) Areas of higher, concentrated pressure exist within the load patch. In general, smaller areas have higher local pressures. Accordingly, the peak pressure factors listed in Table 3 are used to account for the pressure concentration on localized structural members.

Table 3 – Peak Pressure Factors

Structural member		Peak Pressure Factor (PPFi)
Plating	Transversely-framed	$PPF_p = (1.8 - s) \geq 1.2$
	Longitudinally framed	$PPF_p = (2.2 - 1.2 \cdot s) \geq 1.5$
Frames in transverse framing systems Transverse Framing Systems	With load distributing stringers	$PPF_t = (1.6 - s) \geq 1.0$
	With no load distributing stringers	$PPF_t = (1.8 - s) \geq 1.2$
Frames in bottom structures		$PPF_s = 1.0$
Load-carrying stringers Side-longitudinals Web frames		$PPF_s = 1.0$, if $S_w \geq 0.5 \cdot w$ $PPF_s = 2.0 - 2.0 \cdot S_w/w$, if $S_w < (0.5 \cdot w)$
where: s = frame or longitudinal spacing [m] S_w = web frame spacing [m] w = ice load patch width [m]		

I2.3.5 Hull area factors

(i) Associated with each hull area is an Area Factor that reflects the relative magnitude of the load expected in that area. The Area Factor (AF) for each hull area is listed in Table 4.
(ii) In the event that a structural member spans across the boundary of a hull area, the largest hull area factor is to be used in the scantling determination of the member.
(iii) Due to their increased manoeuvrability, ships having propulsion arrangements with azimuth thruster(s) or "podded" propellers are to have specially considered Stern Icebelt (Si) and Stern Lower (Sl) hull area factors.
(iv) For ships assigned the additional notation "Icebreaker", the Area Factor (AF) for each hull area is listed in Table 5.

Table 4 – Hull Area Factors (AF)

Hull area		Area	PRS Polar Class						
			PC1	**PC2**	**PC3**	**PC4**	**PC5**	**PC6**	**PC7**
Bow (B)	All	B	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Bow Intermediate (BI)	Icebelt	BI_i	0.90	0.85	0.85	0.80	0.80	1.00*	1.00*
	Lower	BI_l	0.70	0.65	0.65	0.60	0.55	0.55	0.50
	Bottom	BI_b	0.55	0.50	0.45	0.40	0.35	0.30	0.25
Midbody (M)	Icebelt	M_i	0.70	0.65	0.55	0.55	0.50	0.45	0.45
	Lower	M_l	0.50	0.45	0.40	0.35	0.30	0.25	0.25
	Bottom	M_b	0.30	0.30	0.25	**	**	**	**
Stern (S)	Icebelt	S_i	0.75	0.70	0.65	0.60	0.50	0.40	0.35
	Lower	S_l	0.45	0.40	0.35	0.30	0.25	0.25	0.25
	Bottom	S_b	0.35	0.30	0.30	0.25	0.15	**	**

Note to Table 4:
* See I2.3.1 (iii).
** Indicates that strengthening for ice loads is not necessary.

Table 5 – Hull Area Factors (AF) for ships with additional notation "Icebreaker"

Hull area		Area	PRS Polar Class						
			PC1	**PC2**	**PC3**	**PC4**	**PC5**	**PC6**	**PC7**
Bow (B)	All	B	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Bow Intermediate (BI)	Icebelt	BI_i	0.90	0.85	0.85	0.85	0.85	1.00	1.00
	Lower	BI_l	0.70	0.65	0.65	0.65	0.65	0.65	0.65
	Bottom	BI_b	0.55	0.50	0.45	0.45	0.45	0.45	0.45
Midbody (M)	Icebelt	M_i	0.70	0.65	0.55	0.55	0.55	0.55	0.55
	Lower	M_l	0.50	0.45	0.40	0.40	0.40	0.40	0.40
	Bottom	M_b	0.30	0.30	0.25	0.25	0.25	0.25	0.25
Stern (S)	Icebelt	S_i	0.95	0.90	0.80	0.80	0.80	0.80	0.80
	Lower	S_l	0.55	0.50	0.45	0.45	0.45	0.45	0.45
	Bottom	S_b	0.35	0.30	0.30	0.30	0.30	0.30	0.30

I2.4 Shell plate requirements

I2.4.1 The required minimum shell plate thickness, t, is given by:

$$t = t_{net} + t_s \text{ [mm]}$$

where:
t_{net} = plate thickness required to resist ice loads according to I2.4.2 [mm]
t_s = corrosion and abrasion allowance according to I2.11 [mm].

I2.4.2 The thickness of shell plating required to resist the design ice load, t_{net}, depends on the orientation of the framing.

In the case of transversely-framed plating (Ω ≥ 70 deg), including all bottom plating, i.e. plating in hull areas BI_b, M_b and S_b, the net thickness is given by:

$$t_{net} = 500 \cdot s \cdot ((AF \cdot PPF_p \cdot P_{avg}) / \sigma_y)^{0.5} / (1 + s / (2 \cdot b)) \text{ [mm]}$$

In the case of longitudinally-framed plating (Ω ≤ 20 deg), when b ≥ s, the net thickness is given by:

$$t_{net} = 500 \cdot s \cdot ((AF \cdot PPF_p \cdot P_{avg}) / \sigma y)^{0.5} / (1 + s / (2 \cdot l)) \text{ [mm]}$$

In the case of longitudinally-framed plating (Ω ≤ 20 deg), when b < s, the net thickness is given by:

$$t_{net} = 500 \cdot s \cdot ((AF \cdot PPF_p \cdot P_{avg}) / \sigma_y)^{0.5} \cdot (2 \cdot b / s - (b / s)^2)^{0.5} / (1 + s / (2 \cdot l)) \text{ [mm]}$$

In the case of obliquely-framed plating (70 deg > Ω > 20 deg), linear interpolation is to be used.

where:

Ω = smallest angle between the chord of the waterline and the line of the first level framing as illustrated in Figure 3 [deg].

s = transverse frame spacing in transversely-framed ships or longitudinal frame spacing in longitudinally-framed ships [m]

AF = Hull Area Factor from Table 4 or Table 5

PPF_p = Peak Pressure Factor from Table 3

P_{avg} = average patch pressure as defined in I2.3.4 [MPa]

σ_y = minimum upper yield stress of the material [N/mm^2]

b = height of design load patch [m], where b ≤ is to be taken not greater than (l – s/4) in the case of determination of the net thickness for transversely framed plating.

l = distance between frame supports, i.e. equal to the frame span as given in I2.5.5, but not reduced for any fitted end brackets [m]. When a load-distributing stringer is fitted, the length l need not be taken larger than the distance from the stringer to the most distant frame support.

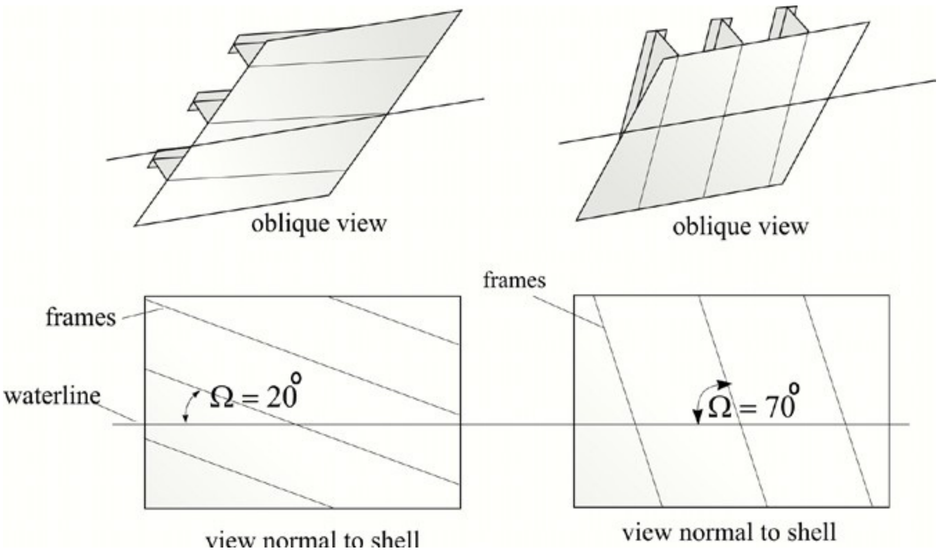


Figure 3 – Shell framing angle Ω

I2.5 Framing - General

I2.5.1 Framing members of Polar Class -ships are to be designed to withstand the ice loads defined in I2.3.

I2.5.2 The term "framing member" refers to transverse and longitudinal local frames, load-carrying stringers and web frames in the areas of the hull exposed to ice pressure, see Figure 1.

Where load-distributing stringers have been fitted, the arrangement and scantlings of these are to be in accordance with the requirements of the PRS.

I2.5.3 The strength of a framing member is dependent upon the fixity that is provided at its supports. Fixity can be assumed where framing members are either continuous through the support or attached to a supporting section with a connection bracket. In other cases, simple support is to be assumed unless the connection can be demonstrated to provide significant rotational restraint. Fixity is to be ensured at the support of any framing which terminates within an ice-strengthened area.

I2.5.4 The details of framing member intersection with other framing members, including plated structures, as well as the details for securing the ends of framing members at supporting sections, are to be in accordance with the requirements of the PRS.

I2.5.5 The effective span of a framing member is to be determined on the basis of its moulded length. If brackets are fitted, the effective span may be reduced in accordance with the usual practice of the PRS.

Brackets are to be configured to ensure stability in the elastic and post-yield response regions.

I2.5.6 When calculating the section modulus and shear area of a framing member, net thicknesses of the web, flange (if fitted) and attached shell plating are to be used. The shear area of a framing member may include that material contained over the full depth of the member, i.e. web area including portion of flange, if fitted, but excluding attached shell plating.

I2.5.7 The actual net effective shear area, A_w, of a transverse or longitudinal local frame is given by:

$$A_w = h \cdot t_{wn} \cdot sin\varphi_w / 100 \quad [\text{cm}^2]$$

where:

h = height of stiffener [mm], see Figure 4
t_{wn} = net web thickness [mm]
= $t_w - t_c$
t_w = as built web thickness [mm], see Figure 4
t_c = corrosion deduction [mm] to be subtracted from the web and flange thickness (as specified by the PRS, but not less than t_s as required by I2.11.3).
φ_w = smallest angle between shell plate and stiffener web, measured at the midspan of the stiffener, see Figure 4. The angle φ_w may be taken as 90 deg-provided the smallest angle is not less than 75 deg.

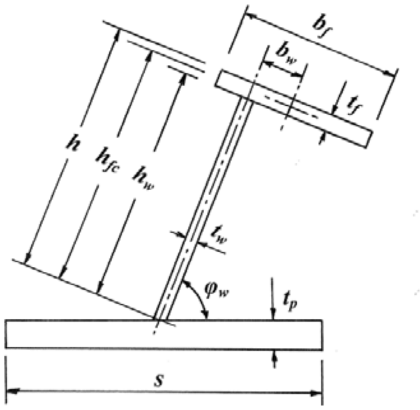


Figure 4 – Stiffener geometry

I2.5.8 When the cross-sectional area of the attached plate flange exceeds the cross-sectional area of the local frame, the actual net effective plastic section modulus, Z_p, of a transverse or longitudinal frame is given by:

$$Z_p = A_{pn} \cdot t_{pn}/20 + \frac{h_w^2 \cdot t_{wn} \cdot sin\varphi_w}{2000} + A_{fn} \cdot (h_{fc} \cdot sin\varphi_w - b_w \cdot cos\varphi_w)/10 \quad [\text{cm}^3]$$

where:

h, t_{wn}, t_c, and φ_w as given in I2.5.7 and s as given in I2.4.2.

A_{pn} = net cross-sectional area of the local frame [cm^2]

t_{pn} = fitted net shell plate thickness [mm] (complying with t_{net} as required by I2.4.2)

h_w = height of local frame web [mm], see Figure 4

A_{fn} = net cross-sectional area of local frame flange [cm^2]

h_{fc} = height of local frame measured to centre of the flange area [mm], see Figure 4

b_w = distance from mid thickness plane of local frame web to the centre of the flange area [mm], see Figure 4.

When the cross-sectional area of the local frame exceeds the cross-sectional area of the attached plate flange, the plastic neutral axis is located at distance z_{na} above the attached shell plate, given by:

$$z_{na} = (100 \cdot A_{fn} + h_w \cdot t_{wn} - 1000 \cdot t_{pn} \cdot s)/(2 \cdot t_{wn}) \text{ [mm]}$$

and the net effective plastic section modulus, Z_p, of a transverse or longitudinal frame is given by:

$$Z_p = t_{pn} \cdot s \cdot \left(z_{na} + \frac{t_{pn}}{2}\right) \cdot sin\varphi_w + \left(\frac{((h_w - z_{na})^2 + z_{na}^2) \cdot t_{wn} \cdot sin\varphi_w}{2000} + A_{fn} \cdot \left((h_{fc} - z_{na}) \cdot sin\varphi_w - b_w \cdot cos\varphi_w\right)/10\right) \text{ [cm}^3\text{]}$$

I2.5.9 In the case of oblique framing arrangement (70 deg > Ω > 20 deg, where Ω is defined as given in I2.4.2), linear interpolation is to be used.

I2.6 Framing – Local frames in bottom structures and transverse local frames in side structures

I2.6.1 The local frames in bottom structures (i.e. hull areas B_{Ib}, M_b and S_b) and transverse local frames in side structures are to be dimensioned such that the combined effects of shear and bending do not exceed the plastic strength of the member. The plastic strength is defined by the magnitude of midspan load that causes the development of a plastic collapse mechanism. For bottom structure the patch load shall be applied with the dimension (b) parallel with the frame direction.

I2.6.2 The actual net effective shear area of the frame, A_w, as defined in I2.5.7, is to comply with the following condition: $A_w \geq A_t$, where:

$$A_t = 100^2 \cdot 0.5 \cdot LL \cdot s \cdot (AF \cdot PPF \cdot P_{avg}) / (0.577 \cdot \sigma_y) \text{ [cm}^2\text{]}$$

where:

LL = length of loaded portion of span
= lesser of a and b [m]
a = local frame span as defined in I2.5.5 [m]
b = height of design ice load patch as defined in I2.3.3 (i) or I2.3.3 (ii)[m]
s = spacing of local frame [m]
AF = Hull Area Factor from Table 4 or Table 5
PPF = Peak Pressure Factor, PPF_t or PPF_s as appropriate from Table 3
P_{avg} = average pressure within load patch as defined in I2.3.4 [MPa]
σ_y = minimum upper yield stress of the material [N/mm^2]

I2.6.3 The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in I2.5.8, is to comply with the following condition: $Z_p \geq Z_{pt}$, where Z_{pt} is to be the greater calculated on the basis of two load conditions: a) ice load acting at the midspan of the local frame, and b) the ice load acting near a support. The A_1 parameter defined below reflects these two conditions:

$$Z_{pt} = 100^3 \cdot LL \cdot Y \cdot s \cdot (AF \cdot PPF \cdot P_{avg}) \cdot a \cdot A_1 / (4 \cdot \sigma_y) \text{ [cm}^3\text{]}$$

where:

AF, PPF, P_{avg}, LL, b, s, a and σ_y are as given in I2.6.2
$Y = 1 - 0.5 \cdot (LL / a)$
A_1 = maximum of
$A_{1A} = 1 / (1 + j / 2 + k_w \cdot j / 2 \cdot [(1 - a_1^2)^{0.5} - 1])$
$A_{1B} = (1 - 1 / (2 \cdot a_1 \cdot Y)) / (0.275 + 1.44 \cdot k_z^{0.7})$
j = 1 for a local frame with one simple support outside the ice-strengthened areas
= 2 for a local frame without any simple supports
$a_1 = A_t / A_w$
A_t = minimum shear area of the local frame as given in I2.6.2 [cm^2]
A_w = effective net shear area of the local frame (calculated according to I2.5.7) [cm^2]
$k_w = 1 / (1 + 2 \cdot A_{fn} / A_w)$ with A_{fn} as given in I2.5.8
$k_z = z_p / Z_p$ in general
= 0.0 when the frame is arranged with end bracket
z_p = sum of individual plastic section moduli of flange and shell plate as fitted [cm^3]
$= (b_f \cdot t_{fn}^2 / 4 + b_{eff} \cdot t_{pn}^2 / 4) / 1000$
b_f = flange breadth [mm], see Figure 4
t_{fn} = net flange thickness [mm]
$= t_f - t_c$ (t_c as given in I2.5.7)
t_f = as-built flange thickness [mm], see Figure 4
t_{pn} = the fitted net shell plate thickness [mm] (not to be less than t_{net} as given in I2.4)
b_{eff} = effective width of shell plate flange [mm]
$= 500 \cdot s$

Z_p = net effective plastic section modulus of the local frame (calculated according to I2.5.8) [cm^3]

I2.6.4 The scantlings of the local frame are to meet the structural stability requirements of I2.9.

I2.7 Framing – Longitudinal local frames in side structures

I2.7.1 Longitudinal local frames in side structures are to be dimensioned such that the combined effects of shear and bending do not exceed the plastic strength of the member. The plastic strength is defined by the magnitude of midspan load that causes the development of a plastic collapse mechanism.

I2.7.2 The actual net effective shear area of the frame, A_w, as defined in I2.5.7, is to comply with the following condition: $A_w \geq A_L$, where:

$$A_L = 100^2 \cdot (AF \cdot PPF_s \cdot P_{avg}) \cdot 0.5 \cdot b_1 \cdot a \,/\, (0.577 \cdot \sigma_y) \text{ [cm}^2\text{]}$$

where:

AF = Hull Area Factor from Table 4 or Table 5

PPF_s = Peak Pressure Factor from Table 3

P_{avg} = average pressure within load patch as defined in I2.3.4 [MPa]

$b_1 = k_o \cdot b_2$ [m]

$k_o = 1 - 0.3 / b'$

$b' = b / s$

b = height of design ice load patch as defined in I2.3.3 (i) or I2.3.3 (ii) [m]

s = spacing of longitudinal frames [m]

$b_2 = b \cdot (1 - 0.25 \cdot b')$ [m], if $b' < 2$

$= s$ [m], if $b' \geq 2$

a = effective span of longitudinal local frame as given in I2.5.5 [m]

σ_y = minimum upper yield stress of the material [N/mm^2]

I2.7.3 The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in I2.5.8, is to comply with the following condition: $Z_p \geq Z_{pL}$, where:

$$Z_{pL} = 100^3 \cdot (AF \cdot PPF_s \cdot P_{avg}) \cdot b_1 \cdot a^2 \cdot A_4 \,/\, (8 \cdot \sigma_y) \text{ [cm}^3\text{]}$$

where:

AF, PPF_s, P_{avg}, b_1, a and σ_y are as given in I2.7.2

$A_4 = 1 / (2 + k_{wl} \cdot [(1 - a_4^2)^{0.5} - 1])$

$a_4 = A_L / A_w$

A_L = minimum shear area for longitudinal as given in I2.7.2 [cm^2]

A_w = net effective shear area of longitudinal (calculated according to I2.5.7) [cm^2]

$k_{wl} = 1 / (1 + 2 \cdot A_{fn} / A_w)$ with A_{fn} as given in I2.5.8

I2.7.4 The scantlings of the longitudinals are to meet the structural stability requirements of I2.9.

I2.8 Framing – Web frames and load carrying stringers

I2.8.1 Web frames and load-carrying stringers are to be designed to withstand the ice load patch as defined in I2.3. The load patch is to be applied at locations where the capacity of these members under the combined effects of bending and shear is minimised.

I2.8.2 Web frames and load-carrying stringers are to be dimensioned such that the combined effects of shear and bending do not exceed the limit state(s) defined by the PRS. Where the structural configuration is such that members do not form part of a grillage system, the appropriate peak pressure factor (PPF) from Table 3 is to be used. Special attention is to be paid to the shear capacity in way of lightening holes and cut-outs in way of intersecting members.

I2.8.3 For determination of scantlings of load carrying stringers, web frames supporting local frames, or web frames supporting load carrying stringers forming part of a structural grillage system, appropriate methods as outlined in I2.17 are normally to be used.

I2.8.4 The scantlings of web frames and load-carrying stringers are to meet the structural stability requirements of I2.9.

I.2.9 Framing – Structural stability

I2.9.1 To prevent local buckling in the web, the ratio of web height (h_w) to net web thickness (t_{wn}) of any framing member is not to exceed:

For flat bar sections: $h_w \;/\; t_{wn} \le 282 \;/\; (\sigma_y)^{0.5}$

For bulb, tee and angle sections: $h_w \;/\; t_{wn} \le 805 \;/\; (\sigma_y)^{0.5}$

where:

h_w = web height

t_{wn} = net web thickness

σ_y = minimum upper yield stress of the material [N/mm^2]

I2.9.2 Framing members for which it is not practicable to meet the requirements of I2.9.1 (e.g. load carrying stringers or deep web frames) are required to have their webs effectively stiffened. The scantlings of the web stiffeners are to ensure the structural stability of the framing member. The minimum net web thickness for these framing members is given by:

$$t_{wn} = 2.63 \cdot 10^{-3} \cdot c_1 \cdot \left(\sigma_y/(5.34 + 4 \cdot (c_1/c_2)^2)\right)^{0.5} \text{ [mm]}$$

where:

c_1 = $h_w - 0.8 \cdot h$ [mm]

h_w = web height of stringer / web frame [mm] (see Figure 5)

h = height of framing member penetrating the member under consideration (0 if no such framing member) [mm] (see Figure 5)

c_2 = spacing between supporting structure oriented perpendicular to the member under consideration [mm] (see Figure 5)

σ_y = minimum upper yield stress of the material [N/mm^2]

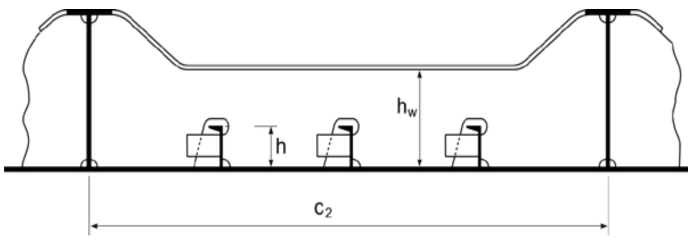


Figure 5 – Parameter definition of web stiffening

I2.9.3 In addition, the following is to be satisfied:

$$t_{wn} \ge 0.35 \cdot t_{pn} \cdot (\sigma_y \;/\; 235)^{0.5}$$

where:

σ_y = minimum upper yield stress of the shell plate in way of the framing member [N/mm^2]
t_{wn} = net thickness of the web [mm]
t_{pn} = net thickness of the shell plate in way of the framing member [mm]

I2.9.4 To prevent local flange buckling of welded profiles, the following are to be satisfied:

(i) The flange width, b_f [mm], is not to be less than five times the net thickness of the web, t_{wn}.
(ii) The flange outstand, b_{out} [mm], is to meet the following requirement:

$$b_{out} \; / \; t_{fn} \leq 155 \; / \; (\sigma_y)^{0.5}$$

where:
t_{fn} = net thickness of flange [mm]
σ_y = minimum upper yield stress of the material [N/mm^2]

I2.10 Plated structures

I2.10.1 Plated structures are those stiffened plate elements in contact with the hull and subject to ice loads. These requirements are applicable to an inboard extent which is the lesser of:
(i) web height of adjacent parallel web frame or stringer; or
(ii) 2.5 times the depth of framing that intersects the plated structure

I2.10.2 The thickness of the plating and the scantlings of attached stiffeners are to be such that the degree of end fixity necessary for the shell framing is ensured.

I2.10.3 The stability of the plated structure is to adequately withstand the ice loads defined in I2.3.

I2.11 Corrosion/-abrasion additions and steel renewal

I2.11.1 Effective protection against corrosion and ice-induced abrasion is recommended for all external surfaces of the shell plating for Polar Class ships.

I2.11.2 The values of corrosion/abrasion additions, ts, to be used in determining the shell plate thickness are listed in Table 6.

I2.11.3 Polar Class ships are to have a minimum corrosion/abrasion addition of t_s = 1.0 mm applied to all internal structures within the ice-strengthened hull areas, including plated members adjacent to the shell, as well as stiffener webs and flanges.

Table 6 – Corrosion/abrasion additions for shell plating

Hull area	ts [mm]					
	With effective protection			Without effective protection		
	PC1 - PC3	PC4 & PC5	PC6 & PC7	PC1 - PC3	PC4 & PC5	PC6 & PC7
Bow; Bow Intermediate Icebelt	3.5	2.5	2.0	7.0	5.0	4.0
Bow Intermediate Lower; Midbody & Stern Icebelt	2.5	2.0	2.0	5.0	4.0	3.0
Midbody & Stern Lower;	2.0	2.0	2.0	4.0	3.0	2.5

I2.11.4 Steel renewal for ice strengthened structures is required when the gauged thickness is less than t_{net} + 0.5 mm.

I2.12 Materials

I2.12.1 Steel grades of plating for hull structures are to be not less than those given in Table 8 based on the as-built thickness, the Polar Class and the material class of structural members according to I2.12.2.

Table 7 – Material classes for structural members

Structural members	Material class
Shell plating within the bow and bow intermediate icebelt hull areas (B, BIi)	II
All weather and sea exposed SECONDARY and PRIMARY, as defined in *PRS Rules Part II - Hull, Table 2.2.1.3-1,* structural members outside 0.4L amidships	I
Plating materials for stem and stern frames, rudder horn, rudder, propeller nozzle, shaft brackets, ice skeg, ice knife and other appendages subject to ice impact loads	II
All inboard framing members attached to the weather and sea-exposed plating, including any contiguous inboard member within 600 mm of the plating	I
Weather-exposed plating and attached framing in cargo holds of ships which by nature of their trade have their cargo hold hatches open during cold weather operations	I
All weather and sea exposed SPECIAL, as defined in *PRS Rules Part II - Hull, Table 2.2.1.3-1*, structural members within 0.2L from FP	II

I2.12.2 Material classes specified in *PRS Rules Part II - Hull, Table 2.2.1.3-1* are applicable to Polar Class ships regardless of the ship's length. In addition, material classes for weather and sea exposed structural members and for members attached to the weather and sea exposed plating are given in Table 7. Where the material classes in Table 7 and those in *PRS Rules Part II – Hull, Table 2.2.1.3-1* differ, the higher material class is to be applied.

I2.12.3 Steel grades for all plating and attached framing of hull structures and appendages situated below the level of 0.3 m below the lower waterline, as shown in Figure 6, are to be obtained from *PRS Rules Part II – Hull, Table 2.2.1.3-5 and Table 2.2.1.3-6* based on the material class for structural members in Table 7 above, regardless of Polar Class.

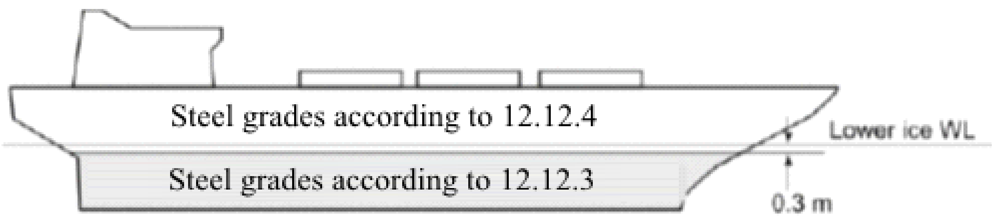


Figure 6 – Steel grade requirements for submerged and weather exposed shell plating

I2.12.4 Steel grades for all weather exposed plating of hull structures and appendages situated above the level of 0.3 m below the lower ice waterline, as shown in Figure 6, are to be not less than given in Table 8.

Table 8 – Steel grades for weather exposed plating[1]

Thickness, t [mm]	Material class I				Material class II				Material class III					
	PC1-5		PC6&7		PC1-5		PC6&7		PC1-3		PC4&5		PC6&7	
	MS	HT	MS	HT	MS	HT	MS	HT	MS	HT	MS	HT	MS	HT
$t \leq 10$	B	AH	B	AH	B	AH	B	AH	E	EH	E	EH	B	AH
$10 < t \leq 15$	B	AH	B	AH	D	DH	B	AH	E	EH	E	EH	D	DH
$15 < t \leq 20$	D	DH	B	AH	D	DH	B	AH	E	EH	E	EH	D	DH
$20 < t \leq 25$	D	DH	B	AH	D	DH	B	AH	E	EH	E	EH	D	DH
$25 < t \leq 30$	D	DH	B	AH	E	EH[2)]	D	DH	E	EH	E	EH	E	EH
$30 < t \leq 35$	D	DH	B	AH	E	EH	D	DH	E	EH	E	EH	E	EH
$35 < t \leq 40$	D	DH	D	DH	E	EH	D	DH	Ø	FH	E	EH	E	EH
$40 < t \leq 45$	E	EH	D	DH	E	EH	D	DH	Ø	FH	E	EH	E	EH
$45 < t \leq 50$	E	EH	D	DH	E	EH	D	DH	Ø	FH	Ø	FH	E	EH

Notes to Table 8:

Includes weather-exposed plating of hull structures and appendages, as well as their outboard framing members, situated above a level of 0.3 m below the lowest ice waterline.

Grades D, DH are allowed for a single strake of side shell plating not more than 1.8 m wide from 0.3 m below the lowest ice waterline.

Ø Not applicable

I2.12.5 Castings are to have specified properties consistent with the expected service temperature for the cast component.

I2.13 Longitudinal strength

I2.13.1 Application

I2.13.1.1 A ramming impact on the bow is the design scenario for the evaluation of the longitudinal strength of the hull.

I2.13.1.2 Intentional ramming is not considered as a design scenario for ships which are designed with vertical or bulbous bows, see I1.1.6. Hence the longitudinal strength requirements given in I2.13 are not to be considered for ships with stem angle γ_{stem} equal to or larger than 80 deg.

I2.13.1.3 Ice loads are only to be combined with still water loads. The combined stresses are to be compared against permissible bending and shear stresses at different locations along the ship's length. In addition, sufficient local buckling strength is also to be verified.

I2.13.2 Design vertical ice force at the bow

I2.13.2.1 The design vertical ice force at the bow, F_{IB}, is to be taken as

$$F_{IB} = \text{minimum}\ (F_{IB,1};\ F_{IB,2})\ [\text{MN}]$$

where:

$F_{IB,1} = 0.534 \cdot K_I^{0.15} \cdot \sin^{0.2}(\gamma_{stem}) \cdot (D_{UI} \cdot K_h)^{0.5} \cdot CF_L$ [MN]

$F_{IB,2} = 1.20 \cdot CF_F$ [MN]

K_I = indentation parameter = K_f / K_h

for the case of a blunt bow form

$$K_f = (2 \cdot C \cdot B^{1-eb} / (1 + e_b))^{0.9} \cdot \tan(\gamma_{stem})^{-0.9\cdot(1+eb)}$$

for the case of wedge bow form (α_{stem} < 80 deg), e_b =1 and the above simplifies to

$K_f = (\tan(\alpha_{stem}) / \tan^2(\gamma_{stem}))^{0.9}$

$K_h = 0.01 \cdot A_{wp}$ [MN/m]

CF_L = Longitudinal Strength Class Factor from Table 1

e_b = bow shape exponent which best describes the waterplane (see Figures 7 and 8)

= 1.0 for a simple wedge bow form

= 0.4 to 0.6 for a spoon bow form

= 0 for a landing craft bow form

An approximate e_b determined by a simple fit is acceptable

γ_{stem} = stem angle to be measured between the horizontal axis and the stem tangent at the upper ice waterline [deg] (buttock angle as per Figure 2 measured on the centreline)

α_{stem} = waterline angle measured in way of the stem at the upper ice waterline (UIWL) [deg] (see Figure 7)

$C = 1 / (2 \cdot (L_B / B)^{eb})$

B_{UI} = moulded breadth corresponding to the upper ice waterline (UIWL) [m]

L_B = bow length used in the equation $y = B_{UI} / 2 \cdot (x/L_B)^{eb}$ [m] (see Figures 7 and 8)

D_{UI} = ship displacement as defined in I2.1.2.2, not to be taken less than 10 [kt]

A_{wp} = waterplane area corresponding to the upper ice waterline (UIWL) [m^2]

CF_F = Flexural Failure Class Factor from Table 1.

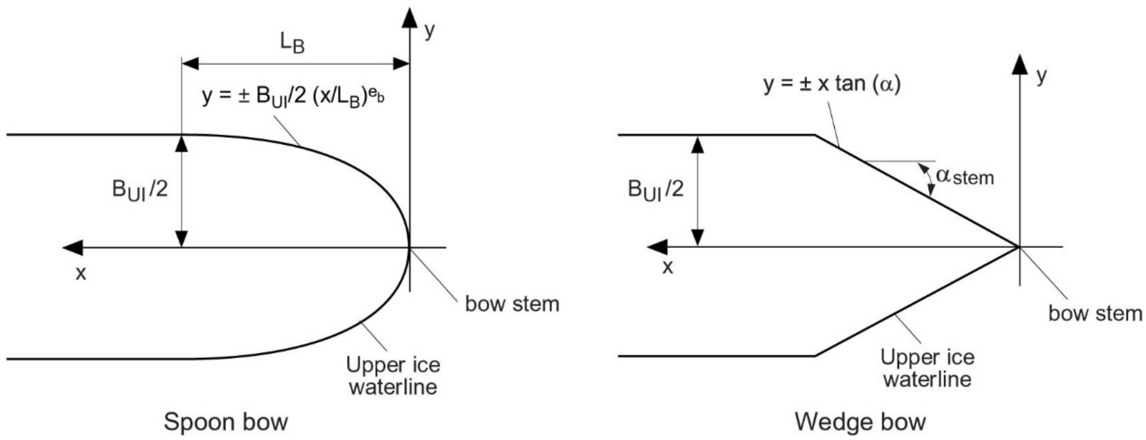


Figure 7 – Bow shape definition

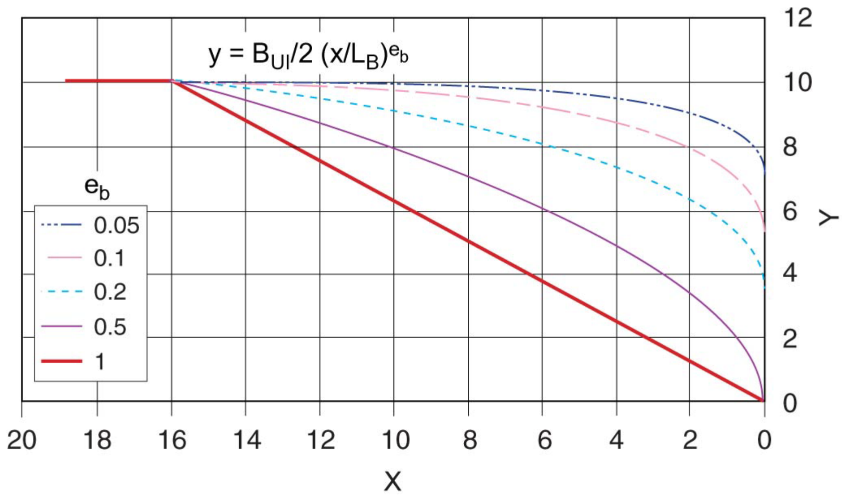


Figure 8 – Illustration of e_b effect on the bow shape for B_{UI} = 20 and L_B =16

I2.13.3 Design vertical shear force

I2.13.3.1 The design vertical ice shear force, F_I, along the hull girder is to be taken as:

$$F_I = C_f \cdot F_{IB} \text{ [MN]}$$

where C_f = longitudinal distribution factor to be taken as follows:

Positive shear force

C_f = 0.0 between the aft end of L_{UI} and 0.6 L_{UI} from aft
C_f = 1.0 between 0.9 L_{UI} from aft and the forward end of L_{UI}

Negative shear force

C_f = 0.0 at the aft end of L_{UI}
C_f = -0.5 between 0.2 L_{UI} and 0.6 L from aft
C_f = 0.0 between 0.8 L_{UI} from aft and the forward end of L_{UI}

Intermediate values are to be determined by linear interpolation

I2.13.3.2 The applied vertical shear stress, τ_a, is to be determined along the hull girder in a similar manner as in *PRS Rules Part II – Hull, par. 13.3.2.8* by substituting the design vertical ice shear force for the design vertical wave shear force.

I2.13.4 Design vertical ice bending moment

I2.13.4.1 The design vertical ice bending moment, M_I, along the hull girder is to be taken as:

$$M_I = 0.1 \cdot C_m \cdot L_{UI} \cdot \sin^{-0.2}(\gamma_{stem}) \cdot F_{IB} \text{ [MNm]}$$

where:
L_{UI} = length as defined in *1.2.2 Part II – Hull* [m]
γ_{stem} is as given in I2.13.2.1
F_{IB} = design vertical ice force at the bow [MN]
C_m = longitudinal distribution factor for design vertical ice bending moment to be taken as follows:

C_m = 0.0 at the aft end of L_{UI}
C_m = 1.0 between 0.5 L_{UI} and 0.7 L_{UI} from aft

C_m = 0.3 at 0.95 L_{UI} from aft
C_m = 0.0 at the forward end of L_{UI}

Intermediate values are to be determined by linear interpolation

I2.13.4.2 The applied vertical bending stress, σ_a, is to be determined along the hull girder in a similar manner as in *PRS Rules Part II – Hull, par. 13.3.2.7*, by substituting the design vertical ice bending moment for the design vertical wave bending moment. The ship still water bending moment is to be taken as the permissible still water bending moment in sagging condition.

I2.13.5 Longitudinal strength criteria

I2.13.5.1 The strength criteria provided in Table 9 are to be satisfied. The design stress is not to exceed the permissible stress.

Table 9 – Longitudinal strength criteria

Failure mode	Applied stress	Permissible stress when $\sigma y / \sigma u \leq 0.7$	Permissible stress when $\sigma y / \sigma u > 0.7$
Tension	σa	$\eta \cdot \sigma y$	$\eta \cdot 0.41 (\sigma u + \sigma y)$
Shear	τa	$\eta \cdot \sigma y / (3)^{0.5}$	$\eta \cdot 0.41 (\sigma_u + \sigma_y) / (3)^{0.5}$
Buckling	σa	σc for plating and for web plating of stiffeners $\sigma c / 1.1$ for stiffeners	
	τa	τc	

where:

σ_a = applied vertical bending stress [N/mm^2]
τ_a = applied vertical shear stress [N/mm^2]
σ_y = minimum upper yield stress of the material [N/mm^2]
σ_u = ultimate tensile strength of material [N/mm^2]
σ_c = critical buckling stress in compression, according to UR S11.5 [N/mm^2]
τ_c = critical buckling stress in shear, according to UR S11.5 [N/mm^2]

η= 0.8

η= 0.6 for ships which are assigned the additional notation "Icebreaker"

I2.14 Stem and stern frames

I2.14.1 The stem and stern frame are to be designed according to the regulations of the PRS. For **PC6/PC7** ships requiring **L1A/L1** equivalency, the stem and stern requirements of the PRS Regulations may need to be additionally considered.

I2.15 Appendages

I2.15.1 All appendages are to be designed to withstand forces appropriate for the location of their attachment to the hull structure or their position within a hull area.

I2.15.2 Load definition and response criteria are to be determined by the PRS.

I2.16 Local details

I2.16.1 For the purpose of transferring ice-induced loads to supporting structure (bending moments and shear forces), local design details are to comply with the requirements of the PRS.

I2.16.2 The loads carried by a member in way of cut-outs are not to cause instability. Where necessary, the structure is to be stiffened.

I2.17 Direct calculations

I2.17.1 Direct calculations are not to be utilised as an alternative to the analytical procedures prescribed for the shell plating and local frame requirements given in I2.4, I2.6, and I2.7.

I2.17.2 Direct calculations are to be used for load carrying stringers and web frames forming part of a grillage system.

I2.17.3 Where direct calculation is used to check the strength of structural systems, the load patch specified in I2.3 is to be applied., without being combined with any other loads. The load patch is to be applied at locations where the capacity of these members under the combined effects of bending and shear is minimised. Special attention is to be paid to the shear capacity in way of lightening holes and cut-outs in way of intersecting members.

I2.17.4 The strength evaluation of web frames and stringers may be performed based on linear or non-linear analysis. Recognized structural idealisation and calculation methods are to be applied, but the detailed requirements are to be specified by the PRS. In the strength evaluation, the guidance given in I2.17.5 and I2.17.6 may generally be considered.

I2.17.5 If the structure is evaluated based on linear calculation methods, the following are to be considered.

(1) Web plates and flange elements in compression and shear to fulfil relevant buckling criteria as specified by the PRS;
(2) Nominal shear stresses in member web plates to be less than $\sigma_y / \sqrt{3}$;
(3) Nominal von Mises stresses in member flanges to be less than 1.15 σ_y.

I2.17.6 If the structure is evaluated based on non-linear calculation methods, the following are to be considered:

(1) The analysis is to reliably capture buckling and plastic deformation of the structure;
(2) The acceptance criteria are to ensure a suitable margin against fracture and major buckling and yielding causing significant loss of stiffness;
(3) Permanent lateral and out-of plane deformation of considered member are to be minor relative to the relevant structural dimensions;
(4) Detailed acceptance criteria to be decided by the PRS.

I2.18 Welding

I2.18.1 All welding within ice-strengthened areas is to be of the double continuous type.

I2.18.2 Continuity of strength is to be ensured at all structural connections.

I3 MACHINERY REQUIREMENTS FOR POLAR CLASS SHIPS

I3.1 Application

The contents of this Chapter apply to main propulsion, steering gear, emergency and essential auxiliary systems essential for the safety of the ship and the survivability of the crew.

I3.2 General

2.1 The following drawings and particulars are to be submitted.

2.1.1 Details of the intended environmental operational conditions and the required ice strengthening for the machinery, if different from ship's ice class

2.1.2 Detailed drawings and descriptions of the main propulsion, steering, emergency and auxiliary machinery and information on the essential main propulsion load control functions. The descriptions are to include operational limitations.

2.1.3 Description detailing where main, emergency and auxiliary systems are located and how they are protected to prevent problems from freezing, ice and snow accumulation and evidence of their capability to operate in the intended environmental conditions

2.1.4 Calculations and documentation indicating compliance with the requirements of this chapter

I3.2.2 System Design

2.2.1 Systems subject to damage by freezing, shall be drainable.

2.2.2 Vessels classed PC1 to PC5 inclusive shall have means provided to ensure sufficient vessel operation in the case of propeller damage including the Controllable Pitch (CP) mechanism. Sufficient vessel operation means that the vessel should be able to reach safe haven (safe location) where repairs can be undertaken. This may be achieved either by a temporary repair at sea, or by towing, assuming assistance is available. This would lead however to a condition of approval.

2.2.3 Means shall be provided to free a stuck propeller by turning it in reverse direction. This shall also be possible for a propulsion plant intended for unidirectional rotation.

2.2.4 The propeller shall be fully submerged at the ships LIWL.

I3.3 Materials

Materials shall be of an approved ductile material. Ferritic nodular cast iron may be used for parts other than bolts. For nodular cast iron an averaged impact energy value of 10 J at testing temperature is regarded as equivalent to the Charpy V test requirements defined below.

I3.3.1 Materials exposed to sea water

Materials exposed to sea water, such as propeller blades, propeller hubs and cast thruster bodies shall have an elongation not less than 15% on a test specimen according to UR W2.

Charpy V-notch impact testing is to be carried out for materials other than bronze and austenitic steel. The tests shall be carried out on three specimens at minus 10 ºC, and the average energy value is to be not less than 20 J. However, Charpy V impact test requirements of UR W7 or UR W27 as applicable for ships with ice class notation, shall also be applied to ships covered by this UR.

I3.3.2 Materials exposed to sea water temperature

Charpy V-notch impact testing is to be carried out for materials other than bronze and austenitic steel. The tests shall be carried out on three specimens at minus 10 ºC, and the average energy value is to be not less than 20 J. However, the Charpy V impact test requirements of UR W7 as applicable for ships with ice class notation, shall also be applied to ships covered by this UR.

This requirement applies to components such as but not limited to blade bolts, CP-mechanisms, shaft bolts, propeller shaft, strut-pod connecting bolts, etc. This requirement does not apply to surface hardened components, such as bearings and gear teeth or sea water cooling lines (heat exchangers, pipes, valves, fittings etc.). For a definition of structural boundaries exposed to sea water temperature see UR I2 Figure 6.

I3.3.3 Material exposed to low air temperature

Materials of exposed machinery and foundations shall be manufactured from steel or other approved ductile material. An average impact energy value of 20 J taken from three Charpy V tests is to be obtained at 10 ºC below the lowest design temperature. Charpy V impact tests are not required for bronze and austenitic steel.

This requirement does not apply to surface hardened components, such as bearings and gear teeth. For a definition of structural boundaries exposed to air temperature see UR I2 Figure 6.

4 Definitions

4.1 Definition of Symbols

Table 1: Definition of symbols

Symbol	Unit	Definition
c	m	chord length of blade section
$c_{0.7}$	m	chord length of blade section at $0.7R$ propeller radius
CP	-	controllable pitch
D	m	propeller diameter
d	m	external diameter of propeller hub (at propeller plane)
d_{pin}	mm	diameter of shear pin
D_{limit}	m	limit value for propeller diameter
EAR		expanded blade area ratio
F_b	kN	maximum backward blade force for the ship's service life (negative sign)
F_{ex}	kN	ultimate blade load resulting from blade failure through plastic bending
F_f	kN	maximum forward blade force for the ship's service life (positive sign)
F_{ice}	kN	ice load
$(F_{ice})_{max}$	kN	maximum ice load for the ship's service life
FP	-	fixed pitch
h_0	m	depth of the propeller centreline from lower ice waterline ($LIWL$)
(H_{ice})	m	Ice block dimension for propeller load definition
I	kgm²	equivalent mass moment of inertia of all parts on engine side of component under consideration
I_t	kgm²	equivalent mass moment of inertia of the whole propulsion system
k	-	shape parameter for Weibull distribution
$LIWL$	m	lower ice waterline
m	-	slope for S-N curve in log/log scale
M_{BL}	kNm	blade bending moment
MCR	-	maximum continuous rating
N	-	number of ice load cycles
n	rev/s	propeller rotational speed
n_n	rev/s	nominal propeller rotational speed at MCR in free running condition
N_{class}	-	reference number of ice impacts per propeller revolution per ice class
N_{ice}	-	total number of ice load cycles on propeller blade for the ship's service life
N_R	-	reference number of ice load cycles for equivalent fatigue stress (10^8 cycles)
N_Q	-	number of propeller revolutions during a milling sequence
$P_{0.7}$	m	propeller pitch at $0.7R$ radius
$P_{0.7n}$	m	propeller pitch at $0.7R$ radius at MCR in free running condition
$P_{0.7b}$	m	propeller pitch at $0.7R$ radius at MCR in bollard condition
PCD	m	pitch circle diameter
$Q(\varphi)$	kNm	Torque
Q_{Amax}	kNm	maximum response torque amplitude as a simulation result

Q_{emax}	kNm	maximum engine torque
$Q_F(\varphi)$	kNm	Ice torque excitation for frequency domain calculations
Q_{fr}	kNm	friction torque in pitching mechanism; reduction of spindle torque
Q_{max}	kNm	maximum torque on the propeller resulting from propeller/ice interaction
Q_{motor}	kNm	electric motor peak torque
Q_n	kNm	nominal torque at MCR in free running condition
$Q_r(t)$	kNm	response torque along the propeller shaft line
Q_{peak}	kNm	maximum of the response torque $Q_r(t)$
Q_{smax}	kNm	maximum spindle torque of the blade for the ship's service life
Q_{sex}	kNm	extreme spindle torque corresponding to the blade failure load F_{ex}
Q_{vib}	kNm	Vibratory torque at considered component, taken from frequency domain open water TVC
R	m	propeller radius
S	-	Safety factor
S_{fat}	-	Safety factor for fatigue
S_{ice}	-	Ice strength index for blade ice force
r	m	blade section radius
T	kN	Hydrodynamic propeller thrust in bollard condition
T_b	kN	maximum backward propeller ice thrust for the ship's service life
T_f	kN	maximum forward propeller ice thrust for the ship's service life
T_n	kN	propeller thrust at MCR in free running condition
T_r	kN	maximum response thrust along the shaft line
T_{kmax}	kNm	maximum torque capacity of flexible coupling
T_{kmax2}	kNm	T_{kmax} at $N = 1$ load cycle
T_{max1}	kNm	T_{kmax} at $N = 5 \times 10^4$ load cycles
T_{kv}	kNm	vibratory torque amplitude at $N = 10^6$ load cycles
ΔT_{kmax}	kNm	maximum range of T_{kmax} at $N = 5 \times 10^4$ load cycles
t	m	maximum blade section thickness
z	-	number of propeller blades
z_{pin}	-	number of shear pins
α_i	deg	duration of propeller blade/ice interaction expressed in rotation angle
γ_ε	-	the reduction factor for fatigue; scatter and test specimen size effect
γ_v	-	the reduction factor for fatigue; variable amplitude loading effect
γ_m	-	the reduction factor for fatigue; mean stress effect
ρ	-	a reduction factor for fatigue correlating the maximum stress amplitude to the equivalent fatigue stress for 10^8 stress cycles
$\sigma_{0.2}$	MPa	proof yield strength (at 0.2% plastic strain) of material
σ_{exp}	MPa	mean fatigue strength of blade material at 10^8 cycles to failure in sea water
σ_{fat}	MPa	equivalent fatigue ice load stress amplitude for 10^8 stress cycles
σ_{fl}	MPa	characteristic fatigue strength for blade material
σ_{ref1}	MPa	reference stress $\sigma_{ref1} = 0.6\,\sigma_{0.2} + 0.4\,\sigma_u$
σ_{ref2}	MPa	reference stress $\sigma_{ref2} = 0.7\,\sigma_u$ or $\sigma_{ref2} = 0.6\,\sigma_{0.2} + 0.4\,\sigma_u$ whichever is less
σ_{st}	MPa	maximum stress resulting from F_b or F_f
σ_u	MPa	ultimate tensile strength of blade material

$(\sigma_{ice})_{bmax}$	MPa	principal stress caused by the maximum backward propeller ice load
$(\sigma_{ice})_{fmax}$	MPa	principal stress caused by the maximum forward propeller ice load
$(\sigma_{ice})_{Amax}$	MPa	maximum ice load stress amplitude at the considered location on the blade
σ_{mean}	MPa	mean stress
$(\sigma_{ice})_A(N)$	MPa	blade stress amplitude distribution

4.2 Definition of Loads

Table 2: Definitions of loads

	Definition	Use of the load in design process
F_b	The maximum lifetime backward force on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to 0.7R chord line. See Figure 1.	Design force for strength calculation of the propeller blade.
F_f	The maximum lifetime forward force on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to 0.7R chord line.	Design force for calculation of strength of the propeller blade.
Q_{smax}	The maximum lifetime spindle torque on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade.	In designing the propeller strength, the spindle torque is automatically taken into account because the propeller load is acting on the blade as distributed pressure on the leading edge or tip area.
T_b	The maximum lifetime thrust on propeller (all blades) resulting from propeller/ice interaction. The direction of the thrust is the propeller shaft direction and the force is opposite to the hydrodynamic thrust.	Is used for estimation of the response thrust T_r. T_b can be used as an estimate of excitation for axial vibration calculations. However, axial vibration calculations are not required in the rules.
T_f	The maximum lifetime thrust on propeller (all blades) resulting from propeller/ice interaction. The direction of the thrust is the propeller shaft direction acting in the direction of hydrodynamic thrust.	Is used for estimation of the response thrust T_r. T_f can be used as an estimate of excitation for axial vibration calculations. However, axial vibration calculations are not required in the rules.
Q_{max}	The maximum ice-induced torque resulting from propeller/ice interaction on one propeller blade, including hydrodynamic loads on that blade.	Is used for estimation of the response torque Q_r along the propulsion shaft line and as excitation for torsional vibration calculations.

F_{ex}	Ultimate blade load resulting from blade loss through plastic bending. The force that is needed to cause total failure of the blade so that plastic hinge is caused to the root area. The force is acting on 0.8R.	Blade failure load is used to dimension the blade bolts, pitch control mechanism, propeller shaft, propeller shaft bearing and trust bearing. The objective is to guarantee that total propeller blade failure should not cause damage to other components.
Q_{sex}	Maximum spindle torque resulting from blade failure load	Is used to ensure pyramid strength principle for the pitching mechanism
Q_r	Maximum response torque along the propeller shaft line, taking into account the dynamic behaviour of the shaft line for ice excitation (torsional vibration) and hydrodynamic mean torque on propeller.	Design torque for propeller shaft line components.
T_r	Maximum response thrust along shaft line, taking into account the dynamic behaviour of the shaft line for ice excitation (axial vibration) and hydrodynamic mean thrust on propeller.	Design thrust for propeller shaft line components.

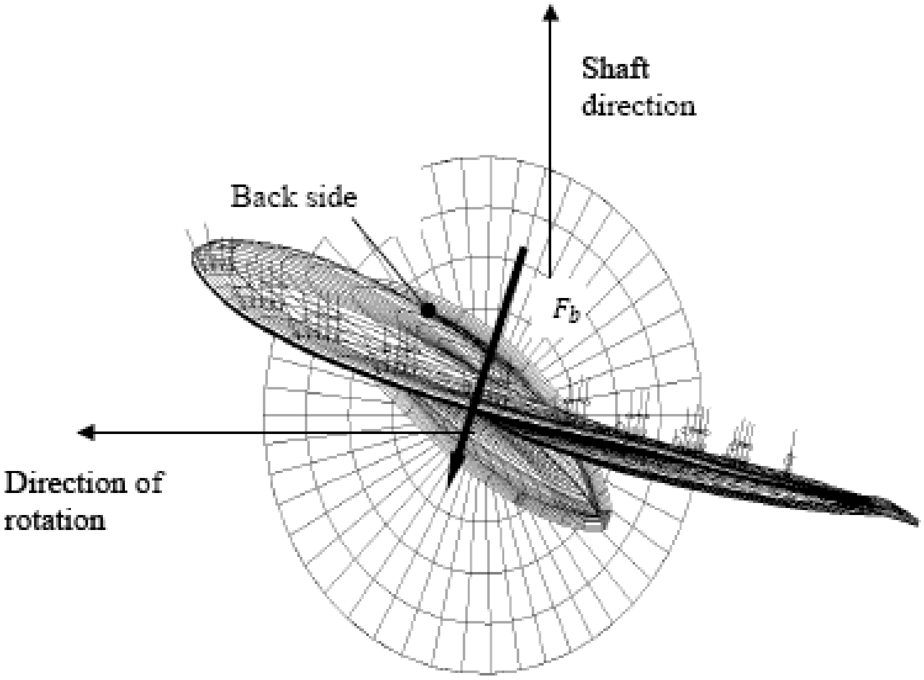


Figure 1 : Direction of the backward blade force resultant taken perpendicular to the chord line at radius 0.7R. Ice contact pressure at leading edge is shown with small arrows.

5 Design Ice Loads

5.1 General

These Rules cover open and ducted type propellers situated at the stern of a vessel having controllable pitch or fixed pitch blades. Ice loads on bow-mounted propellers shall receive special consideration at the discretion of each classification society. The given loads are expected, single occurrence, maximum values for the whole ship's service life for normal operational conditions, including loads resulting from directional change of rotation where applicable. These loads do not cover off-design operational conditions, for example when a stopped propeller is dragged through ice. These Rules also cover loads due to propeller ice interaction for azimuthing and fixed thrusters with geared transmission or an integrated electric motor ("geared and podded propulsors"). However, the load models of the regulations do not include propeller/ice interaction loads when ice enters the propeller of a turned azimuthing thruster from the side (radially) or loads when ice blocks hit on the propeller hub of a pulling propeller. Ice loads resulting from ice impacts on the body of thrusters shall be estimated on a case by case basis, however are not included within the following section.

The loads given in section 5.3 are total loads including ice-induced loads and hydrodynamic loads (unless otherwise stated) during ice interaction and are to be applied separately (unless otherwise stated) and are intended for component strength calculations only.

F_b is the maximum force experienced during the lifetime of the ship that bends a propeller blade backwards when the propeller mills an ice block while rotating ahead. F_f is the maximum force experienced during the lifetime of the ship that bends a propeller blade forwards when the propeller mills an ice block while rotating ahead. F_b and F_f originate from different propeller/ice interaction phenomena, which do not act simultaneously. Hence they are to be applied separately.

I3.4.2 Ice Class Factors

The dimensions of the considered design ice block are $H_{ice} \times 2H_{ice} \times 3H_{ice}$. The design ice block and ice strength index (S_{ice}) are used for the estimation of propeller ice loads. Both H_{ice} and S_{ice} are defined for each Ice class in Table 3 below.

Table 3: Design Ice Class Factors

Ice Class	H_{ice} [m]	S_{ice} [-]
PC1	4.0	1.2
PC2	3.5	1.1
PC3	3.0	1.1
PC4	2.5	1.1
PC5	2.0	1.1
PC6	1.75	1
PC7	1.5	1

5.3 Propeller Ice Interaction Loads

5.3.1 Maximum backward blade force F_b for open propellers

when $D < D_{limit}$:

$$F_b = 27 \cdot S_{ice} \cdot (n \cdot D)^{0.7} \cdot \left(\frac{EAR}{Z}\right)^{0.3} \cdot D^2 \quad \text{[kN]} \quad \text{[Equation 1]}$$

when $D \geq D_{limit}$:

$$F_b = 23 \cdot S_{ice} \cdot (n \cdot D)^{0.7} \cdot \left(\frac{EAR}{Z}\right)^{0.3} \cdot (H_{ice})^{1.4} \cdot D \quad \text{[kN]} \quad \text{[Equation 2]}$$

where:

$$D_{limit} = 0.85 \cdot (H_{ice})^{1.4} \quad \text{[m]} \quad \text{[Equation 3]}$$

Here n is the nominal rotational speed at MCR in the free running open water condition for CP-propellers and 85% of the nominal rotational speed (at MCR free running condition) for a FP-propeller (regardless driving engine type) [rps].

For vessels with the additional notation Icebreaker, the above stated backward blade force F_b shall be multiplied by a factor of 1.1.

5.3.2 Maximum forward blade force F_f for open propellers

when $D < D_{limit}$:

$$F_f = 250 \cdot \left(\frac{EAR}{Z}\right) \cdot D^2 \quad \text{[kN]} \quad \text{[Equation 4]}$$

when $D \geq D_{limit}$:

$$F_f = 500 \cdot \left(\frac{1}{1-\frac{d}{D}}\right) \cdot H_{ice} \cdot \left(\frac{EAR}{Z}\right) \cdot D \quad \text{[kN]} \quad \text{[Equation 5]}$$

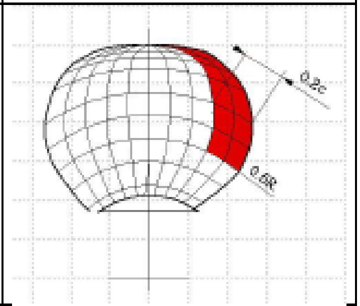
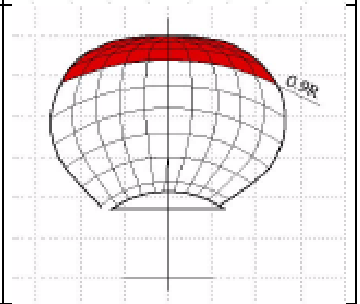
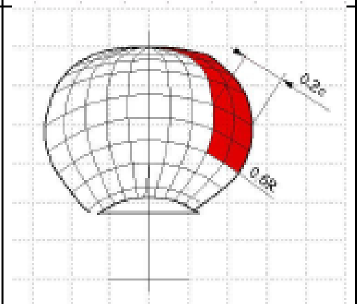
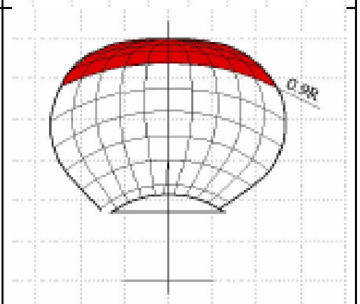
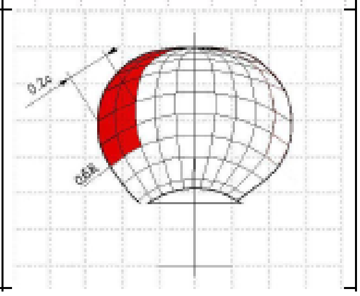
where:

$$D_{limit} = \left(\frac{2}{1-\frac{d}{D}}\right) \cdot H_{ice} \quad \text{[m]} \quad \text{[Equation 6]}$$

5.3.3 Loaded area on the blade for open propellers

Load cases 1-4 shall be covered, as given in Table 4, for CP and FP propellers. In order to obtain blade ice loads for a reversing propeller, load case 5 shall also be covered for propellers where reversing is possible.

Table 4: Loaded areas and load case definition for open propellers

	Force	Loaded area	Right-handed propeller blade seen from behind
Load case 1	F_b	Uniform pressure applied on the back of the blade (suction side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length.	
Load case 2	50% of F_b	Uniform pressure applied on the back of the blade (suction side) on the propeller tip area outside 0.9R radius.	
Load case 3	F_f	Uniform pressure applied on the blade face (pressure side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length.	
Load case 4	50% of F_f	Uniform pressure applied on propeller face (pressure side) on the propeller tip area outside 0.9R radius.	
Load case 5	60% of F_f or 60% of F_b, whichever is greater	Uniform pressure applied on propeller face (pressure side) to an area from 0.6R to the tip and from the trailing edge to 0.2 times the chord length	

5.3.4 Maximum backward blade ice force F_b for ducted propellers

when $D < D_{limit}$:

$$F_b = 9.5 \cdot S_{ice} \cdot (n \cdot D)^{0.7} \cdot \left(\frac{EAR}{Z}\right)^{0.3} \cdot D^2 \qquad \text{[kN]} \qquad \text{[Equation 7]}$$

when $D \geq D_{limit}$:

$$F_b = 66 \cdot S_{ice} \cdot (n \cdot D)^{0.7} \cdot \left(\frac{EAR}{Z}\right)^{0.3} \cdot (H_{ice})^{1.4} \cdot D^{0.6} \qquad \text{[kN]} \qquad \text{[Equation 8]}$$

where:

$$D_{limit} = 4 \cdot H_{ice} \qquad \text{[m]} \qquad \text{[Equation 9]}$$

n shall be taken as in 5.3.1

For vessels with the additional notation Icebreaker, the above stated backward blade force F_b shall be multiplied by a factor 1.1.

5.3.5 Maximum forward blade ice force F_f for ducted propellers

when $D \leq D_{limit}$:

$$F_f = 250 \cdot \left(\frac{EAR}{Z}\right) \cdot D^2 \qquad \text{[kN]} \qquad \text{[Equation 10]}$$

when $D > D_{limit}$:

$$F_f = 500 \cdot \left(\frac{EAR}{Z}\right) \cdot D \cdot \frac{1}{\left(1-\frac{d}{D}\right)} \cdot H_{ice} \qquad \text{[kN]} \qquad \text{[Equation 11]}$$

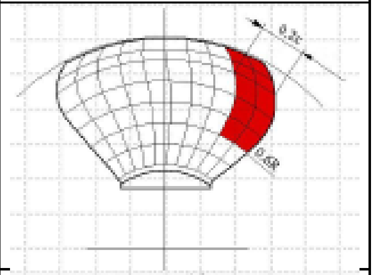
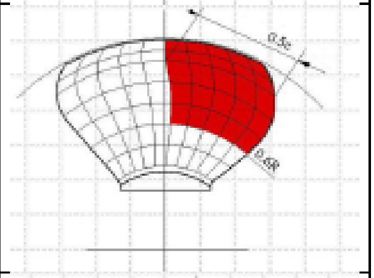
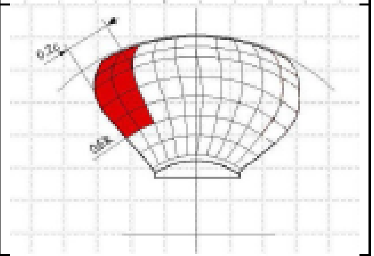
where:

$$D_{limit} = \frac{2}{\left(1-\frac{d}{D}\right)} \cdot H_{ice} \qquad \text{[m]} \qquad \text{[Equation 12]}$$

5.3.6 Loaded area on the blade for ducted propellers

Load cases 1 and 3 shall be covered as given in Table 5 for all propellers. In order to obtain blade ice loads for a reversing propeller, load case 5 shall also be covered for propellers, where reversing is possible.

Table 5: Loaded areas and load case definition for ducted propellers

	Force	Loaded area	Right handed propeller blade seen from behind
Load case 1	F_b	Uniform pressure applied on the back of the blade (suction side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length.	
Load case 3	F_f	Uniform pressure applied on the blade face (pressure side) to an area from 0.6R to the tip and from the leading edge to 0.5 times the chord length.	
Load case 5	60% of F_f or 60% of F_b, whichever is greater	Uniform pressure applied on propeller face (pressure side) to an area from 0.6R to the tip and from the trailing edge to 0.2 times the chord length.	

5.3.7 Maximum blade spindle torque Q_{smax} for open and ducted propellers

The spindle torque Q_{smax} around the axis of the blade fitting shall be determined both for the maximum backward blade force F_b and forward blade force F_f, which are applied as per Table 4 and Table 5. If the above method gives a value which is less than the default value given by the formula below, the default value shall be used.

Default value $Q_{smax} = 0.25 \cdot F \cdot c_{0.7}$ [kNm] [Equation 13]

where:

F is taken as either F_b or F_f, whichever has the greater absolute value.

The blade failure spindle torque Q_{sex} is defined under 5.4.

5.3.8 Load distributions (spectra) for blade loads

The Weibull-type distribution (probability that F_{ice} exceeds $(F_{ice})_{max}$), as given in Figure 2 is used for the fatigue design of the blade.

$$P\left(\frac{F_{ice}}{(F_{ice})_{max}} \geq \frac{F}{(F_{ice})_{max}}\right) = e^{\left(-\left(\frac{F}{(F_{ice})_{max}}\right)^{k} \cdot \ln(N_{ice})\right)} \qquad \text{[Equation 14]}$$

where:

k = shape parameter of the spectrum

N_{ice} = number of load cycles in the spectrum, see 5.3.9

F_{ice} = random variable for ice loads on the blade, $0 \leq F_{ice} \leq (F_{ice})_{max}$.

This results in a blade stress amplitude distribution

$$(\sigma_{ice})_A(N) = (\sigma_{ice})_{Amax} \cdot \left(1 - \frac{\log(N)}{\log(N_{ice})}\right)^{\frac{1}{k}} \qquad \text{[Equation 15]}$$

where:

$$(\sigma_{ice})_{Amax} = \frac{(\sigma_{ice})_{fmax} - (\sigma_{ice})_{bmax}}{2} \qquad \text{[Equation 16]}$$

The shape parameter k = 0.75 shall be used for the ice force distribution of an open propeller and the shape parameter k = 1.0 for that of a ducted propeller blade.

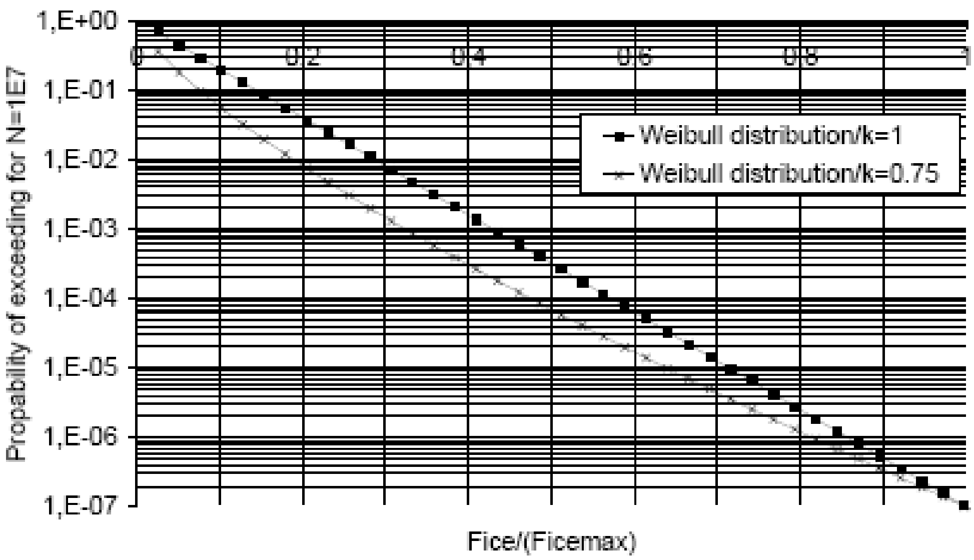


Figure 2: The Weibull-type distribution (probability that F_{ice} exceeds $(F_{ice})_{max}$) that is used for fatigue design.

5.3.9 Number of ice loads

Number of load cycles N_{ice} in the load spectrum per blade is to be determined according to the formula:

$$N_{ice} = k_1 \cdot k_2 \cdot N_{class} \cdot n \qquad \text{[Equation 17]}$$

where:

N_{class} = reference number of impacts per propeller revolution for each ice class (Table 6)

Table 6: Reference number of impacts

Ice Class	PC1	PC2	PC3	PC4	PC5	PC6	PC7
N_{class}	21 x 10^6	17 x 10^6	15 x 10^6	13 x 10^6	11 x 10^6	9 x 10^6	6 x 10^6

k_1 = 1 for centre propeller
= 2 for wing propeller
= 3 for pulling propeller (wing and centre)

k_2 = 0.8 - f when f < 0
= 0.8 - 0.4·f when 0 ≤ f ≤ 1
= 0.6 - 0.2·f when 1< f ≤ 2.5
= 0.1 when f > 2.5

where the immersion function f is:

$$f = \frac{h_0 - H_{ice}}{D/2} \qquad \text{[Equation 18]}$$

If h_0 is not known, $h_o = D/2$.

For vessels with the additional notation Icebreaker, the above stated number of load cycles N_{ice} shall be multiplied by a factor of 3.

For components that are subject to loads resulting from propeller/ice interaction with all the propeller blades, the number of load cycles (N_{ice}) is to be multiplied by the number of propeller blades (Z).

5.4 Blade Failure Load for both Open and Ducted Propellers

5.4.1 Bending Force, F_{ex}

The minimum load required resulting in blade failure through plastic bending. This shall be calculated iteratively along the radius of the blade from blade root to 0.5R using below Equation 19 with the ultimate load assumed to be acting at 0.8R in the weakest direction.

The blade failure load is:

$$F_{ex} = \frac{0.3 \cdot c \cdot t^2 \cdot \sigma_{ref1}}{0.8 \cdot D - 2 \cdot r} \cdot 10^3$$ [kN] [Equation 19]

where:

$$\sigma_{ref1} = 0.6 \cdot \sigma_{0.2} + 0.4 \cdot \sigma_u$$ [MPa]

σ_u (minimum ultimate tensile strength to be specified on the drawing) and $\sigma_{0.2}$ (minimum yield or 0.2% proof strength to be specified on the drawing) are representative values for the blade material

c, t and r are respectively the actual chord length, maximum thickness and radius of the cylindrical root section of the blade, which is the weakest section outside the root fillet located typically at the termination of the fillet into the blade profile.

The classification society may approve alternative means of failure load calculation by means of an appropriate stress analysis reflecting the non-linear plastic material behaviour of the actual blade. A blade is regarded as having failed, if the tip is bent by more than 10% of the propeller diameter.

5.4.2 Spindle Torque, Q_{sex}

The maximum spindle torque due to a blade failure load acting at 0.8R shall be determined. The force that causes blade failure typically reduces when moving from the propeller centre towards the leading and trailing edges. At a certain distance from the blade centre of rotation the maximum spindle torque will occur. This maximum spindle torque shall be defined by an appropriate stress analysis or using equation 20 below.

$$Q_{sex} = max(c_{LE0.8}; 0.8 \cdot c_{TE0.8}) \cdot C_{spex} \cdot F_{ex}$$ [kNm] [Equation 20]

where :

$$C_{spex} = C_{sp} \cdot C_{fex} = 0.7 \cdot \left(1 - \left(4 \cdot \frac{EAR}{Z}\right)^3\right)$$ [-] [Equation 21]

C_{sp} is non-dimensional parameter taking into account the spindle arm

C_{fex} is non-dimensional parameter taking into account the reduction of blade failure force at the location of maximum spindle torque.

If C_{spex} is below 0.3, a value of 0.3 shall be used for C_{spex}.

$c_{LE0.8}$c is the leading edge portion of the chord length at 0.8R

$c_{TE0.8}$ is the trailing edge portion of the chord length at 0.8R

The figure below illustrates the spindle torque values due to blade failure loads across the whole chord length.

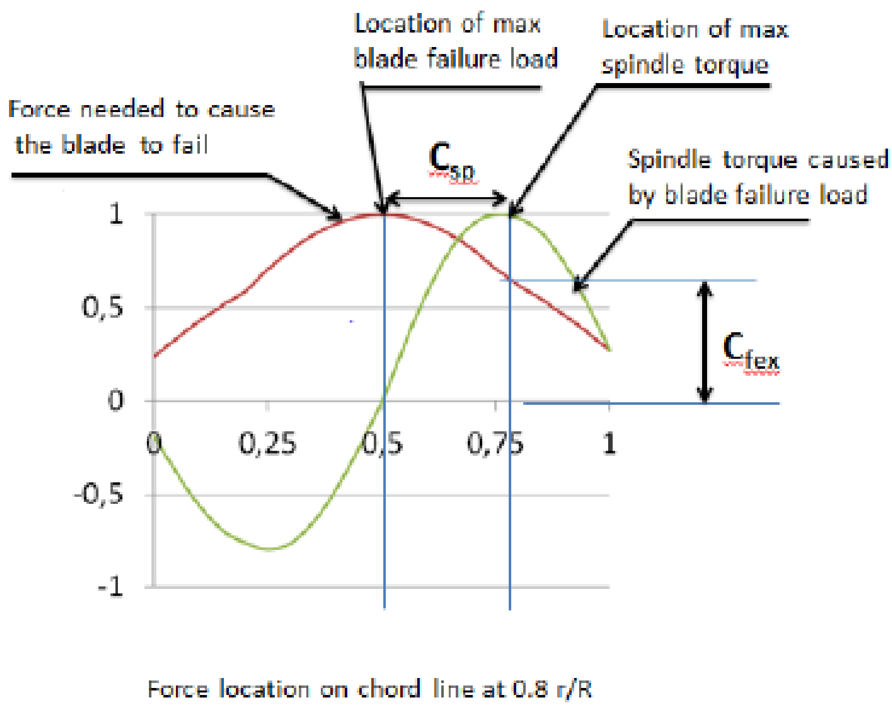


Figure 3: Schematic figure showing blade failure load and related spindle torque when the force acts at different location on the chord line at radius 0.8R.

5.5 Axial design loads acting on open and ducted propellers

5.5.1 Maximum ice thrust on propeller T_f and T_b acting on open and ducted propellers

The maximum forward and backward ice thrusts are given by the following formula:

$T_f = 1.1 \cdot F_f$ [kN] [Equation 22]

$T_b = 1.1 \cdot F_b$ [kN] [Equation 23]

However, the load models within this UR do not include propeller/ice interaction loads where an ice block hits the propeller hub of a pulling propeller.

5.5.2 Design thrust along the propulsion shaft line for open and ducted propellers

The design thrust along the propeller shaft line is to be calculated with the formulae below. The greater value of the forward and backward directional load shall be taken as the design load for both directions. The factors 2.2 and 1.5 take into account the dynamic magnification resulting from axial vibration.

In a forward direction

$T_r = T + 2.2 \cdot T_f$ [kN] [Equation 24]

In a backward direction

$T_r = 1.5 \cdot T_b$ [kN] [Equation 25]

If the hydrodynamic bollard thrust, T, is not known, T is to be taken as follows:

Table 7: Guidance for bollard thrust values

Propeller type	T
CP propellers (open)	1.25 T_n
CP propellers (ducted)	1.1 T_n
FP propellers driven by turbine or electric motor	T_n
FP propellers driven by diesel engine (open)	0.85 T_n
FP propellers driven by diesel engine (ducted)	0.75 T_n

Here, T_n is the nominal propeller thrust at MCR in the free running open water condition.

For pulling type propellers ice interaction loads on propeller hub must be considered in addition to the above. These will be specially considered by the Classification Society.

5.6 Torsional design loads acting on open and ducted propellers

5.6.1 Design ice torque on propeller Q_{max} for open propellers

Q_{max} is the maximum torque on a propeller resulting from ice/propeller interaction.

when $D < D_{limit}$:

$$Q_{max} = k_{open} \cdot \left(1 - \frac{d}{D}\right) \cdot \left(\frac{P_{0.7}}{D}\right)^{0.16} \cdot (n \cdot D)^{0.17} \cdot D^3 \quad \text{[kNm]} \qquad \text{[Equation 26]}$$

where:

k_{open} = 14.7 for PC1 – PC5; and

k_{open} = 10.9 for PC6 – PC7

when $D \geq D_{limit}$:

$$Q_{max} = 1.9 \cdot k_{open} \cdot \left(1 - \frac{d}{D}\right) \cdot (H_{ice})^{1.1} \cdot \left(\frac{P_{0.7}}{D}\right)^{0.16} \cdot (n \cdot D)^{0.17} \cdot D^{1.9} \quad \text{[kNm]} \qquad \text{[Equation 27]}$$

where:

$$D_{limit} = 1.8 \cdot H_{ice} \quad \text{[m]} \qquad \text{[Equation 28]}$$

n is the rotational propeller speed in rev/s in bollard condition. If not known, n is to be taken as follows:

Table 8: Guidance for rotational speeds to calculate torsional loads

Propeller type	Rotational speed n
CP propellers	n_n
FP propellers driven by turbine or electric motor	n_n
FP propellers driven by diesel engine	0.85 n_n

For CP propellers, the propeller pitch $P_{0.7}$ shall correspond to MCR in bollard condition. If not known, $P_{0.7}$ is to be taken as $0.7 \cdot P_{0.7n}$, where $P_{0.7n}$ is the propeller pitch at MCR in free running condition.

5.6.2 Design ice torque on propeller Q_{max} for ducted propellers

when $D < D_{limit}$:

$$Q_{max} = k_{ducted} \cdot \left(1 - \frac{d}{D}\right) \cdot \left(\frac{P_{0.7}}{D}\right)^{0.16} \cdot (n \cdot D)^{0.17} \cdot D^3 \quad \text{[kNm]} \qquad \text{[Equation 29]}$$

where:

k_{ducted} = 10.4 for PC1 – PC5; and

k_{ducted} = 7.7 for PC6 – PC7

when $D \geq D_{limit}$:

$$Q_{max} = 1.9 \cdot k_{ducted} \cdot \left(1 - \frac{d}{D}\right) \cdot (H_{ice})^{1.1} \cdot \left(\frac{P_{0.7}}{D}\right)^{0.16} \cdot (nD)^{0.17} \cdot D^{1.9} \ \text{[kNm]} \qquad \text{[Equation 30]}$$

where:

$$D_{limit} = 1.8 \cdot H_{ice} \quad \text{[m]} \qquad \text{[Equation 31]}$$

n shall be taken as in 5.6.1.

For CP propellers, the propeller pitch $P_{0.7}$ shall correspond to MCR in bollard condition. If not known, $P_{0.7}$ is to be taken as $0.7 \cdot P_{0.7n}$, where $P_{0.7n}$ is the propeller pitch at MCR in free running condition.

5.6.3 Ice torque excitation for open and ducted propellers

The given excitations are used to estimate the maximum torque likely to be experienced once during the service life of the ship. The following load cases are intended to reflect the operational loads on the propulsion system when the propeller interacts with ice and the corresponding reaction of the complete system. The ice impact and system response cause loads in the individual shaft line components. The ice torque Q_{max} may be taken as a constant value in the complete speed range. When considerations at specific shaft speeds are performed a relevant Q_{max} may be calculated using the relevant speed.

Diesel engine plants without an elastic coupling shall be calculated at the least favourable phase angle for ice versus engine excitation, when calculated in time domain. The engine firing pulses shall be included in the calculations and their standard steady state harmonics can be used. A phase angle between ice and gas force excitation does not need to be regarded in frequency domain analysis. Misfiring does not need to be considered.

If there is a blade order resonance just above MCR speed, calculations shall cover the rotational speeds up to 105% of MCR speed.

See also Guidelines for calculations given in 5.7

5.6.3.1 Excitation for the time domain calculation

The propeller ice torque excitation for shaft line transient dynamic analysis (time domain) is defined as a sequence of blade impacts which are of half sine shape and occur at the blade. The torque due to a single blade ice impact as a function of the propeller rotation angle is then defined as:

$$Q(\varphi) = C_q \cdot Q_{max} \cdot \sin\big(\varphi(180/\alpha_i)\big) \qquad \text{[Equation 32]}$$

when φ rotates from 0 to α_i plus integer revolutions.

$$Q(\varphi) = 0$$

when φ rotates from α_i to 360 plus integer revolutions.

Where

φ = rotation angle starting when the first impact occurs

C_q and α_i parameters are given in the Table 9 below. α_i is the duration of propeller blade/ice interaction expressed in propeller rotation angle.

Table 9: Ice impact magnification and duration factors for different blade numbers

Torque excitation	Propeller/ ice interaction	C_q	α_i [deg]			
			Z=3	Z=4	Z=5	Z=6
Excitation case 1	Single ice block	0.75	90	90	72	60
Excitation case 2	Single ice block	1.0	135	135	135	135
Excitation case 3	Two ice blocks (phase shift 360/(2$\cdot Z$) deg.)	0.5	45	45	36	30
Excitation case 4	Single ice block	0.5	45	45	36	30

The total ice torque is obtained by summing the torque of single blades, taking into account the phase shift 360 deg./Z.

At the beginning and at the end of the milling sequence (within calculated duration) linear ramp functions shall be used to increase C_q to its maximum within one propeller revolution and vice versa to decrease it to zero (see examples for different Z numbers in the appendix).

The number of propeller revolutions during a milling sequence shall be obtained from the formula:

$$N_Q = 2 \cdot H_{ice} \qquad \text{[Equation 33]}$$

The number of impacts is $Z \cdot N_Q$ for blade order excitation.

An illustration of all excitation cases for different blade numbers is given in the Appendix.

The dynamic simulation shall be performed for all excitation cases starting at MCR nominal, MCR bollard condition and just above all resonance speeds (1st engine and 1st blade harmonic), so that the resonant vibration responses can be obtained. For a fixed pitch propeller plant the dynamic simulation shall also cover bollard pull condition with a corresponding speed assuming maximum possible output of the engine.

If a speed drop occurs down to stand still of the main engine, it indicates that the engine may not be sufficiently powered for the intended service task. For the consideration of loads, the maximum occurring torque during the speed drop process shall be applied. On these cases, the excitation shall follow the shaft speed.

5.6.3.2 Frequency domain excitation

For frequency domain calculations the following torque excitation may be used. The excitation has been derived so that the time domain half sine impact sequences have been assumed to be continuous and the Fourier series components for blade order and twice the blade order components have been derived. The frequency domain analysis is generally considered as conservative compared to the time domain simulation provided there is a first blade order resonance in the considered speed range.

$$Q_{F(\varphi)} = Q_{max} \cdot \left(C_{q0} + C_{q1} \cdot \sin(Z \cdot E_0 \cdot \varphi + \alpha_1) + C_{q2} \cdot \sin(2 \cdot Z \cdot E_0 \cdot \varphi + \alpha_2)\right) \quad \text{[kNm]}$$

[Equation 34]

where :

C_{q0} = mean torque component

C_{q1} = first blade order excitation amplitude

C_{q2} = second blade order excitation amplitude

φ = angle of rotation

$\alpha_{1,2}$ = phase angle of excitation component

Z = number of blades

Table 10: Coefficients for simplified excitation torque estimation

Torque excitation	Z=3					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0,375	0.375	-90	0	0	1
Excitation case 2	0.7	0.33	-90	0.05	-45	1
Excitation case 3	0.25	0.25	-90	0	0	2
Excitation case 4	0.2	0.25	0	0.05	-90	1
Torque excitation	Z=4					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.45	0.36	-90	0.06	-90	1
Excitation case 2	0.9375	0	-90	0.0625	-90	1
Excitation case 3	0.25	0.251	-90	0	0	2
Excitation case 4	0.2	0.25	0	0.05	-90	1
Torque excitation	Z=5					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.45	0.36	-90	0.06	-90	1
Excitation case 2	1.19	0.17	-90	0.02	-90	1
Excitation case 3	0.3	0.25	-90	0.048	-90	2
Excitation case 4	0.2	0.25	0	0.05	-90	1

Torque excitation	$Z=6$					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.45	0.375	-90	0.05	-90	1
Excitation case 2	1.435	0.1	-90	0	0	1
Excitation case 3	0.3	0.25	-90	0.048	-90	2
Excitation case 4	0.2	0.25	0	0.05	-90	1

Torsional vibration responses shall be calculated for all excitation cases.

The results of the relevant excitation cases at the most critical rotational speeds are to be used in the following way:

The highest response torque (between the various lumped masses in the system) is in the following referred to as peak torque Q_{peak}.

The highest torque amplitude during a sequence of impacts is to be determined as half of the range from max to min torque and is referred to as Q_{Amax}.

An illustration of Q_{Amax} is given in Figure 4. It can be determined by

$$Q_{Amax} = \left(\frac{max(Q_r(time)) - min(Q_r(time))}{2}\right) \qquad \text{[kNm]} \qquad \text{[Equation 35]}$$

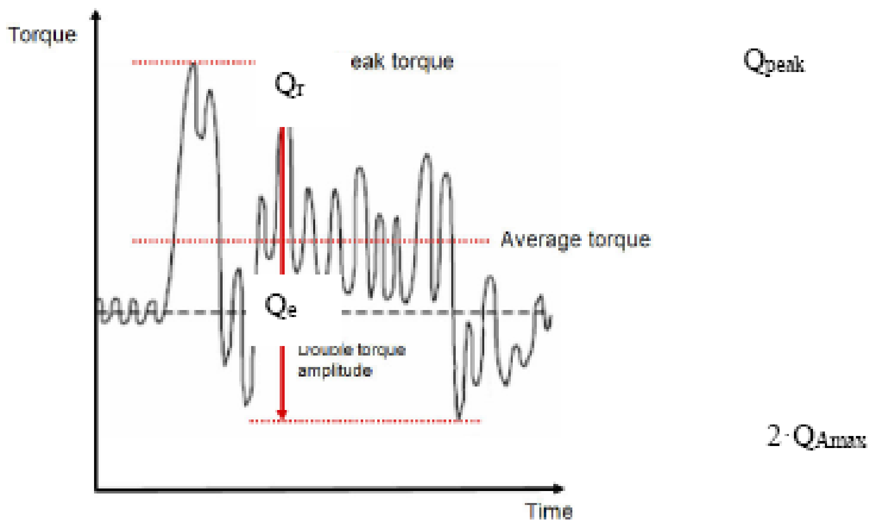


Figure 4: Interpretation of different torques in a measured curve, as example

5.6.4 Design torque along shaft line

a) If there is no relevant first order propeller torsional resonance in the range 20% (of n_n) above and 20% below the maximum operating speed in bollard condition (see Table 8), the following estimation ([Equation 36] and [Equation 37] respectively) of the maximum response torque can be used to calculate the design torque along the propeller shaft line.

$$Q_r = Q_{emax} + Q_{vib} + Q_{max} \cdot \frac{I}{I_t} \quad \text{[kNm]} \qquad \text{[Equation 36]}$$

Equation 36 is to be applied for directly coupled two stroke Diesel engines without flexible coupling.

For all other plants:

$$Q_r = Q_{emax} + Q_{max} \cdot \frac{I}{I_t} \quad \text{[kNm]} \qquad \text{[Equation 37]}$$

where:

- I = equivalent mass moment of inertia of all parts on engine side of component under consideration and
- I_t = equivalent mass moment of inertia of the whole propulsion system.

All the torques and the inertia moments shall be reduced to the rotation speed of the component being examined.

If the maximum torque, Q_{emax}, is not known, it shall be taken as follows:

Table 11: Guideline for the determination of maximum motor torque

Propeller type	Q_{emax}
Propellers driven by electric motor	Q_{motor}
CP propellers not driven by electric motor	Q_n
FP propellers driven by turbine	Q_n
FP propellers driven by diesel engine	0.75 Q_n

Here Q_{motor} is the electric motor peak torque.

b) If there is a first blade order torsional resonance in the range 20% (of n_n) above and 20% below the maximum operating speed (bollard condition), the design torque (Q_r) of the shaft component shall be determined by means of a dynamic torsional vibration analysis of the entire propulsion line in the time domain or alternatively in the frequency domain. It is then assumed that the plant is sufficiently designed to avoid harmful operation in barred speed range.

5.7 Guideline for torsional vibration calculation

The aim of torsional vibration calculations is to estimate the torsional loads for individual shaft line components over the life time in order to determine scantlings for safe operation. The model can be taken from the normal lumped mass elastic torsional vibration model (frequency domain) including the damping. Standard harmonics may be used to consider the gas forces. The engine torque - speed curve of the actual plant shall be applied.

For time domain analysis the model should include the ice excitation at propeller, the mean torques provided by the prime mover and the hydrodynamic mean torque produced by the propeller as well as any other relevant excitations. The calculations should cover the variation of phase between the ice excitation and prime mover excitation. This is extremely relevant for propulsion lines with direct driven combustion engines.

For frequency domain calculations the load should be estimated as a Fourier component analysis of the continuous sequence of half sine load peaks. The first and second order blade components should be used for excitation. The calculation should cover the whole relevant shaft speed range. The analysis of the responses at the relevant torsional vibration resonances may be performed for open water (without ice excitation) and ice excitation separately. The resulting maximum torque can be obtained for directly coupled plants by the following superposition:

$$Q_{peak} = Q_{emax} + Q_{opw} + Q_{ice} \quad \text{[kNm]} \qquad \text{[Equation 38]}$$

where:

Q_{emax} is the maximum engine torque at considered rotational speed

Q_{opw} is the maximum open water response of engine excitation at considered shaft speed and determined by frequency domain analysis

Q_{ice} is the calculated torque using frequency domain analysis for the relevant shaft speeds, ice excitation cases 1-4, resulting in the maximum response torque due to ice excitation

6 Design

6.1 Design Principle

The propulsion line shall be designed according to the pyramid strength principle in terms of its strength. This means that the loss of the propeller blade shall not cause any significant damage to other propeller shaft line components.

The propulsion line components shall withstand maximum and fatigue operational loads with the relevant safety margin. The loads do not need to be considered for shaft alignment or other calculations of normal operational conditions.

6.2 Fatigue design in general

The design loads shall be based on the ice excitation and where necessary (shafting) dynamic analysis, described as a sequence of blade impacts (5.6.3.1). The shaft response torque shall be determined according 5.6.4.

The propulsion line components are to be designed so as to prevent accumulated fatigue failure when considering the relevant loads using the linear elastic Miner's rule as defined below.

$$D = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \cdots + \frac{n_k}{N_k} \le 1 \qquad \text{[Equation 39]}$$

or

$$D = \sum_{j=1}^{j=k} \frac{n_j}{N_j} \le 1 \qquad \text{[Equation 40]}$$

Where:

k is the number of stress levels

$N_{1...k}$ is the number of load cycles to failure of the individual stress level class

$n_{1...k}$ is the accumulated number of load cycles of the case under consideration, per class

D Miners damage sum

Guidance:

The stress distribution should be divided into a frequency load spectrum having minimum 10 stress blocks (every 10 % of the load). Calculation with 5 stress blocks has been found to be too conservative. The maximum allowable load is limited by $\sigma_{ref\ 2}$ for propeller blades and yield strength for all other components. The load distribution (spectrum) should be in accordance with the Weibull distribution.

6.3 Propeller blades

6.3.1 Calculation of blade stresses due to static loads

The blade stresses (equivalent and principal stresses) shall be calculated for the design loads given in section 5.3. Finite element analysis (FEA) shall be used for stress analysis as part of the final approval for all propeller blades. The von Mises stresses, taken as σ_{st} , shall comply with Equation 42.

Alternatively, the following simplified [Equation 41] can be used in estimating the blade stresses for all propellers in the root area (r/R < 0.5) for final approval

$$\sigma_{st} = C_1 \frac{M_{BL}}{100 \cdot ct^2} \qquad \text{[MPa]} \qquad \text{[Equation 41]}$$

where:
constant C_1 is the $\frac{actual\ stress}{stress\ obtained\ with\ beam\ equation}$.

If the actual value is not available, C_1 should be taken as 1.6.

- $M_{BL} = (0.75 - r/R) \cdot R \cdot F$, for relative radius r/R < 0.5
- F is the maximum of F_b and F_f , whichever is greater.

6.3.2 Acceptability criterion for static loads

The following criterion for calculated blade stresses shall be fulfilled:

$$\frac{\sigma_{ref\ 2}}{\sigma_{st}} \geq 1.3 \qquad \text{[-]} \qquad \text{[Equation 42]}$$

where:
σ_{st} calculated stress for the design loads. If FE analysis is used in estimating the stresses, von Mises stresses shall be used.

6.3.3 Fatigue design of propeller blade

6.3.3.1 General

For materials with a two slope S-N curve (Figure 5) the fatigue calculations defined in this chapter are not required if the following criterion is fulfilled.

$$\sigma_{exp} \geq B_1 \cdot \sigma_{ref\ 2}{}^{B_2} \cdot log(N_{ice})^{B_3}$$ [Equation 43]

where:

$B1$, $B2$ and $B3$ are coefficients for open and ducted propellers, given in the Table 12 below.

Table 12: Coefficients to check a dispense from fatigue calculation

	Open propeller	Ducted propeller
$B1$	0.00328	0.00223
$B2$	1.0076	1.0071
$B2$	2.101	2.471

Where the above criterion is not fulfilled the fatigue requirements defined below shall be applied:

The fatigue design of the propeller blade is based on an estimated load distribution for the service life of the ship and the S-N curve for the blade material. An equivalent stress σ_{fat} that produces the same fatigue damage as the expected load distribution shall be calculated according to Miner's rule and the acceptability criterion for fatigue should be fulfilled as given in this section. The equivalent stress is normalised for 100 million cycles.

The blade stresses at various selected load levels for fatigue analysis are to be taken proportional to the stresses calculated for maximum loads given in section 5.3.
The peak principal stresses σ_f and σ_b are determined from F_f and F_b using FEA. The peak stress range $\Delta\sigma_{max}$ and the maximum stress amplitude σ_{Amax} are determined on the basis of load cases 1 and 3, 2 and 4.

$$\Delta\sigma_{max} = 2 \cdot \sigma_{Amax} = \left|(\sigma_{ice})_{f\ max}\right| + \left|(\sigma_{ice})_{b\ max}\right|$$ [Equation 44]

The load spectrum for backward loads is normally expected to have a lower number of cycles than the load spectrum for forward loads. Taking this into account in a fatigue analysis introduces complications that are not justified considering all uncertainties involved.
For the calculation of equivalent stress two types of S-N curves are available.

Two slope S-N curve (slopes 4.5 and 10), see Figure 5.

One slope S-N curve (the slope can be chosen), see Figure 6.

The type of the S-N-curve shall be selected to correspond with the material properties of the blade. If the S-N-curve is not known the two slope S-N curve shall be used.

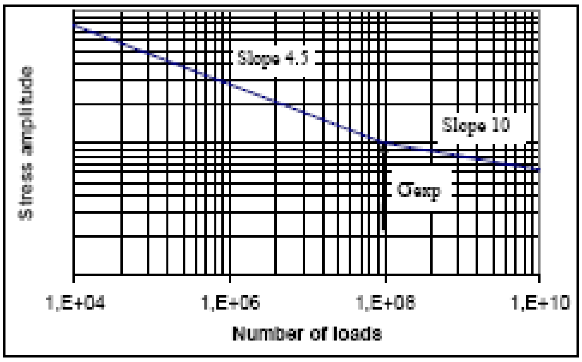


Figure 5: Two-slope S-N curve

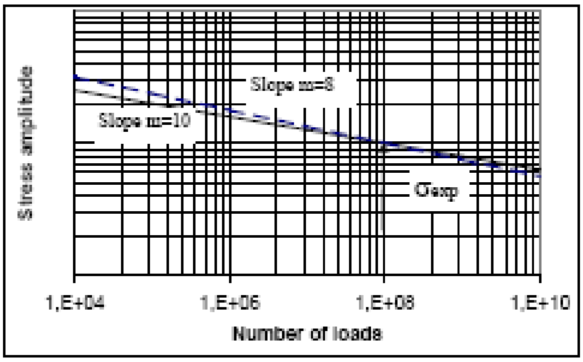


Figure 6: Constant-slope S-N curve

6.3.3.2 Equivalent fatigue stress

Note: A more general method of determining the equivalent fatigue stress of propeller blades is described in 6.5, where the principal stresses are considered according to 5.3 using the Miner's rule. For a total number of load blocks n_{bl} > 100, both methods deliver the same result. Therefore, they are regarded as equivalent.
The equivalent fatigue stress for 10^8 cycles which produces the same fatigue damage as the load distribution is:

$$\sigma_{fat} = \rho \cdot (\sigma_{ice})_{max} \quad \text{[Equation 45]}$$

where:

$$(\sigma_{ice})_{max} = 0.5 \cdot \left((\sigma_{ice})_{f\,max} - (\sigma_{ice})_{b\,max}\right) \quad \text{[Equation 46]}$$

$(\sigma_{ice})_{max}$ = mean value of the principal stress amplitudes resulting from design forward and backward blade forces at the location being studied.
$(\sigma_{ice})_{f\,max}$ = principal stress resulting from forward load
$(\sigma_{ice})_{b\,max}$ = principal stress resulting from backward load

In the calculation of $(\sigma_{ice})_{max}$, case 1 and case 3 or case 2 and case 4 are considered as pairs for $(\sigma_{ice})_{f\,max}$, and $(\sigma_{ice})_{b\,max}$ calculations. Case 5 is excluded from the fatigue analysis.
Calculation of parameter ρ for two-slope S-N curve

The error of the following method to determine the parameter ρ is sufficiently small, if the number of load cycles N_{ice} is in the range

$$5 \cdot 10^6 \le N_{ice} \le 10^8$$

The parameter ρ relates the maximum ice load to the distribution of ice loads according to the regression formula

$$\rho = C_1 \cdot (\sigma_{ice})_{max}{}^{C_2} \cdot \sigma_{fl}{}^{C_2} \cdot log(N_{ice})^{C_4} \quad \text{[Equation 47]}$$

where:

$\sigma_{fl} = \gamma_{\varepsilon 1} \cdot \gamma_{\varepsilon 2} \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp}$ is the blade material fatigue strength at 10^8 load cycles, see 6.3.3.3.

The coefficients C_1, C_2 , C_3 , and C_4 are given in Table 13

Table 13 Coefficients to evaluate material fatigue strength

	Open propeller	Ducted propeller
C_1	0.000747	0.000534
C_2	0.0645	0.0533
C_3	-0.0565	-0.0459
C_4	2.22	2.584

Calculation of parameter ρ for constant-slope S-N curve

For materials with a constant-slope S-N curve, see Figure 6, - the factor ρ shall be calculated from the following formula:

$$\rho = \left(G\frac{N_{ice}}{N_R}\right)^{\frac{1}{m}}\left(ln(N_{ice})\right)^{-\frac{1}{k}}$$ [Equation 48]

where:

k = shape parameter of the Weibull distribution

k = 1.0 for ducted propellers and

k = 0.75 for open propellers

N_R = reference number of load cycles (=10^8)

Values for the parameter G are given in Table 14 below. Linear interpolation may be used to calculate the value of G for m/k ratios other than those given in the Table 14.

Table 14: Value for the parameter G for different m/k ratios

m/k	G	m/k	G	m/k	G
3	6	5.5	287.9	8	40320
3.5	11.6	6	720	8.5	119292
4	24	6.5	1871	9	362880
4.5	52.3	7	5040	9.5	1.133×10^6
5	120	7.5	14034	10	3.623×10^6

6.3.3.3 Acceptability criterion for fatigue

The equivalent fatigue stress σ_{fat} at all locations on the blade shall fulfil the following acceptability criterion:

$$\frac{\sigma_{fl}}{\sigma_{fat}} \geq 1.5$$ [Equation 49]

where:

$$\sigma_{fl} = \gamma_{\varepsilon1} \cdot \gamma_{\varepsilon2} \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp}$$ [Equation 50]

$\gamma_{\varepsilon1}$ = reduction factor due to scatter (equal to one standard deviation)
$\gamma_{\varepsilon2}$ = reduction factor for test specimen size effect
γ_v = reduction factor for variable amplitude loading.
γ_m = reduction factor for mean stress.
σ_{exp} = mean fatigue strength of the blade material at 10^8 cycles to failure in seawater
σ_{exp} in Table 15 has been defined from the results of constant amplitude loading fatigue tests at 10^7 load cycles and 50% survival probability and has been extended to 10^8 load cycles.

Fatigue strength values and correction factors other than those given in Table 15 may be used, provided the values are determined under conditions approved by the classification society.

The S-N curve characteristics are based on two slopes, the first slope 4.5 is from 1000 to 10^8 load cycles; the second slope 10 is above 10^8 load cycles.

The maximum allowable stress for one or low number of cycles is limited to σ_{ref2}/S, with S=1.3 for static loads.

The fatigue strength σ_{fat} is the fatigue limit at 100 million load cycles.
The geometrical size factor ($\gamma_{\varepsilon 2}$) is:

$$\gamma_{\varepsilon 2} = 1 - a \cdot ln\left(\frac{t}{0.025}\right) \qquad \text{[Equation 51]}$$

where:

"a" is as given in Table 15 below and "t" is the maximum blade thickness at the considered point

The mean stress effect (γ_m) is

$$\gamma_m = 1.0 - \left(\frac{1.4 \cdot \sigma_{mean}}{\sigma_u}\right)^{0.75} \qquad \text{[Equation 52]}$$

The following values should be used for the reduction factors if actual values are not available: $\gamma_{\varepsilon 1}$ = 0.85, γ_v = 0.75, and γ_m= 0.75.

Table 15: Mean fatigue strength σ_{exp} for different material types at 10^8 load cycles and stress ratio R = -1 with a survival probability of 50%.

Mean fatigue strength σ_{exp} for different material types at 10^8 load cycles			
Bronze and brass (a=0.10)		Stainless steel (a=0.05)	
Mn-Bronze, CU1 (high tensile brass)	84 MPa	Ferritic (12Cr 1Ni)	144*) Mpa
Mn-Ni-Bronze, CU2 (high tensile brass)	84 Mpa	Martensitic (13Cr 4Ni/13Cr 6Ni)	156 Mpa
Ni-Al-Bronze, CU3	120 Mpa	Martensitic (16Cr 5Ni)	168 Mpa
Mn-Al-Bronze, CU4	113 Mpa	Austenitic (19Cr 10Ni)	132 Mpa

*) This value may be used, provided a perfect galvanic protection is active. Otherwise a reduction of about 30 MPa shall be applied.

6.4 Blade bolts, propeller hub and CP mechanism

6.4.1 General

The blade bolts, CP mechanism, propeller boss and the fitting of the propeller to the propeller shaft shall be designed to withstand the maximum static and fatigue design loads (as applicable), as defined in 5.3 and 6.3. The safety factor S against yielding due to static loads and against fatigue shall be greater than 1.5, if not stated otherwise. The safety factor S for loads, resulting from propeller blade failure as defined in 5.4 shall be greater than 1.0 against yielding.

Provided that calculated stresses duly considering local stress concentrations are less than yield strength, or maximum of 70% of σ_u of respective materials, detailed fatigue analysis is not required. In all other cases components shall be analysed for cumulative fatigue. An approach similar to that used for shafting assessment may be applied (6.5).

6.4.2 Blade bolts

Blade bolts shall withstand the following bending moment considered around a tangent on bolt pitch circle, or any other relevant axis for non-circular joints, parallel to considered root section:

$$M_{bolt} = S \cdot F_{ex\left(0.8\frac{D}{2} - r_{bolt}\right)} \quad \text{[kNm]} \quad \text{[Equation 53]}$$

where:

r_{bolt} = radius to the bolt plane [m]

S = 1.0 safety factor

Blade bolt pre-tension shall be sufficient to avoid separation between mating surfaces when the maximum forward and backward ice loads defined in 5.3 (open and ducted propellers respectively) are applied. For conventional arrangements, the following formula may be applied:

$$d_{bb} = 41 \cdot \sqrt[2]{\frac{F_{ex} \cdot (0.8 \cdot D - d) \cdot S \cdot \alpha}{\sigma_{0.2} \cdot Z_{bb} \cdot PCD}} \quad \text{[mm]} \quad \text{[Equation 54]}$$

where:

α = 1.6 torque guided tightening
= 1.3 elongation guided
= 1.2 angle guided
= 1.1 elongated by other additional means
other factors may be used, if evidence is demonstrated

d_{bb} effective diameter of blade bolt in way of thread [mm]

Z_{bb} number of blade bolts

S = 1.0 safety factor

6.4.3 CP mechanism

Separate means, e.g. dowel pins, shall be provided in order to withstand the spindle torque resulting from blade failure Q_{sex} (5.4.2) or ice interaction Q_{smax} (5.3.7), whichever is greater. Other components of the CP mechanism shall not be damaged by the maximum spindle torques (Q_{smax}, Q_{sex}). One third of the spindle torque is assumed to be consumed by friction, if not otherwise documented trough further analysis.

The diameter of fitted pins d_{fp} between the blade and blade carrier can be calculated using the formula:

$$d_{fp} = 66 \cdot \sqrt{\frac{(Q_s - Q_{fr})}{PCD \cdot z_{pin} \cdot \sigma_{0.2}}} \quad \text{[mm]} \quad \text{[Equation 55]}$$

where:

Q_s = $max(S \cdot Q_{smax}; S \cdot Q_{sex})$ [kNm] [Equation 56]

S = 1.3 for Q_{sex} and

= 1.0 for Q_{sex}

Q_{fr} = friction between connected surfaces = 0.33·Q_s

The classification society may approve alternative Q_{fr} calculation according to reaction forces due to F_{ex}, or F_f, F_b whichever is relevant, utilising a friction coefficient = 0.15.
The stress in the actuating pin can be estimated by

$$\sigma_{vMises} = \sqrt{\left(\frac{\left(F \cdot \frac{h_{pin}}{2}\right)}{\frac{\pi \cdot d_{pin}{}^3}{32}}\right)^2 + 3 \cdot \left(\frac{F}{\frac{\pi}{4} \cdot d_{pin}{}^2}\right)^2} \quad \text{[MPa]} \quad \text{[Equation 57]}$$

where:

$$F = \frac{Q_s - Q_{fr}}{l_m} \quad \text{[kN]} \quad \text{[Equation 58]}$$

l_m distance pitching centre of blade – axis of pin [m]

h_{pin} height of actuating pin [mm]

d_{pin} diameter of actuating pin [mm]

Q_{fr} friction torque in blade bearings acting on the blade palm and caused by the reaction forces due to F_{ex}, or F_f, F_b whichever is relevant; taken to one third of spindle torque Q_s

The blade failure spindle torque Q_{sex} shall not lead to any consequential damage.
Fatigue strength is to be considered for parts transmitting the spindle torque from the blade to a servo system considering the ice spindle torque acting on one blade. The maximum amplitude Q_{samax} is defined as:

$$Q_{samax} = \frac{Q_{sb} + Q_{sf}}{2} \quad \text{[kNm]} \quad \text{[Equation 59]}$$

where:

Q_{sb} spindle torque due to $|F_b|$ [kNm]

Q_{sf} spindle torque due to $|F_f|$ [kNm]

6.4.4 Servo pressure

The design pressure for the servo system shall be taken as the pressure caused by Q_{smax} or, Q_{sex} when not protected by relief valves on the hydraulic actuator side, reduced by relevant friction losses in bearings caused by the respective ice loads. The design pressure shall in any case not be less than relief valve set pressure.

6.5 Propulsion line components

The ultimate load resulting from total blade failure F_{ex} as defined in 5.4 shall consist of combined axial and bending load components, wherever this is significant. The minimum safety factor against yielding is to be 1.0 for all shaft line components.

The shafts and shafting components, such as bearings, couplings and flanges shall be designed to withstand the operational propeller/ice interaction loads as given in 5.

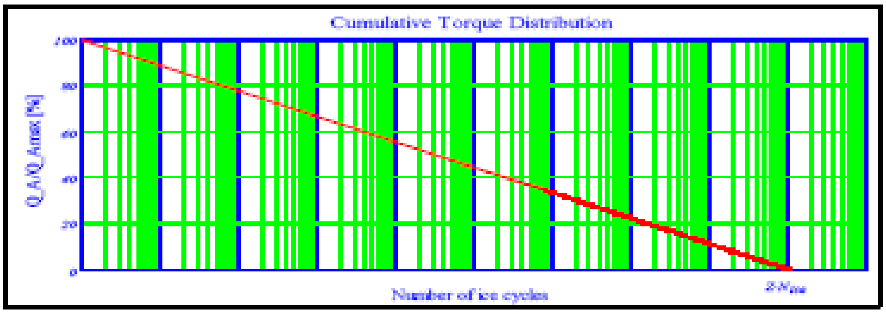
The given loads are not intended to be used for shaft alignment calculation.
Cumulative fatigue calculations shall be conducted according to the Miner's rule. A fatigue calculation is not necessary, if the maximum stress is below fatigue strength at 10^8 load cycles.

The torque and thrust amplitude distribution (spectrum) in the propulsion line is to be taken as (because Weibull exponent k = 1):

$$Q_A(N) = Q_{Amax} \cdot \left(1 - \frac{log(N)}{log(Z \cdot N_{ice})}\right) \quad \text{[Equation 60]}$$

This is illustrated by the example in the Figure 7.

Figure7: Cumulative torque distribution

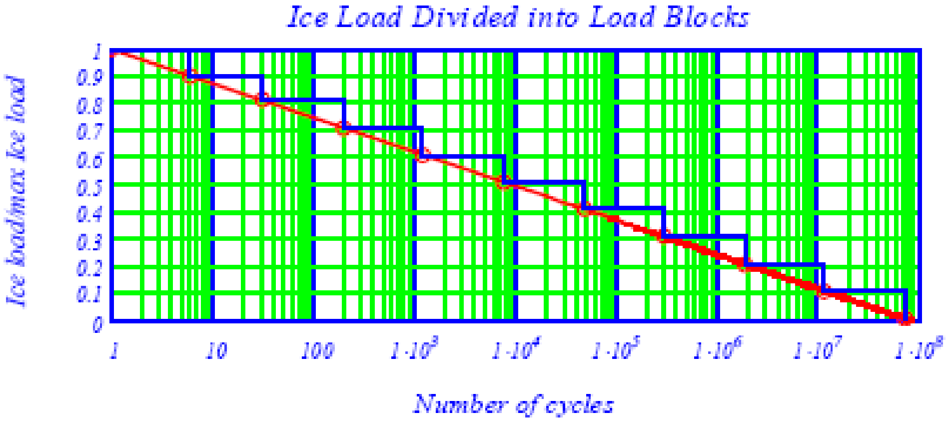


The number of load cycles in the load spectrum is defined as $Z \cdot N_{ice}$.

The Weibull exponent should be considered as k = 1.0 for both open and ducted propeller torque and bending forces. The load distribution is an accumulated load spectrum, and the load spectrum should be divided into a minimum of ten load blocks when using the Miner summation method.

The load spectrum used counts the number of cycles for 100% load to be the number of cycles above the next step, e.g. 90 % load. This ensures that the calculation is on the conservative side. Consequently, the fewer stress blocks used the more conservative the calculated safety margin.

Figure 8: Example of ice load distribution (spectrum) for the shafting (*k* = 1)



The load spectrum is divided into n_{bl}-number of load blocks for the Miner summation method.

The following formula can be used for calculation of the number of cycles for each load block.

$$n_i = N_{ice}^{\,1-\left(1-\frac{i}{n_{bl}}\right)^k} - \sum_{i=1}^{i} n_{i-1} \qquad \text{[Equation 61]}$$

where:

i = single load block i and n_{bl} is the number of load blocks

6.5.1 Propeller fitting to the shaft

6.5.1.1 Keyless cone mounting

The friction capacity (at 0° C) shall be at least S = 2.0 times the highest peak torque Q_{peak} as determined in 5.6 without exceeding the permissible hub stresses.

The necessary surface pressure $P_{0°C}$ can be determined as:

$$P_{0°C} = \frac{2 \cdot S \cdot Q_{peak}}{\pi \cdot \mu \cdot {D_S}^2 \cdot L \cdot 10^3} \qquad \text{[MPa]} \qquad \text{[Equation 62]}$$

where:

μ = 0.15 for steel-steel,
= 0.13 for steel-bronze

D_S = is the shrinkage diameter at the mid-length of the taper [m]

L = is the effective length of taper [m]

Above friction coefficients may be increased by 0.04 if glycerine is used in wet mounting.

6.5.1.2 Key mounting

Key mounting is not permitted.

6.5.1.3 Flange mounting

The flange thickness is to be at least 25% of the required aft end shaft diameter (IACS UR M34).

Any additional stress raisers such as recesses for bolt heads shall not interfere with the flange fillet unless the flange thickness is increased correspondingly.

The flange fillet radius is to be at least 10% of the required shaft diameter.

The diameter of shear pins shall be calculated according to the following equation:

$$d_{pin} = 66 \cdot \sqrt[2]{\frac{Q_{peak} \cdot S}{PCD \cdot z_{pin} \cdot \sigma_{0.2}}} \qquad \text{[mm]} \qquad \text{[Equation 63]}$$

where

z_{pin} = number of shear pins

S = 1.3 safety factor

The bolts are to be designed so that the blade failure load F_{ex} (5.4) in backward direction does not cause yielding of the bolts. The following equation should be applied:

$$d_b = 41 \cdot \sqrt{\frac{F_{ex} \cdot \left(0.8 \frac{D}{PCD} + 1\right) \cdot \alpha}{\sigma_{0.2} \cdot z_b}} \qquad \text{[mm]} \qquad \text{[Equation 64]}$$

where:

α = 1.6 torque guided tightening
= 1.3 elongation guided
= 1.2 angle guided
= 1.1 elongated by other additional means
other factors may be used, if evidence is demonstrated

d_b diameter flange bolt [mm]

z_b number of flange bolts

6.5.2 Propeller shaft

The propeller shaft is to be designed to fulfil the following:

6.5.2.1 The blade failure load F_{ex} (5.4) applied parallel to the shaft (forward or backwards) shall not cause yielding. The bending moment need not to be combined with any other loads. The diameter d_p in way of the aft stern tube bearing shall not be less than:

$$d_p = 160 \cdot \sqrt[3]{\frac{F_{ex} \cdot D}{\sigma_{0.2} \cdot \left(1 - \frac{{d_i}^4}{{d_p}^4}\right)}} \quad \text{[mm]} \qquad \text{[Equation 65]}$$

where:

d_p = propeller shaft diameter [mm]

d_i = propeller shaft inner diameter [mm]

Forward from the aft stern tube bearing the shaft diameter may be reduced based on direct calculation of the actual bending moment, or by the assumption that the bending moment caused by F_{ex} is linearly reduced to 25% at the next bearing and in front of this linearly to zero at third bearing.

Bending due to maximum blade forces F_b and F_f have been disregarded since the resulting stress levels are much lower than the stresses caused by the blade failure load.

6.5.2.2 The stresses due to the peak torque Q_{peak} shall have a minimum safety factor of S=1.5 against yielding in plain sections and S=1.0 in way of stress concentrations in order to avoid bent shafts.

Minimum diameter of:

plain shaft:

$$d_p = 210 \cdot \sqrt[3]{\frac{Q_{peak} \cdot S}{\sigma_{0.2} \cdot \left(1 - \frac{{d_i}^4}{d^4}\right)}} \quad \text{[mm]} \qquad \text{[Equation 66]}$$

notched shaft:

$$d_p = 210 \cdot \sqrt[3]{\frac{Q_{peak} \cdot S \cdot \alpha_t}{\sigma_{0.2} \cdot \left(1 - \frac{{d_i}^4}{d^4}\right)}} \quad \text{[mm]} \qquad \text{[Equation 67]}$$

where:

α_t = local stress concentration factor in torsion.

Notched shaft diameter shall in any case not be less than the required plain shaft diameter.

6.5.2.3 The torque amplitudes (5.6.4) with the corresponding number of load cycles shall be used in an accumulated fatigue evaluation where the safety factor is S_{fat}=1.5. If the plant has high engine excited torsional vibrations (e.g. direct coupled 2-stroke engines), this shall also be considered.

6.5.2.4 The fatigue strengths σ_F and τ_F (3 million cycles) of shaft materials may be assessed on the basis of the material's yield or 0.2% proof strength as:

$$\sigma_F = 0.436 \cdot \sigma_{0.2} + 77 = \tau_F \cdot \sqrt{3} \qquad \text{[MPa]} \qquad \text{[Equation 68]}$$

This is valid for small polished specimens (no notch) and reversed stresses, see "VDEH 1983 Bericht Nr. ABF11 Berechnung von Wöhlerlinien für Bauteile aus Stahl".

The high cycle fatigue (HCF) is to be assessed based on the above fatigue strengths, notch factors (i.e. geometrical stress concentration factors and notch sensitivity), size factors, mean stress influence and the required safety factor of 1.6 at 3 million cycles increasing to 1.8 at 10^9 cycles.

The low cycle fatigue (LCF) representing 10^4 cycles is to be based on the smaller value of yield or 0.7 of tensile strength/$\sqrt{3}$. The criterion utilises a safety factor of 1.25.

The LCF and HCF as given above represent the upper and lower knees in a stress-cycle diagram. Since the required safety factors are included in these values, a Miner sum of unity is acceptable.

6.5.3 Intermediate shafts

The intermediate shafts are to be designed to fulfil 6.5.2.2 to 6.5.2.4.

6.5.4 Shaft connections

6.5.4.1 Shrink fit couplings (keyless)

See 6.5.1.1. A safety factor of S = 1.8 shall be applied.

6.5.4.2 Key mounting

Key mounting is not permitted.

6.5.4.3 Flange mounting

The flange thickness is to be at least 20% of the required shaft diameter (IACS UR M34).

Any additional stress raisers such as recesses for bolt heads shall not interfere with the flange fillet unless the flange thickness is increased correspondingly.

The flange fillet radius is to be at least 8% of the shaft diameter (IACS UR M34).

The diameter of ream fitted (light press fit) bolts shall be chosen so that the peak torque is transmitted with a safety factor of 1.9. This accounts for a prestress. Pins shall transmit the peak torque with a safety factor of 1.5 against yielding ([Equation 63]).

The bolts are to be designed so that the blade failure load (5.4) in backward direction does not cause yielding.

6.5.4.4 Splined shaft connections

Splined shaft connections can be applied where no axial or bending loads occur. A safety factor of S = 1.5 against allowable contact and shear stress resulting from Q_{peak} shall be applied.

6.5.4.5 Gear transmissions

6.5.4.6 Shafts

Shafts in gear transmissions shall meet the same safety level as intermediate shafts, but where relevant, bending stresses and torsional stresses shall be combined (e.g. by von Mises for static loads). Maximum permissible deflection in order to maintain sufficient tooth contact pattern shall be considered for the relevant parts of the gear shafts.

6.5.4.7 Gearing

The gearing shall fulfil following three acceptance criteria:

- Tooth root stresses

- Pitting of flanks

- Scuffing

In addition to above 3 criteria subsurface fatigue may need to be considered.

Common for all criteria is the influence of load distribution over the face width. All relevant parameters are to be considered, such as elastic deflections (of mesh, shafts and gear bodies), accuracy tolerances, helix modifications, and working positions in bearings (especially for multiple input single output gears).

The load spectrum (see 6.5) may be applied in such a way that the numbers of load cycles for the output wheel are multiplied by a factor of (number of pinions on the wheel / number of propeller blades Z). For pinions and wheels operating at higher speeds the numbers of load cycles are found by multiplication with the gear ratios. The peak torque (Q_{peak}) is also to be considered during calculations.

Cylindrical gears can be assessed on the basis of the international standard ISO 6336 series (i.e. ISO 6336-1:2019, ISO 6336-2:2019, ISO 6336-3:2019, ISO 6336-4:2019, ISO 6336-5:2016 and ISO 6336-6:2019), provided that "method B" is used. Standards within the classification societies can also be applied provided that they are considered equivalent to the above mentioned ISO 6336.

For Bevel Gears the methods or standards used or acknowledged by the classification society can be applied provided that they are properly calibrated.

Tooth root safety shall be assessed against the peak torque, torque amplitudes (with the pertinent average torque) as well as the ordinary loads (open water free running) by means of accumulated fatigue analyses. The resulting factor of safety is to be at least 1.5. (Ref ISO 6336 Pt 1, 3 and 6 and IACS UR M56)

The safety against pitting shall be assessed in the same way as tooth root stresses, but with a minimum resulting safety factor of 1.2. (Ref ISO 6336-1:2019, ISO 6336-2:2019 and ISO 6336-6:2019 as well as IACS UR M56).
The scuffing safety (flash temperature method – ref. ISO/TR 13989-1:2000 and ISO/TR 13989-2:2000) based on the peak torque shall be at least 1.2 when the FZG class of the oil is assumed one stage below specification.

The safety against subsurface fatigue of flanks for surface hardened gears (oblique fracture from active flank to opposite root) is to be assessed at the discretion of each Classification Society. (It should be noted that high overloads can initiate subsurface fatigue cracks that may lead to a premature failure. In lieu of analyses UT inspection intervals may be used.)

6.5.4.8 Bearings

See section 6.5.8.

6.5.4.9 Gear wheel shaft connections

The torque capacity shall be at least 1.8 times the highest peak torque Q_{peak} (at considered rotational speed) as determined in 6.5 without exceeding the permissible hub stresses of 80% yield.

6.5.5 Clutches

Clutches shall have a static friction torque of at least 1.3 times the peak torque Q_{peak} and dynamic friction torque 2/3 of the static.
Emergency operation of clutch after failure of e.g. operating pressure shall be made possible within reasonably short time. If this is arranged by bolts, it shall be on the engine side of the clutch in order to ensure access to all bolts by turning the engine.

6.5.6 Elastic couplings

There shall be a separation margin of at least 20% between the peak torque and the torque where any twist limitation is reached.

$$Q_{peak} < 0.8 \cdot T_{kmax}\ (N = 1) \qquad \text{[kNm]} \qquad \text{[Equation 69]}$$

There shall be a separation margin of at least 20% between the maximum response torque Q_{peak} (see Figure 4) and the torque where any mechanical twist limitation and/or the permissible maximum torque of the elastic coupling, valid for at least a single load cycle (N=1), is reached.

A sufficient fatigue strength shall be demonstrated at design torque level $Q_r(N = x)$ and $Q_A(N = x)$. This may be demonstrated by interpolation in a Weibull torque distribution (similar to Figure 7):

$$\frac{Q_r(N=x)}{Q_r(N=1)} = 1 - \frac{\log(x)}{\log(Z \cdot N_{ice})} \qquad [-] \qquad \text{[Equation 70]}$$

respectively

$$\frac{Q_A(N=x)}{Q_A(N=1)} = 1 - \frac{\log(x)}{\log(Z \cdot N_{ice})} \qquad [-] \qquad \text{[Equation 71]}$$

Where Q_r (N=1) corresponds to Q_{peak} and Q_A (N=1) to Q_{Amax}.

Q_r $(N\text{=}5E4)\cdot S < T_{Kmax}$ $(N\text{=}5E4)$	[kNm]	[Equation 72]
Q_r $(N\text{=}1E6)\cdot S < T_{KV}$	[kNm]	[Equation 73]
Q_A $(N\text{=}5E4)\cdot S < \Delta T_{max}$ $(N\text{=}5E4)$	[kNm]	[Equation 74]

S is the general safety factor for fatigue, equal to 1.5.

See illustration in below Figure 9, Figure 10 and Figure 11.

The torque amplitude (or range Δ) shall not lead to fatigue cracking, i.e. exceeding the permissible vibratory torque. The permissible torque may be determined by interpolation in a Weibull torque distribution where T_{Kmax1} respectively ΔT_{Kmax} refer to 50000 cycles and T_{KV} refer to 10^6 cycles. See illustration in below Figure 9, Figure 10 and Figure 11.

$T_{Kmax1} \geq Q_r$ at $5 \cdot 10^4$ load cycles [kNm] [Equation 75]

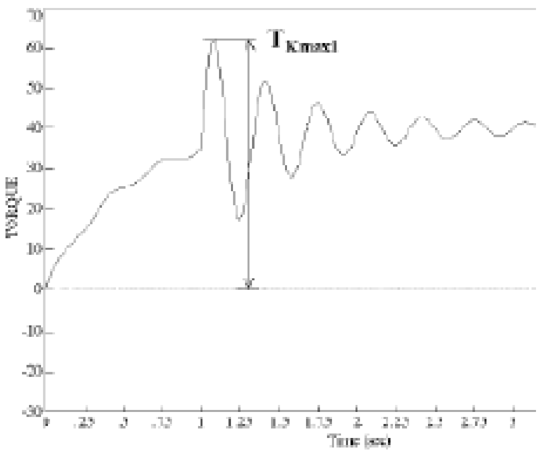


Figure 9

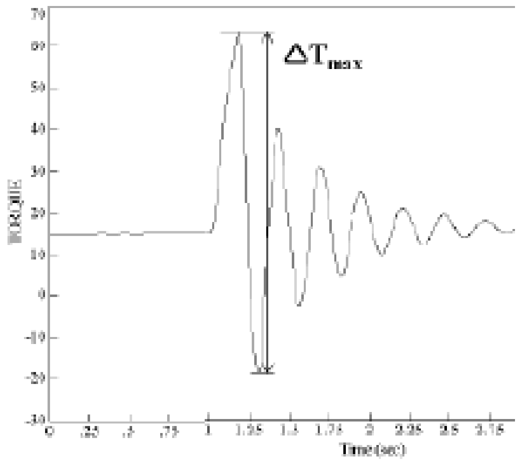


Figure 10

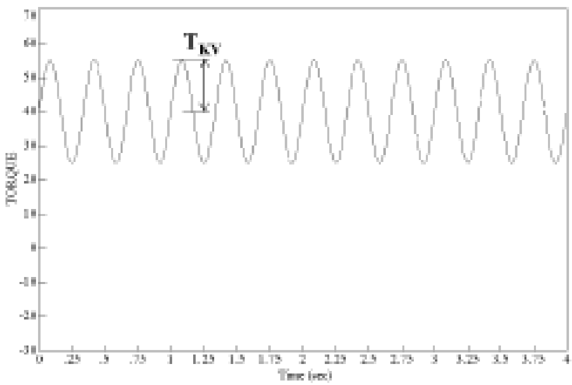


Figure 11

6.5.7 Crankshafts

Special considerations apply for plants with large inertia (e.g. flywheel, tuning wheel or PTO) in the non-driving end front of the engine (opposite to main power take off).

6.5.8 Bearings

The aft stern tube bearing as well as the next shaft line bearing are to withstand F_{ex} as given in 5.4, in such a way that the ship can maintain operational capability. Rolling bearings are to have an L_{10a} lifetime of at least 40 000 hours according to ISO 281:2007. Thrust bearings and their housings are to be designed to withstand with a safety factor S = 1.0 the maximum response thrust 5.5 and the axial force resulting from the blade failure load F_{ex} in 5.4. For the purpose of calculation, except for F_{ex}, the shafts are assumed to rotate at rated speed. For pulling propellers special consideration is to be given to loads from ice interaction on the propeller hub.

6.5.9 Seals

Seals are to prevent egress of pollutants and be suitable for the operating temperatures. Contingency plans for preventing the egress of pollutants under failure conditions are to be documented.

Seals installed are to be suitable for the intended application. The manufacturer is to provide service experience in similar applications and/or testing results for consideration.

I3.5.2 Azimuthing Main Propulsors

In addition to the above requirements, special consideration shall be given to those loading cases which are extraordinary for propulsion units when compared with conventional propellers. The estimation of load cases shall reflect the way the thrusters are intended to operate on the specific ship. In this respect, for example, the loads caused by the impacts of ice blocks on the propeller hub of a pulling propeller shall be considered. Furthermore, loads resulting from the thrusters operating at an oblique angle to the flow shall be considered. The steering mechanism, the fitting of the unit, and the body of the thruster shall be designed to withstand the loss of a blade without damage. The loss of a blade shall be considered for the propeller blade orientation which causes the maximum load on the component being studied. Typically, top-down blade orientation places the maximum bending loads on the thruster body.

Azimuth thrusters shall also be designed for estimated loads caused by thruster body/ice interaction. The thruster body shall withstand the loads obtained when the maximum ice blocks, which are given in section 5.2, strike the thruster body when the ship is at a typical ice operating speed. In addition, the design situation in which an ice sheet glides along the ship's hull and presses against the thruster body should be considered. The thickness of the sheet should be taken as the thickness of the maximum ice block entering the propeller, as defined in section 5.2.

7 Prime Movers

7.1 Propulsion engines

Engines are to be capable of being started and running the propeller in bollard condition.

Propulsion plants with CP propeller are to be capable being operated even when the CP system is at full pitch as limited by mechanical stoppers.

7.2 Starting arrangements

The capacity of the air receivers shall be sufficient to provide, without recharging, not less than 12 consecutive starts of the propulsion engine, if this has to be reversed for going astern or 6 consecutive starts if the propulsion engine does not have to be reversed for going astern.

If the air receivers serve any other purposes than starting the propulsion engine, they shall have additional capacity sufficient for these purposes.

The capacity of the air compressors shall be sufficient for charging the air receivers from atmospheric to full pressure in one (1) hour, except for a ship with the ice class PC6 to PC1, if its propulsion engine has to be reversed for going astern, in which case the compressor shall be able to charge the receivers in half an hour.

7.3 Emergency power units

Provisions shall be made for heating arrangements to ensure ready starting from cold of the emergency power units at an ambient temperature applicable to the Polar Class of the ship.

Emergency power units shall be equipped with starting devices with a stored energy capability of at least three consecutive starts at the above mentioned temperature. The source of stored energy shall be protected to preclude critical depletion by the automatic starting system, unless a second independent mean of starting is provided. A second source of energy shall be provided for an additional three starts within 30 min., unless manual starting can be demonstrated to be effective.

8 Equipment fastening loading accelerations

8.1 General

Essential equipment and supports shall be suitable for the accelerations as indicated in the following paragraphs. Accelerations are to be considered as acting independently.

8.2 Longitudinal Impact Accelerations, a_1

Maximum longitudinal impact acceleration at any point along the hull girder,

$$a_1 = \frac{F_{IB}}{\Delta} \cdot \left(1.1 \cdot \tan(\gamma + \varphi) + \left(7 \cdot \frac{H}{L}\right)\right) \qquad [\mathrm{m/s^2}] \qquad \text{[Equation 76]}$$

8.3 Vertical acceleration, a_v

Combined vertical impact acceleration at any point along the hull girder,

$$a_v = 2.5 \cdot \left(\frac{F_{IB}}{\Delta}\right) \cdot F_X \qquad [\mathrm{m/s^2}] \qquad \text{[Equation 77]}$$

F_X = 1.3 at FP
= 0.2 at midships
= 0.4 at AP
= 1.3 at AP for vessels conducting ice breaking astern
Intermediate values to be interpolated linearly.

8.4 Transverse impact acceleration, a_t

Combined transverse impact acceleration at any point along hull girder,

$$a_t = 3F_i \frac{F_X}{\Delta} \qquad [\mathrm{m/s^2}] \qquad \text{[Equation 78]}$$

F_X = 1.5 at FP

= 0.25 at midships

= 0.5 at AP

= 1.5 at AP for vessels conducting ice breaking astern

Intermediate values to be interpolated linearly.

where:

φ = maximum friction angle between steel and ice, normally taken as 10 [degrees]

γ = bow stem angle at waterline [degrees]

Δ = displacement

L = length between perpendiculars [m]

H = distance in meters from the water line to the point being considered [m]

F_{IB} = vertical impact force, defined in UR I2.13.2.1

F_i = total force normal to shell plating in the bow area due to oblique ice impact, defined in UR I2.3.2.1

I3.9 Auxiliary Systems

I3.9.1 Machinery shall be protected from the harmful effects of ingestion or accumulation of ice or snow. Where continuous operation is necessary, means should be provided to purge the system of accumulated ice or snow.

I3.9.2 Means should be provided to prevent damage to tanks containing liquids due to freezing.

I3.9.3 Vent pipes, intake and discharge pipes and associated systems shall be designed to prevent blockage due to freezing or ice and snow accumulation.

I3.10 Sea Inlets and Cooling Water Systems

I3.10.1 Cooling water systems for machinery that are essential for the propulsion and safety of the vessel, including sea chests inlets, shall be designed for the environmental conditions applicable to the ice class.

I3.10.2 At least two sea chests are to be arranged as ice boxes for class **PC1** to **PC5** inclusive where. The calculated volume for each of the ice boxes shall be at least 1m^3 for every 750 kW of the total installed power. For **PC6** and **PC7** there shall be at least one ice box located preferably near centre line.

I3.10.3 Ice boxes are to be designed for an effective separation of ice and venting of air.

I3.10.4 Sea inlet valves are to be secured directly to the ice boxes. The valve shall be a full bore type.

I3.10.5 Ice boxes and sea bays are to have vent pipes and are to have shut off valves connected direct to the shell.

I3.10.6 Means are to be provided to prevent freezing of sea bays, ice boxes, ship side valves and fittings above the load waterline.

I3.10.7 Efficient means are to be provided to re-circulate cooling seawater to the ice box. Total sectional area of the circulating pipes is not to be less than the area of the cooling water discharge pipe.

I3.10.8 Detachable gratings or manholes are to be provided for ice boxes. Manholes are to be located above the deepest load line. Access is to be provided to the ice box from above.

I3.10.9 Openings in ship sides for ice boxes are to be fitted with gratings, or holes or slots in shell plates. The net area through these openings is to be not less than 5 times the area of the inlet pipe. The diameter of holes and width of slot in shell plating is to be not less than 20 mm. Gratings of the ice boxes are to be provided with a means of clearing. Clearing pipes are to be provided with screw-down type non return valves.

I3.11 Ballast tanks

I3.11.1 Efficient means are to be provided to prevent freezing in fore and after peak tanks and wing tanks located above the water line and where otherwise found necessary.

12 Ventilation Systems

12.1 The air intakes for machinery and accommodation ventilation are to be located on both sides of the ship at locations where manual de-icing is possible. Anti-icing protection of the air inlets may be accepted as an equivalent solution to location on both sides of the ship and manual de-icing at the Society's discretion. Notwithstanding the above, multiple air intakes are to be provided for the emergency generating set and are to be as far apart as possible.

12.2 The temperature of the inlet air shall be suitable for:

- the safe operation of the machinery; and
- the thermal comfort in the accommodation.

Accommodation and ventilation air intakes shall be provided with means of heating, if needed.

13 Steering Systems

13.1 Rudder stops are to be provided. The design ice force on rudder shall be transmitted to the rudder stops without damage to the steering system.

An ice knife shall in general be fitted to protect the rudder in centre position. The ice knife shall extend below BWL. Design forces shall be determined according to the I2.15.

13.2 The rudder actuator is to comply with the following requirements 13.2.1 and 13.2.2:

13.2.1 The rudder actuator is to be designed for a holding torque obtained by multiplying the open water torque resulting from the application of SOLAS Reg. II-1 /29.3.2 (considering however a maximum speed of 18 knots, by following factors:

Ice Class	PC1	PC2	PC3	PC4	PC5	PC6	PC7
Factor	5	5	3	3	3	1.5	1.5

13.2.2 The design pressure for calculations to determine the scantlings of the rudder actuator is to be at least 1.25 times the maximum working pressure corresponding to the holding torque defined in 13.2.1 (Derived from SOLAS Reg. II-1 / 29.2.2).

13.3 The rudder actuator is to be protected by torque relief arrangements, assuming the following turning speeds [deg/s] without an undue pressure rise (ref UR M42 for undue pressure rise):

Table 17: Steering gear turning speeds

Ice Class	PC1 and PC2	PC3 to PC5	PC6 and PC7
Turning speeds [deg/s]	10	7.5	6

If the rudder and actuator design can withstand such rapid loads, this special relief arrangement is not necessary and a conventional one may be used instead (UR M42).

13.4 Additionally for icebreakers, fast-acting torque relief arrangements are to be fitted in order to provide effective protection of the rudder actuator in case of the rudder being pushed rapidly hard over against the stops.

For hydraulically operated steering gear, the fast-acting torque relief arrangement is to be so designed that the pressure cannot exceed 115% of the set pressure of the safety valves when the rudder is being forced to move at the speed indicated in Table 18, also when taking into account the oil viscosity at the lowest expected ambient temperature in the steering gear compartment.

For alternative steering systems the fast-acting torque relief arrangement is to demonstrate an equivalent degree of protection to that required for hydraulically operated arrangements.

The turning speeds to be assumed for each ice class are shown in Table 18 below.

Table 18: Steering gear turning speeds for icebreakers

Ice Class	PC1 and PC2	PC3 to PC5	PC6 and PC7
Turning speeds [deg/s]	40	20	15

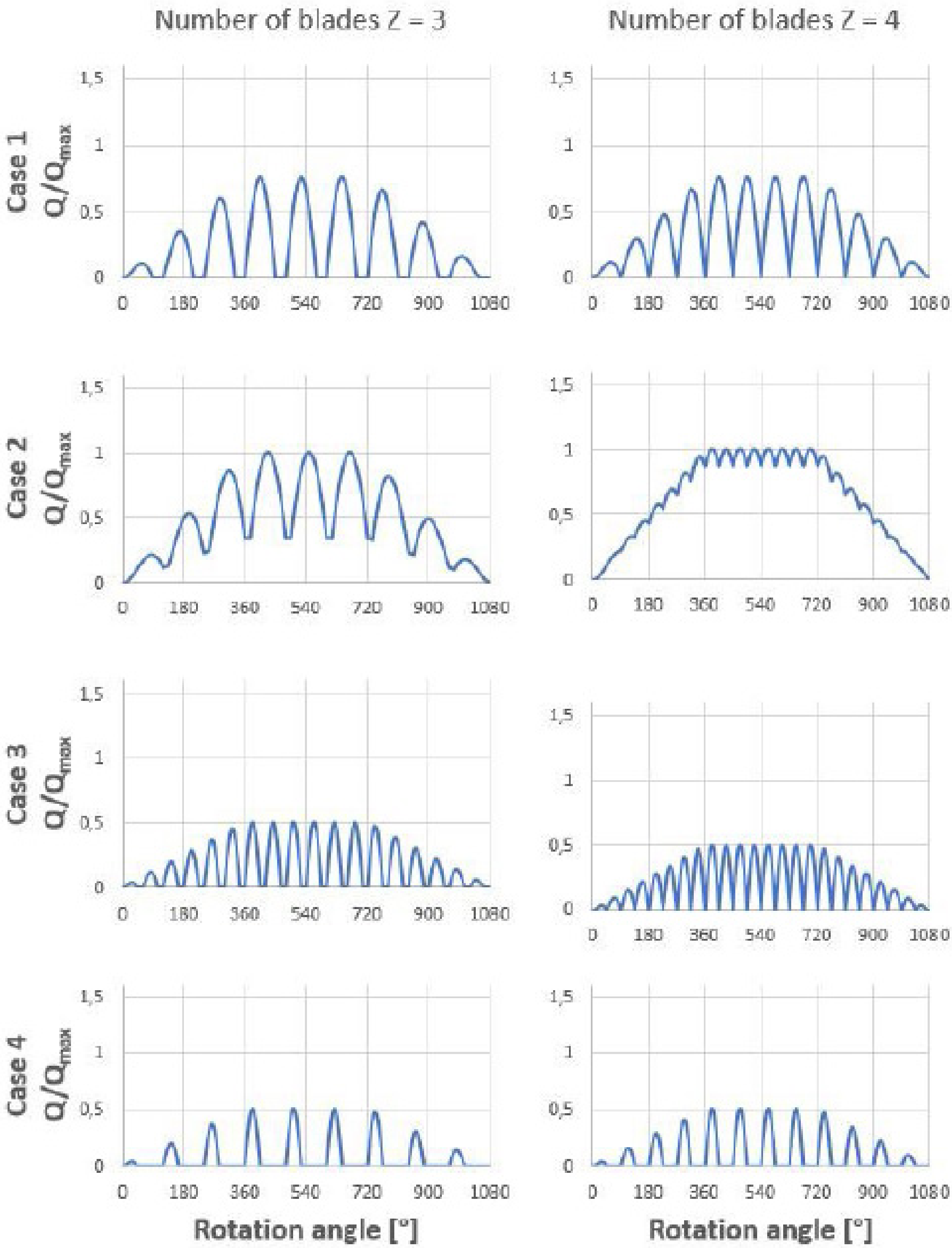
The arrangement is to be designed such that steering capacity can be speedily regained.

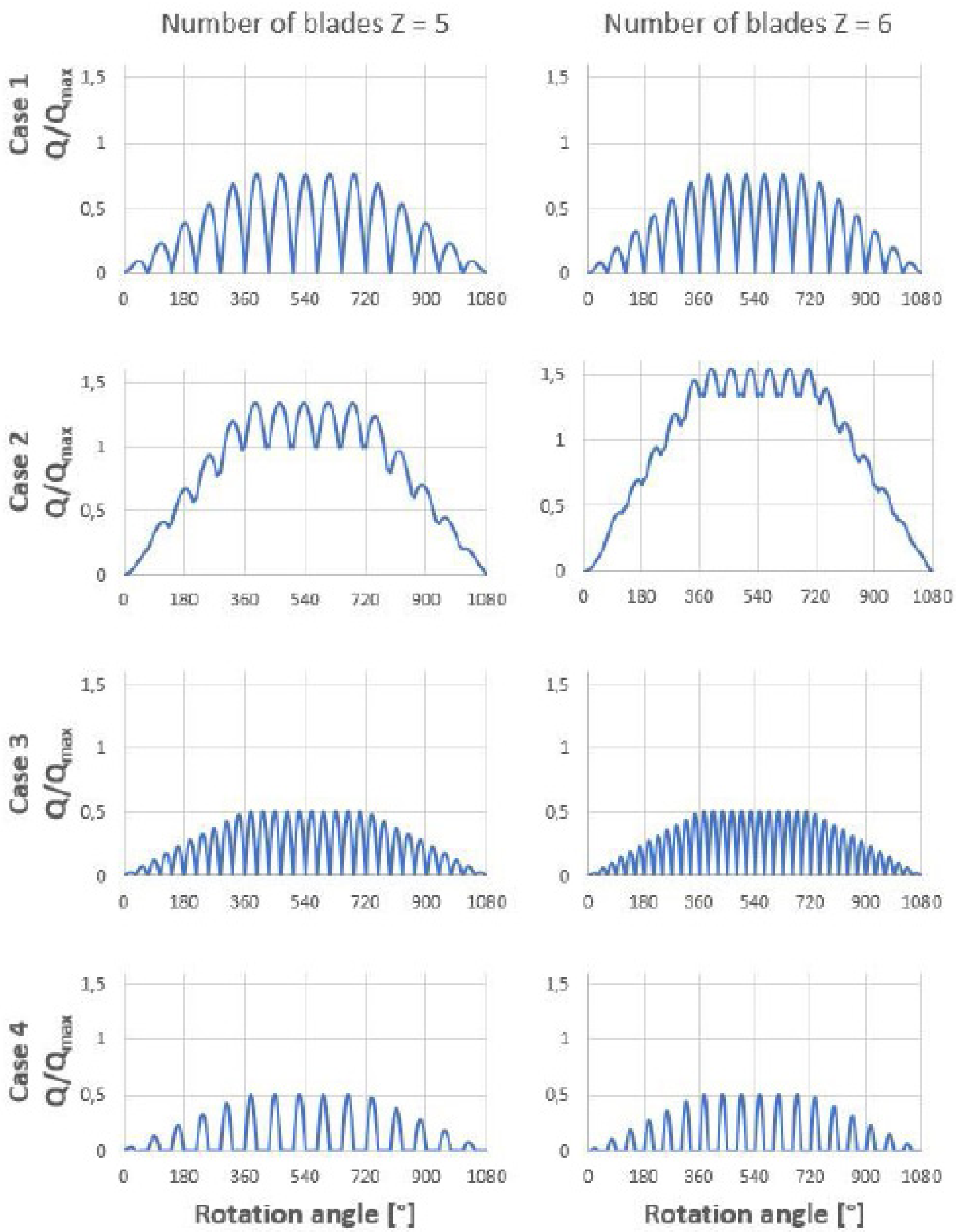
I3.14 Alternative Design

I3.14.1 As an alternative – a comprehensive design study may be submitted and may be requested to be validated by an agreed test programme.

APPENDIX

The following illustrations show the excitation torque for all torsional load cases given in this UR for different blade numbers (Z). The plots have been made using data for PC7 (H_{ice} = 1.5)





List of amendments effective as of 1 July 2024

Item	*Title/Subject*	*Source*
I.3	Machinery Requirements for PRS Polar Class Ships	IACS UR I3 (Rev.2 Jan 2023)