RULES
R047-2024

CHINA CLASSIFICATION SOCIETY

RULES FOR HEAVY ICEBREAKERS

2024

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Beijing

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Section 1 GENERAL

1.1 General requirements

1.1.1 The Rules apply to enhanced icebreakers engaged in year-round operation in moderate multi-year ice conditions and above.

1.1.2 Ships applying for CCS enhanced icebreaker notation are to comply with the applicable requirements of CCS Rules for Classification of Sea-going Steel Ships and relevant guidelines in addition to the applicable provisions of the Rules.

1.1.3 Icebreakers complying with the Rules are to be capable of independent navigation in polar waters in moderate first year ice conditions as a minimum and be assigned a winter protection notation ACC-POLAR (DST).

1.1.4 For icebreakers with novel structure and new characteristics, alternative and equivalent means are accepted upon agreement.

1.2 Definitions

1.2.1 Unless specifically provided otherwise, the definitions of the Rules are as follows:

(1) Enhanced icebreaker means a dedicated ship having an operational profile that includes escort, scientific research and material logistics or ice management functions, having powering and dimensions that allow it to undertake aggressive operations in ice-covered waters, and being assigned the Icebreaker* notation.

(2) Heavy icebreaker means an enhanced icebreaker with Ice Class not less than PC2 and continuous ice-breaking capability not less than 2 m (2~3 knots) (hereinafter referred to as “the icebreaker”).

(3) Polar Class (PC): see 13.1.2, Chapter 13, PART EIGHT of CCS Rules for Classification of Sea-going Steel Ships.

(4) Ship intended to operate in low air temperature means a ship intended to undertake voyages to or through areas where the Lowest Mean Daily Low Temperature (LMDLT) is below -10℃.

(5) Upper ice waterline (UIWL): waterline defined by the maximum draughts fore, amidships and aft.

(6) Lower ice waterline (LIWL): waterline defined by the minimum draughts fore, amidships and aft. 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.

(7) Length L_{UI}: 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). In ships with unusual stern and bow arrangement the length L_{UI} will be specially considered.

(8) Ship displacement D_{UI}: 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.

(9) ARC-M: high strength steel complying with Appendix 8 High Strength Steel for Polar Ship of CCS Guidelines for Polar Ship.

(10) Design Service Temperature (DST): see 23.1.2.1(3), Chapter 23, PART EIGHT of CCS Rules

for Classification of Sea-going Steel Ships.
(11) Anti-icing and de-icing: see 23.1.2.1(7)&(8), Chapter 23, PART EIGHT of CCS Rules for Classification of Sea-going Steel Ships.

1.3 Operation profile

1.3.1 Main functions of icebreakers

1.3.1.1 The functions of icebreakers are the basis for determining operation scenarios and operation profile. The functions of icebreakers generally are escort, scientific research and material logistics or ice management. Where alternative functions are selected, the descriptions of such functions are to be included in relevant documents.

(1) Escort: engaged in patrol and search and rescue missions in ice-covered waters, opening up waterways in ice-covered waters for other ships, rescuing stranded ships, and towing them if necessary. Within the scope of icebreaker safety operations, the easiest and fastest route to rescue is generally chosen.

(2) Scientific research and material logistics: engaged in independent scientific research operations in ice-covered waters, opening up waterways in ice-covered waters and escorting ships that supply the scientific research stations, and at the same time transporting materials for the scientific research stations. Rerouting is usually possible based on ice conditions and changes.

(3) Ice management: providing de-icing, support and protection for offshore installations/operations. Large chunks of floating ice are often actively broken to remove or reduce the threat of sea ice to offshore installations/operations.

1.3.1.2 The owner and/or designer is to select the appropriate ice-breaking capability indicators based on the icebreaker's designed function and expected ice conditions. According to the main functions and operation profile of icebreakers, the ice-breaking capability indicators generally include the following:

(1) Continuous ice-breaking capability (in terms of speed and ice thickness);

(2) The turning capability in level ice (in terms of diameter, turning speed and ice thickness);

(3) Ramming capability (in terms of speed and ice condition).

1.3.1.3 For icebreakers designed with special operating modes, such as oblique ice breaking and shallow water operation in ice-covered waters, special considerations are to be given to the ice load in the hull area. The method in Appendix 1 of the Rules or the ice pond test may be used, and the relevant documents are to be submitted to CCS.

1.4 Plans and documents

1.4.1 The following plans/documents are to be submitted for approval:

(1) Structure of ice-strengthened areas (including ice belt divisions which may be included in the relevant structural plans);

(2) Details of main propulsion machinery. The instructions for main propulsion, steering gear, emergency and essential auxiliary systems are to include their operation limitations and instructional documents on the necessary load control function of the main propulsion machinery;

(3) Arrangement on main propulsion, emergency and auxiliary systems, protection against freezing and icing, as well as detailed design instructions on their operation capabilities in the intended environmental conditions;

(4) Strength calculations for propellers;
(5) Shaft torsional vibration and strength calculations.
1.4.2 The following plans/documents are to be submitted for information:
(1) The functions, design ice conditions and ice-breaking capability indicators of the icebreaker;
(2) Structural strength calculations;
(3) Pod strength calculations, where applicable;
(4) Report on sea trials in ice-covered waters or ice pond model test report or other approved equivalent method verification report that fully validates the ship's ice-breaking capability.

1.5 Symbols and class notations

1.5.1 Polar ships complying with the requirements of the Rules may be assigned the class notation Icebreaker*. The notation is to be attached with the corresponding Polar Class, e.g. Icebreaker* PC2.
1.5.2 When high strength steel are used for hull structures exposed to weather and sea, the class notation ARC-M(x) may be assigned, where x indicates the design service temperature.

1.6 General requirements

1.6.1 General arrangement
1.6.1.1 Ship arrangements are to comply with the requirements of Section 12 Structural Arrangement, Chapter 1, PART TWO and Section 3 Ship Arrangements, Chapter 23, PART EIGHT of CCS Rules for Classification of Sea-going Steel Ships.
1.6.1.2 The bow shape of the icebreaker is to avail ramming ice-breaking operation. At the same time, ice obstacles are to be fitted, such as ice skeg, to prevent stern deck immersion. The speed, stability and freeboard are generally to be considered for design. When the ship stops at the front toe or other lowest position is on the ice, the stern deck is to be prevented from immersion and the ship is to maintain sufficient positive stability.
1.6.1.3 A transverse bulkhead is to be provided behind the ice skeg, which is to extend watertight to the freeboard deck. When the ship is loaded at the deepest subdivision draught, flooding of all compartments forward of the ice skeg will not result in flooding of any other compartments and will not cause unacceptable loss of stability to the ship, in accordance with the survivability factor *Si* after damage specified in 1.6.2.1 of this Section of not less than 1 or a residual stability criterion after damage which meets the requirements of the ship.

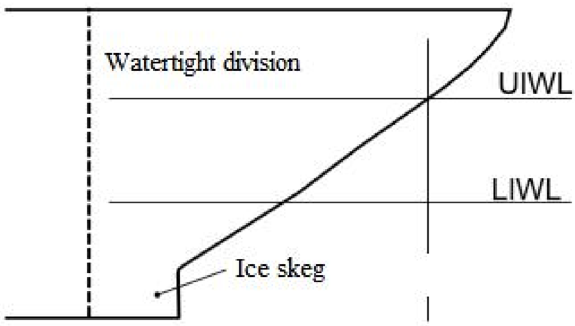


Figure 1.6.1.3 Ice skeg

1.6.2 Stability

1.6.2.1 Ship stability is to comply with the requirements of Section 9 Intact Stability and Section 10 Damage Stability, Chapter 1, PART TWO, Section 4 Stability, Chapter 23, PART EIGHT of CCS Rules for Classification of Sea-going Steel Ships and Chapter 4 Subdivision and Stability, Part I-A of IMO International Code for Ships Operating in Polar Waters.

1.6.2.2 The loading conditions under icing conditions are to comply with the residual stability criteria specified by the ship.

1.6.3 Watertight and weathertight integrity

1.6.3.1 Ship tightness is to comply with Section 5 Watertight and Weathertight Integrity, Chapter 23, PART EIGHT of CCS Rules for Classification of Sea-going Steel Ships and Chapter 5 Watertight and Weathertight Integrity, Part I-A of IMO International Code for Ships Operating in Polar Waters.

1.6.4 Life-saving appliances

1.6.4.1 Life-saving appliances are to be fully operable at the design service temperature (DST).

1.6.4.2 Lifeboats are to be of totally enclosed type. Rescue boats are to be rigid rescue boats.

1.6.4.3 Life-saving appliances are to comply with the following requirements in addition to the relevant requirements of Chapter III of SOLAS or the Code of Safety for Special Purpose Ships, 2008:

(1) Relevant requirements of Chapter 8, Part I-A and Section 9, Part I-B of IMO International Code for Ships Operating in Polar Waters;

(2) Relevant requirements of Chapter 23, PART EIGHT of CCS Rules for Classification of Sea-going Steel Ships;

(3) Relevant requirements of IMO MSC.1/Circ.1614/Rev.1.

1.6.5 Towing arrangement

1.6.5.1 For an icebreaker providing escort services, the towing arrangement is to be made, such as a recessed area at the stern, two fairleads, and two mooring bollards. The stern plating and frame are to be reinforced to consider collisions with the escorted vessel, and the arrangement of propulsion and steering gears is also to be considered to avoid collisions with the bulbous bow of the escorted vessel.

Section 2 STRUCTURAL REQUIREMENTS

2.1 Hull areas

2.1.1 The hull of all icebreaker 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 vertically into the Bottom, Lower and Icebelt regions. The extent of each hull area is illustrated in Figure 2.2.1 and complies with the provisions of 2.1.2 to 2.1.5.

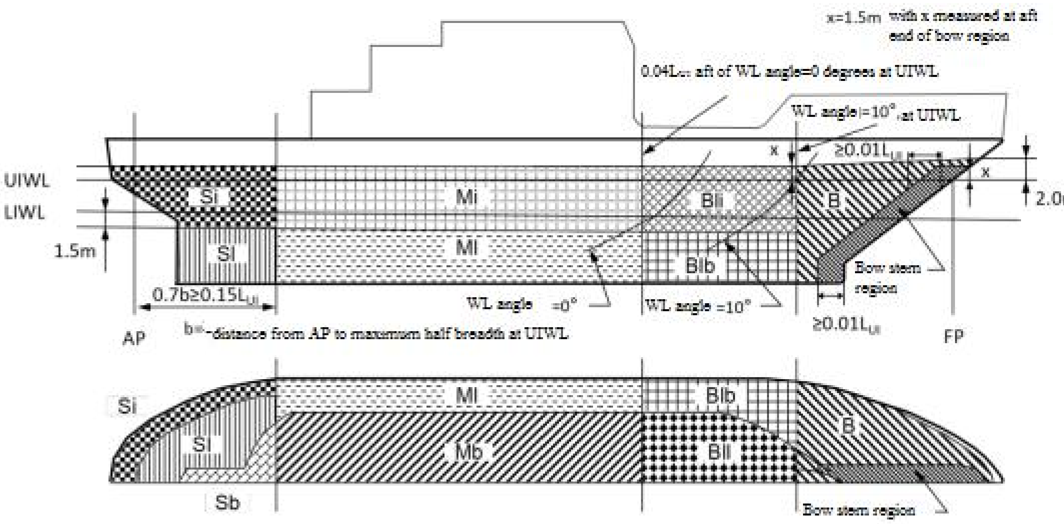


Figure 2.2.1 Extent of Hull Areas

2.1.2 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. However, 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).

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

2.1.4 If a ship is intended to operate astern in ice regions, the aft section of the ship is subject to special consideration and is to be approved by CCS.

2.1.5 Figure 2.2.1 notwithstanding, the forward boundary of the stern region is to be at least $0.04L_{UI}$ forward of the section where the parallel ship side at the upper ice waterline (UIWL) ends.

2.2 Design ice loads

2.2.1 General

2.2.1.1 For icebreakers, a glancing impact on the bow is the design scenario for determining the scantlings required to resist ice loads.

2.2.1.2 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).

2.2.1.3 Within the bow area of icebreakers, 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).

2.2.1.4 In ice-strengthened areas other than those specified in 2.2.1.3, 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.

2.2.1.5 Design ice forces calculated according to 2.2.2.2(3) are applicable for bow forms where the buttock angle γ at the stem is positive and less than 80°, and the normal frame angle β' at the centre of the foremost sub-region, as defined in 2.2.2.2(1), is greater than 10°.

2.2.1.6 For ships with bow forms other than those defined in 2.2.1.5, design forces are to be specially considered by CCS. Methods in Appendix 1 may be adopted.

2.2.1.7 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. Refer to the calculation methods in 4.7, Section 4 for the acceleration calculation.

2.2.2 Glancing impact load characteristics

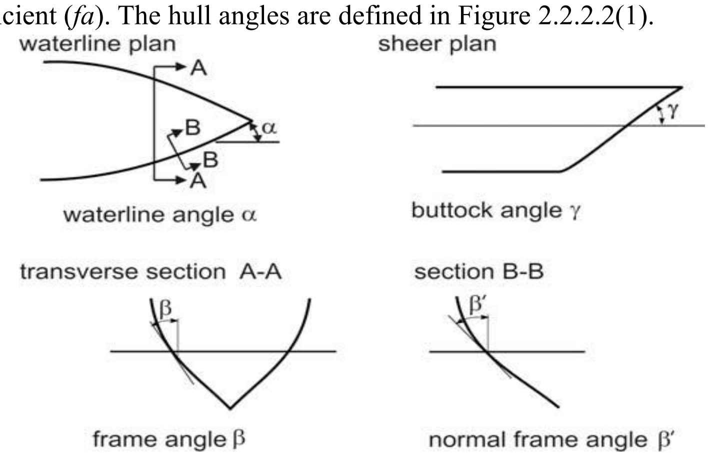
2.2.2.1 The parameters defining the glancing impact load characteristics are reflected in the class factors listed in Table 2.2.2.1。

Class Factors **Table 2.2.2.1**

Icebreaker*	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

2.2.2.2 Bow area

(1) 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 2.2.2.2(1).



Notes: β' — normal frame angle at upper ice waterline [°];

α — upper ice waterline angle [°];

γ — buttock angle at upper ice waterline (angle of buttock line measured from horizontal) [°];

$\tan(\beta) = \tan(\alpha)/\tan(\gamma)$;

$\tan(\beta') = \tan(\beta)\cdot\cos(\alpha)$

Figure 2.2.2.2(1) Definition of Hull Angles

(2) 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).

(3) The bow area load characteristics for bow forms are determined as follows:

① Shape coefficient, fa_i, is to be taken as:

$$fa_i = \min(fa_{i,1}, fa_{i,2}, fa_{i,3})$$

where: $$fa_{i,1} = \frac{\left[0.097\text{-}0.68\left(\frac{x}{\mathrm{L}_{\mathrm{UI}}} - 0.15\right)^2\right]\alpha_i}{\left(\beta_i'\right)^{0.5}};$$

$$fa_{i,2} = \left[\frac{1.2CF_F}{\sin\left(\beta_i'\right)\cdot CF_C \cdot D_{\mathrm{UI}}^{0.64}}\right];$$

$$fa_{i,3} = 0.60;$$

② Force, F_i:

$$F_i = fa_i \cdot CF_C \cdot D_{UI}^{0.64} \qquad \text{MN}$$

③ Load patch aspect ratio, AR_i:

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

④ Line load, Q_i:

$$Q_i = F_i^{0.61} \cdot \frac{CF_D}{AR_i^{0.35}} \qquad \text{MN/m}$$

⑤ Pressure, P_i:

$$P_i = F_i^{0.22} \cdot CF_D^2 \cdot AR_i^{0.3} \qquad \text{MPa}$$

where: i — sub-region considered;

L_{UI} — length, in m, as defined in 1.2;

x — distance from the fore side of the stem at the intersection with the upper ice waterline (UIWL) to station under consideration, in m;

α — waterline angle [°], see Figure 2.2.2.2(1);

β' — normal frame angle [°], Figure 2.2.2.2(1);

D_{UI} — displacement as defined in 1.2, in kt, not to be taken less than 5 kt;

CF_C — crushing failure class factor from Table 2.2.2.1;

CF_F — flexural failure class factor from Table 2.2.2.1;

CF_D — load patch dimensions class factor from Table 2.2.2.1.

2.2.2.3 Hull areas other than the bow

(1) 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 calculated as follows:

① Force, F_{NonBow}:

$$F_{NonBow} = 0.36CF_C \cdot DF \qquad \text{MN}$$

② Line load, Q_{NonBow}:

$$Q_{NonBow} = 0.639(F_{NonBow})^{0.61} \cdot CF_D \qquad \text{MN/m}$$

where: CF_C — crushing force class factor from Table 2.2.2.1;

DF — ship displacement factor, taken as follows:

$D_{UI}^{0.64}$ if $D_{UI} \leq CF_{DIS}$

$CF_{DIS}^{0.64} + 0.10(D_{UI} - CF_{DIS})$ if $D_{UI} > CF_{DIS}$

D_{UI} — displacement as defined in 1.2, in kt, not to be taken less than 10 kt;

CF_{DIS} — displacement class factor from Table 2.2.2.1;

CF_D — load patch dimensions class factor from Table 2.2.2.1.

2.2.3 Design load patch

2.2.3.1 In the bow area, the design load patch has dimensions of width, w_{Bow}, and height, b_{Bow}, defined as follows:

$$w_{Bow} = F_{Bow} / Q_{Bow} \quad \text{m}$$

$$b_{Bow} = Q_{Bow} / P_{Bow} \quad \text{m}$$

where: F_{Bow} — maximum F_i in the bow area, in MN;

Q_{Bow} — maximum Q_i in the bow area, in MN/m;

P_{Bow} — maximum P_i in the bow area, in MPa

2.2.3.2 In hull areas other than those covered by 2.2.3.1, the design load patch has dimensions of width, w_{Bow}, and height, b_{Bow}, defined as follows:

$$w_{NonBow} = F_{NonBow} / Q_{NonBow} \quad \text{m}$$

$$b_{NonBow} = w_{NonBow} / 3.6 \quad \text{m}$$

where: F_{NonBow} — force determined using 2.2.2.3(1)①, in MN;

Q_{NonBow} — line load determined using 2.2.2.3(1)②, in MN/m.

2.2.4 Pressure within the design load patch

2.2.4.1 The average pressure, P_{avg}, within a design load patch is determined as follows:

$$P_{avg} = F / (b \cdot w) \quad \text{MPa}$$

where: F — F_{Bow} or F_{NonBow} as appropriate for the hull area under consideration, in MN;

b — b_{Bow} or b_{NonBow} as appropriate for the hull area under consideration, in m;

w — w_{Bow} or w_{NonBow} as appropriate for the hull area under consideration, in m.

2.2.4.2 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 2.2.4.2 are used to account for the pressure concentration on localized structural members.

Peak Pressure Factors **Table 2.2.4.2**

Structural member		Peak pressure factor (PPF_i)
Plating	Transversely-framed	$PPF_p = (1.8 - s) \geq 1.2$
	Longitudinally-framed	$PPF_p = (2.2 - 1.2s) \geq 1.5$
Frames in 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		$PPF_s = 1.0$, when $S_w \geq 0.5w$, $PPF_s = 2.0 - 2.0S_w/w$, when $S_w < (0.5w)$

where: s — frame or longitudinal spacing, in m;

S_w — web frame spacing, in m;

w — ice load patch width, in m.

2.2.5 Hull area factors

2.2.5.1 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 2.2.5.1.

2.2.5.2 Due to their increased maneuverability, ships having propulsion arrangements with azimuthing (geared and podded) thrusters are to have factors *AF* listed in Table 2.2.5.2.

2.2.5.3 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.

Hull Area Factors **Table 2.2.5.1**

Hull Area		Area	Icebreaker*	
			PC1	PC2
Bow (*B*)	All	B	1.00	1.00
Bow Intermediate (*BI*)	Ice belt	BI_i	0.90	0.85
	Lower	BI_l	0.70	0.65
	Bottom	BI_b	0.55	0.50
Midbody (*M*)	Ice belt	M_i	0.70	0.65
	Lower	M_l	0.50	0.45
	Bottom	M_b	0.30	0.30
Stern (*S*)	Ice belt	S_i(1)	0.95	0.90
	Lower	S_l(1)	0.55	0.50
	Bottom	S_b	0.35	0.30

Note: (1) If supported by data, a smaller hull area factor is acceptable, but it is not be less than the corresponding polar class hull area factor.

Hull Area Factors **Table 2.2.5.2**

Hull Area		Area	Icebreaker*	
			PC1	PC2
Bow (*B*)	All	B	1.00	1.00
Bow Intermediate (*BI*)	Ice belt	BI_i	0.90	0.85
	Lower	BI_l	0.70	0.65
	Bottom	BI_b	0.55	0.50
Midbody (*M*)	Ice belt	M_i	0.70	0.65
	Lower	M_l	0.50	0.45
	Bottom	M_b	0.30	0.30
Stern (*S*)	Ice belt	S_i(1)	0.95	0.90
	Lower	S_l(1)	0.70	0.65
	Bottom	S_b	0.35	0.30

Note: (1) If supported by data, a smaller hull area factor is acceptable, but it is not be less than the corresponding polar class hull area factor.

2.3 Hull structures

2.3.1 Shell plates

2.3.1.1 The required minimum shell plate thickness, *t*, is given by:

$$t = t_{net} + t_s \qquad \text{mm}$$

where: t_{net} — plate thickness required to resist ice loads according to 2.3.1.2, in mm;

t_s — corrosion and abrasion allowance according to 2.3.9, in mm.

2.3.1.2 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 ($\Omega \geq 70°$), and all bottom plating, i.e. plating in hull areas B_{Ib}, M_b and S_b, the net thickness is given by:

$$t_{net} = 500s[(AF \cdot PPF_P \cdot P_{avg}) / \mathrm{R}_{eH}]^{0.5} / [1 + s / (2b)] \quad \text{mm}$$

In the case of longitudinally-framed plating ($\Omega \geq 20°$), when $b \geq$ s, the net thickness is given by:

$$t_{net} = 500s[(AF \cdot PPF_P \cdot P_{avg}) / R_{eH}]^{0.5} / [1 + s / (2l)] \quad \text{mm}$$

In the case of longitudinally-framed plating ($\Omega \leq 20°$), when $b < s$, the net thickness is given by:

$$t_{net} = 500s[(AF \cdot PPF_P \cdot P_{avg}) / R_{eH}]^{0.5} \cdot [2b / s - (b / s)^2]^{0.5} / [1 + s / (2l)] \quad \text{mm}$$

In the case of obliquely-framed plating ($70° > \Omega > 20°$), linear interpolation is to be used.

where: Ω — smallest angle between the chord of the waterline and the line of the first level framing, [°], illustrated in Figure 2.3.1.2;

s — transverse frame spacing in transversely-framed ships or longitudinal frame spacing in longitudinally-framed ships, in m;

AF — hull area factor from Table 2.2.5.1 or Table 2.2.5.2;

PPF_p — peak pressure factor from Table 2.2.4.2;

P_{avg} — average patch pressure according to 2.2.4.1, in MPa;

R_{eH} — yield stress of the material, in N/mm^2;

b — height of design load patch, in 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, in mm, i.e. equal to the frame span as given in 2.3.2.5, but not reduced for any fitted end brackets. 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.

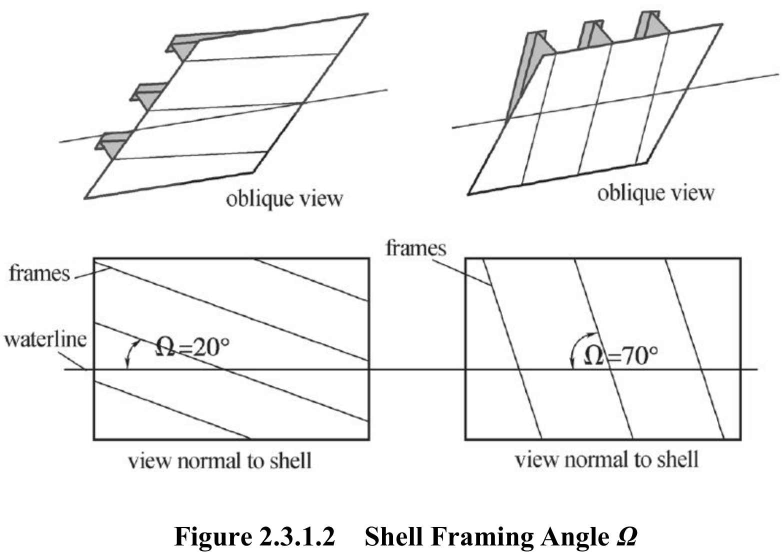


Figure 2.3.1.2 Shell Framing Angle Ω

2.3.2 Framing – General

2.3.2.1 Framing members of icebreakers are to be designed to withstand the ice loads defined in 2.2.

2.3.2.2 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 2.2.1.

2.3.2.3 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.

2.3.2.4 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 relevant requirements of Chapter 1, PART TWO of CCS Rules for Classification of Sea-going Steel Ships.

2.3.2.5 For the effective span of a framing member, see the requirements of 1.2.3, Section 2, Chapter 1, PART TWO of CCS Rules for Classification of Sea-going Steel Ships.

2.3.2.6 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.

2.3.2.7 The actual net effective shear area, A_w, of a local frame is given by:

$$A_w = h t_{wn} sin\varphi_w / 100 \qquad \text{cm}^2$$

where: h — height of stiffener, in mm, see Figure 2.3.2.7;

φ_w — smallest angle between shell plate and stiffener web, [°], measured at the midspan of the stiffener, see Figure 2.3.2.7. The angle φ_w may be taken as 90° provided the smallest angle is not less than 75°.

t_{wn} — net web thickness, in mm:

$t_{wn} = t_w - t_c$

where: t_w —as-built web thickness, in mm, see Figure 2.3.2.7;

t_c —corrosion deduction to be subtracted from the web and flange thickness, in mm, as specified in 2.3.9.3;

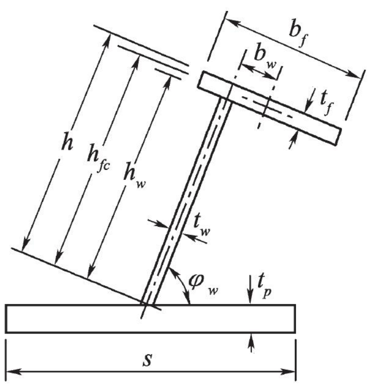


Figure 2.3.2.7 Stiffener Geometry

2.3.2.8 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:

$$Z_p = (A_{pn} t_{pn} / 20) + \frac{h_w^2 t_{wn} \sin\varphi_w}{2000} + A_{fn}(h_{fc}\sin\varphi_w - b_w\cos\varphi_w)/10 \qquad \text{cm}^3$$

where: h, t_{wn}, t_c and φ_w — as given in 2.3.2.7, s — as given in 2.3.1.2;

A_{pn} — net cross-sectional area of attached plate, in cm^2;

t_{pn} — net thickness of fitted shell plate, in mm (complying with t_{net} as required by 2.3.1.2);

h_w — height of local frame web, in mm, see Figure 2.3.2.7;

A_{fn} — net cross-sectional area of local frame flange, in cm^2;

h_{fc} — height of local frame measured to center of the flange area, in mm, see Figure 2.3.2.7;

b_w — distance from mid thickness plane of local frame web to the center of the flange area, in mm, see Figure 2.3.2.7.

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 a distance z_{na} above the attached shell plate, given by:

$$z_{na} = \left(100A_{fn} + h_w t_{wn} - 1000 t_{pn} s\right) / \left(2t_{wn}\right) \qquad \text{mm}$$

and the net effective plastic section modulus, Z_p, of a transverse or longitudinal frame is given by:

$$Z_p = t_{pn} \cdot s \cdot (z_{na} + \frac{t_{pn}}{2})\sin\varphi_w + \left\{ \begin{array}{l} \dfrac{[(h_w - z_{na})^2 + z_{na}^2] t_{wn} \sin\varphi_w}{2000} \\ + A_{fn}[(h_{fc} - z_{na})\sin\varphi_w - b_w\cos\varphi_w]/10 \end{array} \right\} \qquad \text{cm}^3$$

2.3.2.9 In the case of oblique framing arrangement (70° > Ω > 20°, where Ω is defined as given in 2.3.1.2), linear interpolation is to be used.

2.3.3 Framing – Local frames in bottom structures and in transversely-framed side structures

2.3.3.1 The local frames in bottom structures (i.e. hull areas B_{Ib}, M_b and S_b) and in transversely-framed side structures (i.e. hull areas B_{Ib}, M_b and S_b) 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 is to be applied with the dimension (b) parallel with the frame direction.

2.3.3.2 The actual net effective shear area of the frame, A_w, as defined in 2.3.2.7, is to comply with the following condition: $A_w \geq A_t$, where:

$$A_t = 100^2 \cdot 0.5LL \cdot s\left(AF \cdot PPF_t \cdot P_{avg}\right) / \left(0.577R_{eH}\right) \qquad \text{cm}^2$$

where: LL — length of loaded portion of span, in m, to be taken as the lesser of a and b;

a — local frame span as defined in 2.3.2.5, in m;

b — height of design ice load patch according to 2.1.3.1 or 2.1.3.2, in m;

s — spacing of local frame, in m;

AF — hull area factor from Table 2.1.5.2 or Table 2.1.5.3;

PPF_t — peak pressure factor, PPF_t or PPF_s as appropriate, from Table 2.1.4.2;

P_{avg} — average pressure within load patch according to 2.2.4.1, in MPa;

R_{eH} — yield stress of the material, in N/mm^2.

2.3.3.3 The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in 2.3.2.8, is to comply with the following condition: $Z_p \geq Z_p$, where:

$$Z_{pt} = 100^3 \cdot LL \cdot Y \cdot s\left(AF \cdot PPF_t \cdot P_{avg}\right)a \cdot A_1 / \ \left(4R_{eH}\right) \qquad \text{cm}^3$$

where: AF, PPF_t, P_{avg}, LL, s, a and R_{eH} — as given in 2.3.3.2;

Y = 1 − 0.5(LL/a);

A_1 — maximum of:

$A_{1A} = 1/\{1+j/2+(k_w j/2)\,[(1-a_1^2)^{0.5}-1]\}$

$A_{1B} = [1-1/(2a_1 \cdot Y)]/(0.275+1.44k_z^{0.7})$

where:

j = 1 for a local frame with one simple support outside the ice-strengthened areas;

= 2 for a local frame without any simple supports;

$$a_1 = A_t / A_w$$

where:

A_t — minimum shear area of the local frame as given in 2.3.3.2, in cm^2;

A_w — effective net shear area of the local frame (calculated according to 2.3.2.7), in cm^2;

$k_w = 1/(1+2A_{fn}/A_w)$, with A_{fn} as given in 2.3.2.8;

k_z — z_p / Z_p, in general;

= 0.0, when the frame is arranged with end bracket;

where:

z_p — sum of individual plastic section moduli of flange and shell plate as fitted, cm^3;

$$= (b_f\ t_{fn}^2/4 + b_{eff}\ t_{pn}^2/4)/1000$$

where:

b_f — flange breadth, in mm, see Figure 2.3.3.7;

t_{fn} — net flange thickness, in mm, see Figure 2.3.3.7;

= $t_f - t_c$ (t_c as given in 2.3.3.7)

where:

t_f — as-built flange thickness, in mm, see Figure 2.3.3.7;

t_{pn} — the fitted net shell plate thickness, in mm, (not to be less than as t_{net} given in 2.3.1);

b_{eff} — effective width of shell plate flange, in mm:

= 500 s;

Z_p — net effective plastic section modulus of the local frame (calculated according to 2.3.2.8), in cm^3.

2.3.4 Framing – Longitudinal local frames in side structures

2.3.4.1 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.

2.3.4.2 The actual net effective shear area of the frame, A_w, as defined in 2.3.2.7, is to comply with the following condition: $A_w \geq A_L$, where:

$$A_L = 100^2 (AF \cdot PPF_s \cdot P_{avg}) 0.5 b_1 \cdot a / \left(0.577 R_{eH}\right) \quad \text{cm}^2$$

where: AF — hull area factor from Table 2.1.5.2 or Table 2.1.5.3;

PPF_s — peak pressure factor from Table 2.1.4.2;

P_{avg} — average pressure within load patch according to 2.1.4, in MPa;

b_1 $= k_0 b_2$, in m

where: $k_0 = 1 - 0.3/b'$;

where: $b' = b/s$

where: b — height of design ice load patch from 2.1.3.1 or 2.1.3.2, in m;

s — spacing of longitudinal frames, in m;

$b_2 = b(1 - 0.25b')$, in m, if $b' < 2$;

$= s$, in m, if $b' \geq 2$;

a — effective span of longitudinal local frame as given in 2.3.2.5, in m;

R_{eH} — yield stress of the material, in N/mm^2.

2.3.4.3 The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in 2.3.2.8, is to comply with the following condition: $Z_p \geq Z_{pL}$, where:

$$Z_{pL} = 100^3 (AF \cdot PPF_S \cdot P_{avg}) b_1 \cdot a^2 \cdot A_4 / (8 R_{eH}) \quad \text{cm}^3$$

where: AF, PPF_s, P_{avg}, b_1, a and R_{eH} — as given in 2.3.4.2;

$$A_4 = 1/\{2 + k_{wl}[(1 - a_4^2)^{0.5} - 1]\};$$

$$a_4 = A_L / A_w$$

where: A_L — minimum shear area for longitudinal, in cm^2, as given in 2.3.4.2;

A_w — net effective shear area of longitudinal (calculated according to 2.3.2.7), in cm^2;

$k_{wl} = 1/(1 + 2A_{fn}/A_w)$, with A_{fn} as given in 2.3.2.8.

2.3.5 Framing – Web frame and load-carrying stringers

2.3.5.1 Web frames and load-carrying stringers are to be designed to withstand the ice load patch as defined in 2.2. The load patch is to be applied at locations where the capacity of these members under the combined effects of bending and shear is minimized.

2.3.5.2 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 2.5 are normally to be used.

2.3.5.3 The scantlings of web frames and load-carrying stringers are to meet the structural stability requirements of 2.3.6.

2.3.6 Framing – Structural stability

2.3.6.1 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: $h_w/t_{wn} \le 282/(R_{eH})^{0.5}$

For bulb, tee and angle sections: $h_w/t_{wn} \le 805/(R_{eH})^{0.5}$

where: h_w — web height;

t_{wn} — net web thickness;

R_{eH} — yield stress of the material, in N/mm^2.

2.3.6.2 Framing members for which it is not practicable to meet the requirements of 2.3.6.1 (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:

$$t_{wn} = 2.63 \times 10^{-3} c_1 \sqrt{R_{eH} / (5.34 + 4(c_1 / c_2)^2)} \quad \text{mm}$$

where: c_1 $= h_w - 0.8h$, in mm;

where: h_w—web height of stringer / web frame, in mm, see Figure 2.3.6.2;

h — height of framing member penetrating the member under consideration (0 if no such framing member), in mm, see Figure 2.3.6.2;

c_2 — spacing between supporting structure oriented perpendicular to the member under consideration, in mm, see Figure 2.3.6.2;

R_{eH} — yield stress of the material, in N/mm^2.

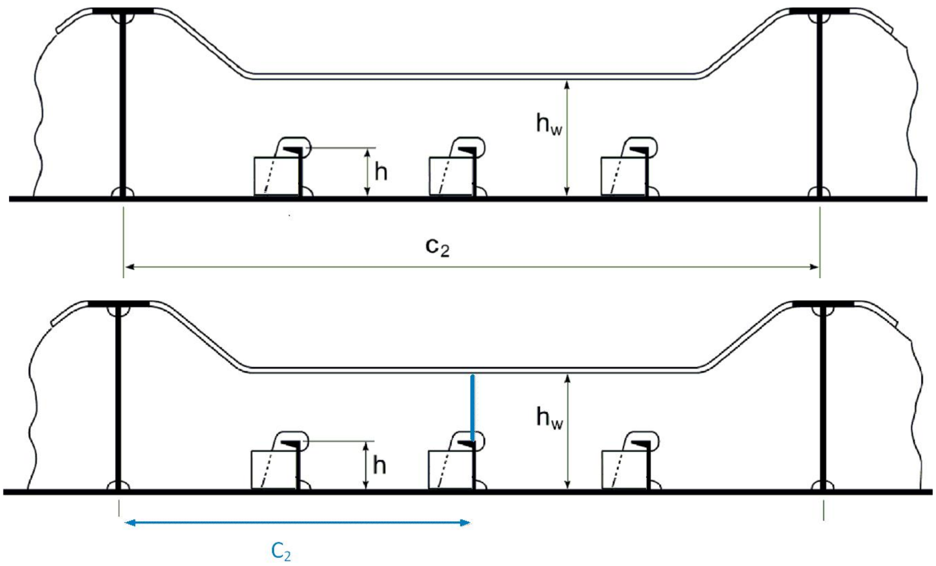


Figure 2.3.6.2 Parameter Definition for Web Stiffening

2.3.6.3 In addition, the following is to be satisfied:

$$t_{wn} \ge 0.35 t_{pn} \left(R_{eH} / 235\right)^{0.5}$$

where: R_{eH} — yield stress of the shell plate in way of the framing member, in N/mm^2;

t_{wn} — net thickness of the web, in mm;

t_{pn} — net thickness of the shell plate in way the framing member, in mm.

2.3.6.4 To prevent local flange buckling of welded profiles, the following are to be satisfied:

(1) the flange width, b_f, in mm, is not to be less than five times the net thickness of the web, t_{wn};

(2) the flange outstand, b_{out}, in mm, is to meet the following requirement:

$$b_{out} / t_{fn} \leq 155 / \left(R_{eH}\right)^{0.5}$$

where: t_{fn}— net thickness of flange, in mm;

R_{eH}— yield stress of the material, in N/mm^2.

2.3.7 Plated Structures

2.3.7.1 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:

(1) web height of adjacent parallel web frame or stringer; or

(2) 2.5 times the depth of framing that intersects the plated structure.

2.3.7.2 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.

2.3.7.3 The stability of the plated structure is to adequately withstand the ice loads defined in 2.2.

2.3.8 Bow stem

2.3.8.1 The shell plates of the bow stem region are to be strengthened. The net thickness of the shell plates within the breadth range of at least 0.01 times the length of ship in the bow stem region as shown in Figure. 2.1.1 is not to be less than 1.15 times the net thickness of t_{net} calculated according to 2.3.1.

2.3.8.2 The bow stem region is to be designed to withstand the ice load plate defined in 2.2. The member dimensions are to be determined using the appropriate methods in 2.5.

2.3.9 Corrosion/abrasion additions and steel renewal

2.3.9.1 Effective protection against corrosion and ice-induced abrasion is recommended for all external surfaces of the shell plating for all icebreakers. Compound steel plates in ice belt areas may also be regarded as effective protection.

2.3.9.2 The values of corrosion/abrasion additions, t_s, to be used in determining the shell plate thickness for each icebreaker class are listed in Table 2.3.9.2。

Corrosion/Abrasion Additions for Shell Plating **Table 2.3.9.2**

Hull area	t_s(mm)	
	With effective protection	Without effective protection
Bow; bow intermediate ice belt	3.5	7.0
Bow intermediate lower; midbody & stern ice belt	2.5	5.0
Midbody & stern lower; bottom	2.0	4.0

2.3.9.3 Icebreakers 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.

2.3.9.4 Steel renewal for ice strengthened structures is required when the gauged thickness is less than t_{net} + 0.5 mm.

2.4 Longitudinal strength

2.4.1 Application

2.4.1.1 A ramming impact on the bow is the design scenario for the evaluation of the longitudinal strength of the hull.

2.4.1.2 The longitudinal strength requirements given in this sub-section are not to be considered for ships with stem angle γ_{stem} equal to or larger than 80°.

2.4.1.3 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.

2.4.2 Design vertical ice force at the bow

2.4.2.1 The design vertical ice force at the bow, F_{IB}, is to be taken as:

$$F_{IB} = \min(F_{IB,1},\ F_{IB,2}) \qquad \text{MN}$$

where: $F_{IB,1} = 0.534 K_I^{0.15} [\sin(\gamma_{stem})]^{0.2} (D_{UI} \cdot K_h)^{0.5} CF_L$ MN;

$F_{IB,2} = 1.20 CF_F$ MN;

K_I——indentation parameter, $= K_f / K_h$;

(a) for the case of a blunt bow form:

$K_f = \left[2CB_{UI}^{\ 1-e_b} / (1+e_b)\right]^{0.9} \tan(\gamma_{stem})^{-0.9(1+e_b)}$;

(b) for the case of wedge bow form ($\alpha_{stem} < 80^\circ$), $e_b = 1$ and the above simplifies to:

$K_f = \left\{\tan(\alpha_{stem}) / [\tan(\gamma_{stem})]^2\right\}^{0.9}$;

$K_h = 0.01 A_{wp}$, MN/m;

CF_L ——longitudinal strength class factor from Table 2.1.2.1;

e_b ——bow shape exponent which best describes the waterplane (see Figures 2.4.2.1(1) and 2.4.2.1(2))

= 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 (UIWL) (buttock angle as per Figure 2.1.2.2(1) measured on the centerline);

α_{stem} ——waterline angle measured in way of the stem at the upper ice waterline (UIWL) (°) (see Figure 2.4.2.1(1));

B_{UI} ——moulded breadth corresponding to the upper ice waterline (UIWL), in m;

L_B ——bow length used in the equation $y = (B_{UI}/2)(x/L_B)^{e_b}$, in m, (see Figures 2.4.2.1(1) and 2.4.2.1(2));

D_{UI} ——displacement as defined in 2.1.2.3, in kt, not to be taken less than 10 kt;

A_{wp} ——waterplane area corresponding to the upper ice waterline (UIWL), in m^2;

CF_F ——flexural failure class factor from Table 2.1.2.1;

Where applicable, the data on which the draught is based is to be determined at the waterline corresponding to the calculated loading condition.

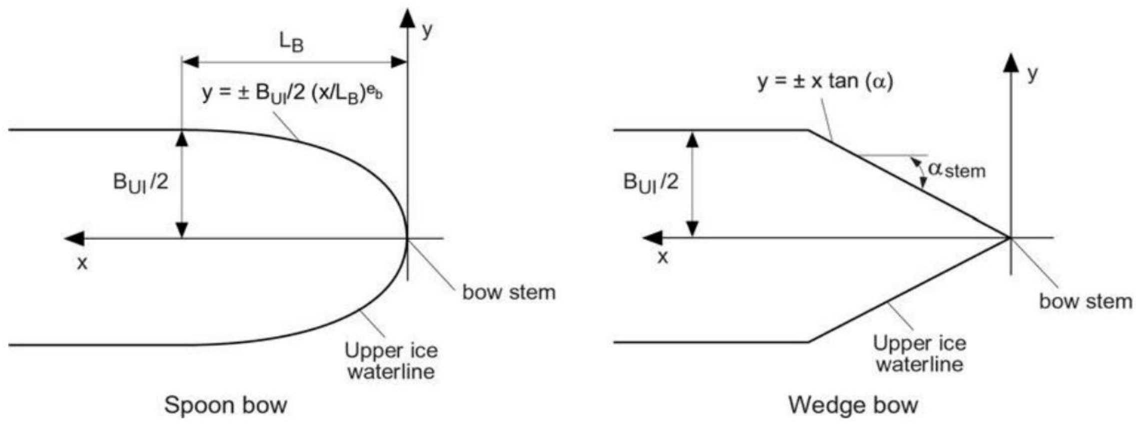


Figure 2.4.2.1(1)   Bow Shape Definition

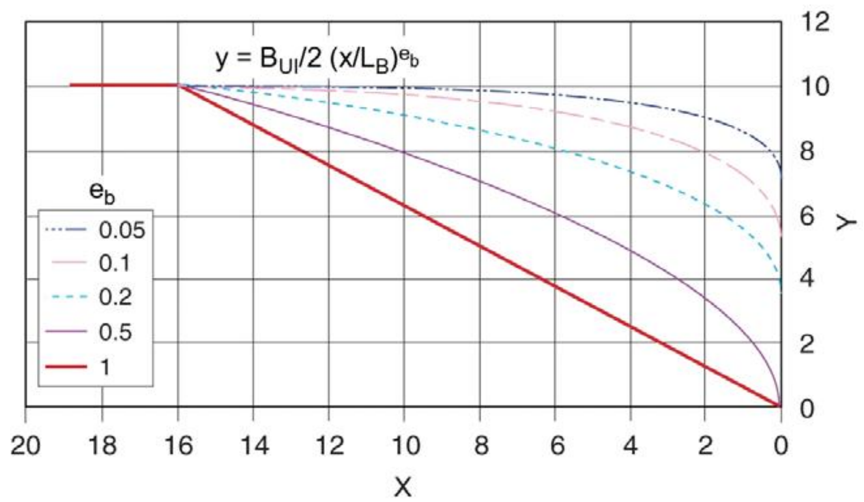


Figure 2.4.2.1(2)   Illustration of e_b Effect on the Bow Shape for B_{UI} = 20 and L_B =16

2.4.3   Design vertical shear force

2.4.3.1   The design vertical ice shear force, F_I, along the hull girder is to be taken as:

$$F_I = C_f F_{IB} \qquad \text{MN}$$

where: C_f — longitudinal distribution factor to be taken as follows:

(a) positive shear force

C_f = 0.0 between the aft end of L_{UI} and 0.6L_{UI} from aft;

C_f = 1.0 between 0.9L_{UI} from aft and the forward end of L_{UI};

(b) negative shear force

C_f = 0.0 at the aft end of L_{UI};

C_f = –0.5 between 0.2L_{UI} and 0.6L_{UI} from aft;

C_f = 0.0 between 0.8L_{UI} from aft and the forward end of L_{UI};

intermediate values are to be determined by linear interpolation.

2.4.3.2   The applied vertical shear stress, τ_a, is to be determined along the hull girder in a similar manner as in 2.2.6 of Section 2, Chapter 2 of PART TWO of CCS Rules for Classification of

Sea-going Steel Ships by substituting the design vertical ice shear force for the design vertical wave shear force.

2.4.4 Design vertical ice bending moment

2.4.4.1 The design vertical ice bending moment, M_I, along the hull girder is to be taken as:

$$M_I = 0.1 C_m L_{UI} [\sin(\gamma_{stem})]^{-0.2} F_{IB} \qquad \text{MN} \cdot \text{m}$$

where: L_{UI} — length, in m, as defined in 1.2.1;

γ_{stem} — as given in 2.4.2.1;

F_{IB} — design vertical ice force at the bow, in 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 $0.5L_{UI}$ and $0.7L_{UI}$ from aft;

C_m = 0.3 at $0.95L_{UI}$ from aft;

C_m = 0.0 at the forward end of L_{UI};

intermediate values are to be determined by linear interpolation.

2.4.4.2 The applied vertical bending stress, σ_a, is to be determined along the hull girder in a similar manner as in 2.2.5 of Section 2, Chapter 2 of PART TWO of CCS Rules for Classification of Sea-going Steel Ships, 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.

2.4.5 Longitudinal strength criteria

2.4.5.1 The strength criteria provided in Table 2.4.5.1 are to be satisfied. The design stress is not to exceed the permissible stress.

Longitudinal Strength Criteria **Table 2.4.5.1**

Failure mode	Applied stress	Permissible stress when $R_{eH}/\sigma_u \leq 0.7$	Permissible stress when $R_{eH}/\sigma_u > 0.7$
Tension	σ_a	ηR_{eH}	$0.41\eta(\sigma_u + R_{eH})$
Shear	τ_a	$\eta R_{eH} / (3)^{0.5}$	$0.41\eta(\sigma_u + R_{eH}) / (3)^{0.5}$
Buckling	σ_a	σ_c for plating and for web plating of stiffeners $\sigma_c / 1.1$ for stiffeners	
	τ_a	τ_c	

Notes: σ_a — applied vertical bending stress, in N/mm²;

τ_a — applied vertical shear stress, in N/mm²;

R_{eH} — yield stress of the material, in N/mm²;

σ_u — ultimate tensile strength of material, in N/mm²;

σ_c — critical buckling stress in compression, according to 2.2.7 of Section 2, Chapter 2 of PART TWO of CCS Rules for Classification of Sea-going Steel Ships, in N/mm²;

τ_c — critical buckling stress in compression, according to 2.2.7 of Section 2, Chapter 2 of PART TWO of CCS Rules for Classification of Sea-going Steel Ships, in N/mm²;

η = 0.6 for ships which are assigned the additional notation "Icebreaker".

2.5 Direct calculations

2.5.1 General

2.5.1.1 Direct calculations are not to be utilized as an alternative to the analytical procedures prescribed for the shell plating and local frame requirements given in 2.3.1, 2.3.3, and 2.3.4 of this Section.

2.5.1.2 Direct calculations are to be used for load carrying stringers and web frames forming part of a grillage system.

2.5.1.3 Where direct calculation is used to check the strength of structural systems, the load patch specified in 2.2 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 minimized. Special attention is to be paid to the shear capacity in way of lightening holes and cut-outs in way of intersecting members.

2.5.1.4 The strength evaluation of web frames and stringers may be performed based on linear or non-linear analysis. Recognized structural idealization and calculation methods are to be applied, but the detailed requirements are to be specified by CCS. In the strength evaluation, the guidance given in 2.5.1.5 and 2.5.1.6 may generally be considered. For detailed requirements, see CCS *Guidelines for Direct Calculation of Structural Strength under Ice Loads*.

2.5.1.5 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 CCS;

(2) nominal shear stresses in member web plates to be less than $R_{eH}/\sqrt{3}$;

(3) nominal von Mises stresses in member flanges to be less than $1.15R_{eH}$.

2.5.1.6 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 CCS.

2.5.2 Supporting structure for deck equipment

2.5.2.1 When checking the supporting structure for deck equipment, the inertia load is also to be considered. See 4.7, Section 4 for the calculation of the acceleration.

2.6 Local details

2.6.1 General

2.6.1.1 For the purpose of transferring ice-induced loads to supporting structure (bending moments and shear forces), local design details are to be appropriate.

2.6.1.2 The loads carried by a member in way of cut-outs are to maintain stability. Where necessary, the structure is to be stiffened.

2.7 Appendages

2.7.1 General

2.7.1.1 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.
2.7.1.2 Load definition and response criteria are to be determined by practical situations.

2.8 Materials and welding

2.8.1 Materials

2.8.1.1 Steel grades of plating for hull structures are to be not lower than those given in Table 2.8.1.4 based on the as-built thickness, the Icebreaker Class and the material class of structural members according to 2.8.1.2.

Material Classes for Structural Members of Icebreaker Table 2.8.1.1

Structural Members	Material Class
Shell plating within the bow and bow intermediate ice belt hull areas (*B*, BI_i)	II
All weather and sea exposed SECONDARY and PRIMARY, as defined in Table 1.3.2.2(1) of PART TWO of the Rules, structural members outside 0.4 L_{UI} 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 shell 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 Table 1.3.2.2(1) of PART TWO of CCS Rules for Classification of Sea-going Steel Ships, structural members within 0.2 L_{UI} from FP	II

2.8.1.2 Material classes specified in Table 1.3.2.2(1) of Section 3, Chapter 1 of PART TWO of CCS Rules for Classification of Sea-going Steel Ships are applicable to icebreakers 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 2.8.1.1. Where the material classes in Table 2.8.1.1 and those in Table 1.3.2.2(1) of Section 3, Chapter 1 of PART TWO of CCS Rules for Classification of Sea-going Steel Ships differ, the higher material class is to be applied.
2.8.1.3 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 2.8.1.3, are to be obtained from Table 1.3.2.2(6) and Table 1.3.2.2(7) of Section 3, Chapter 1 of PART TWO of CCS Rules for Classification of Sea-going Steel Ships based on the material class for structural members in Table 2.8.1.1 above.

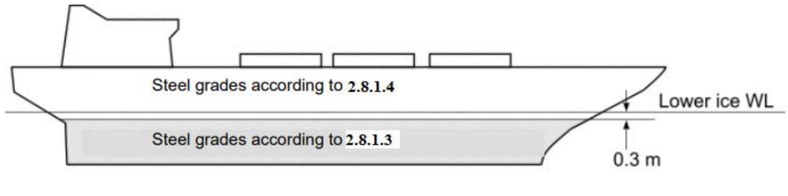


Figure 2.8.1.3 Steel Grade Requirements for Submerged and Weather Exposed Shell Plating

2.8.1.4 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 (LIWL), as shown in Figure 2.8.1.3, are to be not lower than that given in Table 2.8.1.4.

Steel Grades for Weather Exposed Plating **Table 2.8.1.4**

Thickness *t* [mm]	Material Class I		Material Class II		Material Class III	
	MS	HT	MS	HT	MS	HT
$t \le 10$	B	AH	B	AH	E	EH
$10 < t \le 15$	B	AH	D	DH	E	EH
$15 < t \le 20$	D	DH	D	DH	E	EH
$20 < t \le 25$	D	DH	D	DH	E	EH
$25 < t \le 30$	D	DH	E	EH[2]	E	EH
$30 < t \le 35$	D	DH	E	EH	E	EH
$35 < t \le 40$	D	DH	E	EH	F	FH
$40 < t \le 45$	E	EH	E	EH	F	FH
$45 < t \le 50$	E	EH	E	EH	F	FH

Notes:

(1) 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.

(2) 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.

2.8.1.5 High strength steel with thickness over 50 mm may be used subject to CCS special agreement. The steel grades are not to be lower than the requirements for 50-mm thick plate in Table 2.8.1.4.

2.8.1.6 High strength steels with suffix of ARC-M(X) for icebreaker are to comply with the requirements of Appendix 8 High Strength Steels for Polar Ship of the Guidelines for Polar Ship, and the steel grades are determined according to Table 2.8.1.4. When super high strength steels suffixed with ARC-M(X) and the minimum yield strength of which is ≥420MPa are selected, CTOD values are to comply with the requirements of Tables 2.8.1.6(1), 2.8.1.6(2), 2.8.1.6(3) in addition to Appendix 8 of the Guidelines for Polar Ship.

Nil-ductility transition temperature (NDTT)(°C) **Table 2.8.1.6(1)**

Thickness (mm)	ARC-M(-20)	ARC-M(-30)	ARC-M(-40)	ARC-M(-50)	ARC-M(-60)
$16 \le t \le 25$	≤-25	≤-35	≤-45	≤-55	≤-65
$25 < t \le 30$	≤-35	≤-45	≤-55	≤-65	≤-75
$30 < t \le 40$	≤-40	≤-50	≤-60	≤-70	≤-80
$40 < t \le 50$	≤-45	≤-55	≤-65	≤-75	≤-85
$50 < t \le 100$	≤-50	≤-60	≤-70	≤-80	≤-90

Minimum average CTOD value for base metal for steel approval **Table 2.8.1.6(2)**

Thickness (mm)	H420	H460	H500	H550	H620	H690
16≤t≤20mm	0.10	0.15	0.15	0.15	0.20	0.20
20＜t≤30mm	0.15	0.20	0.20	0.20	0.20	0.25
30＜t≤40mm	0,20	0.20	0.20	0.25	0.25	0.30
40＜t≤50mm	0,20	0.25	0.25	0.25	0.25	0.30

50＜t≤70mm	0.25	0.30	0.30	0.30	0.30	0.35

Minimum average CTOD value for weld joint for steel approval **Table 2.8.1.6(3)**

Thickness (mm)	H420	H460	H500	H550	H620	H690
16≤t≤20mm	0.10	0.10	0.15	0.15	0.15	0.20
20＜t≤30mm	0.15	0.15	0.15	0.15	0.20	0.20
30＜t≤40mm	0.15	0.15	0.15	0.15	020	0.20
40＜t≤50mm	0.15	0.15	0.15	0.20	0.20	0.25
50＜t≤70mm	0.20	0.20	0.20	0.25	0.25	0.30

2.8.2 Casting

2.8.2.1 Castings are to comply with the requirements of Chapter 6, PART ONE of CCS Rules for Materials and Welding and have specified properties consistent with the expected service temperature for the cast component.

2.8.3 Forgings

2.8.3.1 Forgings are to comply with the requirements of Chapter 5, PART ONE of CCS Rules for Materials and Welding, where the ramming impact test temperature is to be the design service temperature.

2.8.4 Clad steel plates

2.8.4.1 When clad stainless steel plates are used to reduce the maintenance of shell plating in ice belt regions, the requirements of 2.8.4.2~2.8.4.6 are to be complied with.

2.8.4.2 Clad stainless steel plates are to comply with the relevant provisions of Section 9, Chapter 3, PART ONE of CCS Rules for Materials and Welding.

2.8.4.3 The base material is to comply with the requirements of Table 2.8.1.4. Where high strength steel suffixed with ARC-M(X) is selected as the base material, the provisions for high strength steel suffixed with ARC-M(X) in Appendix 8 of CCS Guidelines for Polar Ship are to be complied with. The thickness of the base material is to be more than 10 mm.

2.8.4.4 When clad stainless steel plates are used as shell plating, the cladding metal is to consider the seawater corrosion resistance. The thickness of the cladding material is not to be less than 2 mm.

2.8.4.5 When clad stainless steel plates are used, appropriate cathodic protection is to be provided. Considering the low temperature and low salinity of the polar environment, the protection potential of the external current is usually increased or more sacrificial anodes are provided. The voltage of the external current is not be to increased excessively to avoid overprotection. The placement of sacrificial anodes is to take into account the possibility of ice ramming damage.

2.8.4.6 When clad stainless steel plates are used in some areas, effective anti-corrosion measures are to be taken to prevent galvanic corrosion in the connection area between the clad stainless steel plates and other materials.

2.8.5 Welding

2.8.5.1 All fillet welding within ice-strengthened areas is to be of the double continuous type. For locations where the double continuous welding is impractical due to space limitation, other welding procedures may be adopted subject to CCS agreement.

2.8.5.2 Continuity of strength is to be ensured at all structural connections.

2.8.5.3 When super high strength steels suffixed with ARC-M(X) and the minimum yield strength of which is ≥420MPa are selected, the welding procedures are to comply with the requirements of Table 2.8.5.3 in addition to Appendix 8 of the Guidelines for Polar Ship.

Minimum average CTOD value for welding procedure qualification tests

Table 2.8.5.3

Thickness (mm)	Average crack tip opening displacement (mm), minimum					
	H420	H460	H500	H550	H620	H690
16≤t≤20mm	0.15	0.15	0.15	0.15	0.20	0.20
20＜t≤30mm	0.15	0.15	0.15	0.15	0.20	0.20
30＜t≤40mm	0.15	0.15	0.15	0.15	0.20	0.20
40＜t≤50mm	0.15	0.15	0.15	0.20	0.20	0.25
50＜t≤70mm	0.20	0.20	0.20	0.25	0.25	0.30
70＜t≤80mm	0.20	0.25	0.25	0.30	0.30	0.35
80＜t≤100mm	0.20	0.25	0.25	0.30	0.30	0.35

Section 3 PROPULSION POWER

3.1 General

3.1.1 For ships applying for the class notation Icebreaker*, their propulsion machinery is to be able to continuously output sufficient power so as to maintain sufficient maneuverability in design ice conditions.

3.1.2 Icebreaking capability is to be validated from ice pond model test or full-scale navigation test in ice areas. Where valid documentation can be provided, validation in the form of numerical analysis or empirical formula is also accepted.

3.1.3 In the early stage of design or when the conditions of the ice pond model test are not yet available, the calculation method in 3.2 can be used to estimate.

3.2 Requirements of propulsion power

3.2.1 The propulsion power provided onboard is to be greater than the icebreaking power calculated according to 3.2.2.

3.2.2 The icebreaking power N_1 required for the ship is not to be less than the value calculated according to the following formula:

$$N_1 = 0.736 f_1 f_2 f_3 f_4 \left[240 B\, h_0 \left(1 + h_0 + 0.035 v^2\right) + 70\, S_c \sqrt{L}\right] \quad \text{kW}$$

where: B ——maximum breadth at UIWL, in m;

L ——length of the waterline at UIWL, in m;

f_1 ——factor, to be calculated according to the following formula and to be taken as not less than 1.0:

$$f_1 = \frac{1.2B}{\sqrt[3]{\Delta}}$$

where: Δ ——displacement of the ship when the draught is at UIWL, in t;

f_2 ——factor, if controllable pitch propellers are provided or electric propulsion are used, f_2 is to be taken as 0.9; if fixed pitch propellers are provided, f_2 is to be taken as 1.0;

f_3 ——factor, if bow stem angle at waterline ≤45°, f_3 is to be taken as 0.9; if bow stem angle at waterline >45°, f_3 is to be taken as 1.0;

f_4 ——factor, for single ore, f_4 is to be taken as 0.88; for double ores, f_4 is to be taken as 0.99; for triple ores, f_4 is to be taken as 1.06;

h_0 ——thickness of level ice, in m; see Table 3.2.1 for the recommended minimum icebreaking thickness;

v ——ship speed, in kn, when the thickness of ice to be broken is h_0, to be taken as not less than 2 kn;

S_c ——the thickness of snow covered on ice, in m, to be taken as not less than 0.3 m.

The Minimum Icebreaking Thickness for Ice Classes Table 3.2.1

Icebreaker*	PC1	PC2
Thickness of level ice, in m	3.0	2.4

Section 4 MACHINERY INSTALLATIONS

4.1 General requirements

4.1.1 The provisions of this Section apply to main propulsion, steering gear, emergency and auxiliary systems essential for the safety of the ship and the crew.

4.1.2 Systems subject to damage by freezing, are to be drainable.

4.1.3 Icebreakers are to have means provided to ensure sufficient ship operation in the case of propeller damage including the Controllable Pitch (CP) mechanism. Sufficient ship operation means that the ship is to 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.

4.1.4 Means are to be provided to free a stuck propeller by turning it in reverse direction. This is also to be possible for a propulsion plant intended for unidirectional rotation.

4.1.5 The propeller is to be fully submerged at the ship's LIWL.

4.2 Materials

4.2.1 Cast iron materials

4.2.1.1 Materials are to 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.

4.2.2 Materials exposed to sea water

4.2.2.1 Materials exposed to sea water, such as propeller blades, propeller hubs and cast thruster bodies are to have an elongation not less than 15% on a test specimen according to CCS Rules for Materials and Welding.

Charpy V-notch impact testing is to be carried out for materials other than bronze and austenitic steel. The tests are to 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 CCS Rules for Materials and Welding as applicable for ships with ice class notation, are also to be applied.

4.2.3 Materials exposed to sea water temperature

4.2.3.1 Charpy V-notch impact testing is to be carried out for materials other than bronze and austenitic steel. The tests are to 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 CCS Rules for Materials and Welding as applicable for ships with ice class notation, are also to be applied.

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 Figure 2.8.1.3, Section 2 of this Chapter.

4.2.4 Materials exposed to low air temperature

Materials of exposed machinery and foundations are to 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 Figure 2.8.1.3, Section 2 of this Chapter.

4.3 Definition of main parameters and loads

4.3.1 The symbols, names, and units of parameters used in this Section are as follows:

c —— chord length of blade section, m;

$c_{0.7}$ —— chord length of blade section at $0.7R$ propeller radius, in m;

$c_{LE0.8}$ —— leading edge portion of chord length of blade section at 0.8R propeller radius, in m;

$c_{TE0.8}$ —— trailing edge portion of chord length of blade section at 0.8R propeller radius, in m;

D —— propeller diameter, in m;

d —— external diameter of propeller hub (at propeller plane), in m;

D_{pin} —— diameter of shear pin, in m;

D_{limit} —— limit value for propeller diameter, in m;

EAR —— expanded blade area ratio;

F_b —— maximum backward blade force for the ship's service life (negative sign), in kN;

F_{ex} —— ultimate blade load resulting from blade failure through plastic bending, abbreviated as blade failure load, in kN;

F_f —— maximum forward blade force for the ship's service life (positive sign), in kN;

F_{ice} —— ice load, in kN;

$(F_{ice})_{max}$ —— maximum ice load for the ship's service life , in kN;

h_0 —— depth of the propeller centreline from lower ice waterline (LIWL), in m;

H_{ice} —— ice block dimension for propeller load definition, in m;

I —— equivalent mass moment of inertia of all parts on engine side of component under consideration , in kgm^2;

I_t —— equivalent mass moment of inertia of the whole propulsion system, in kgm^2;

$LIWL$ —— lower ice waterline, in m;

m —— slope for S-N curve in log/log scale;

M_{BL} —— blade bending moment, in kNm;

n —— propeller rotational speed, in r/s;

n_n —— nominal propeller rotational speed at MCR in free running condition, in r/s;

N —— number of ice load cycles;

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;

$P_{0.7}$ —— propeller pitch at 0.7R radius, in m;

$P_{0.7n}$ —— propeller pitch at 0.7R radius at MCR in free running condition, in m;

$P_{0.7b}$ —— propeller pitch at 0.7R radius at MCR in bollard condition, in m;

PCD —— pitch circle diameter, in m;

$Q(\varphi)$ —— torque, in kNm;

Q_{Amax} —— maximum response torque amplitude as a simulation result, in kNm;

Q_{emax} —— maximum engine torque, in kNm;

$Q_F(\varphi)$ —— Ice torque excitation for frequency domain calculations, in kNm;

Q_{fr} —— friction torque in pitching mechanism; reduction of spindle torque, in kNm;

Q_{max} —— maximum torque on the propeller resulting from propeller/ice interaction, in kNm;

Q_{motor} —— electric motor peak torque, in kNm;

Q_n —— nominal torque at MCR in free running condition, in kNm;

$Q_r(t)$ —— response torque along the propeller shaft line, in kNm;

Q_{peak} —— maximum of the response torque $Q_r(t)$, in kNm;

Q_{smax} —— maximum spindle torque of the blade for the ship's service life, in kNm;

Q_{sex} —— extreme spindle torque corresponding to the blade failure load F_{ex}, in kNm;

Q_{vib} —— Vibratory torque at considered component, taken from frequency domain open water TVC, in kNm;

r —— blade section radius, in m;

R —— propeller radius, in m;

$R_{p0.2}$ —— yield strength of the selected material (when the plastic strain is 0.2%), in MPa, to be selected according to CCS Rules for Materials and Welding;

R_m —— tensile strength of blade material, in MPa, to be selected according to CCS Rules for Materials and Welding;

S —— safety factor;

S_{fat} —— Safety factor for fatigue;

S_{ice} —— Ice strength index for blade ice force;

T —— Hydrodynamic propeller thrust in bollard condition, in kN;

T_b —— maximum backward propeller ice thrust for the ship's service life, in kN;

T_f —— maximum forward propeller ice thrust for the ship's service life, in kN;

T_n —— propeller thrust at MCR in free running condition, in kN;

T_r —— maximum response thrust along the shaft line, in kN;

T_{kmax} —— maximum torque capacity of flexible coupling, in kNm;

T_{kmax2} —— T_{kmax} at N= 1 load cycle, in kNm;

T_{kmax1} —— T_{kmax} at N= 5×10^4 load cycles, in kNm;

T_{kv} —— vibratory torque amplitude at N= 10^6 load cycles, in kNm;

ΔT_{kmax} —— maximum range of T_{kmax} at N=5×10^4 load cycles, in kNm;

t —— maximum blade section thickness, in m;

Z —— number of propeller blades;

z_{pin} —— number of shear pins;

α_i —— duration of propeller blade/ice interaction expressed in rotation angle, in °;

α_1 —— phase angle of the blade order excitation component for the propeller ice torque, in ° ;

α_2 —— phase angle of the blade order excitation component for the propeller ice torque, in ° ;

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;

σ_{exp} —— mean fatigue strength of blade material at 10^8 cycles to failure in sea water, in MPa;

σ_{fat} —— equivalent fatigue ice load stress amplitude for 10^8 stress cycles, in MPa;

σ_{fl} —— characteristic fatigue strength for blade material, in MPa;

σ_{ref1} —— reference stress, $\sigma_{ref1} = 0.6 \cdot R_{p0.2} + 0.4 \cdot R_m$, in MPa;

σ_{ref2} —— reference stress, $\sigma_{ref2} = 0.7 \cdot R_m$ or $\sigma_{ref2} = 0.6 \cdot R_{p0.2} + 0.4 \cdot R_m$, whichever is less, in MPa;

σ_{st} —— maximum stress resulting from F_b or F_f, in MPa;

$(\sigma_{ice})_{bmax}$ —— principal stress caused by the maximum backward propeller ice load, in MPa;

$(\sigma_{ice})_{fmax}$ —— principal stress caused by the maximum forward propeller ice load, in MPa;

$(\sigma_{ice})_{Amax}$ —— maximum ice load stress amplitude at the considered location on the blade, in MPa;

σ_{mean} —— mean stress, in MPa;

$(\sigma_{ice})_A(N)$ —— blade stress amplitude distribution, in MPa.

4.3.2 The definition and use of loads used in this Section are given in Table 4.3.2:

Definition and use of loads **Table 4.3.2**

Load	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 0.7*R* chord line. Ice contact pressure at leading edge is shown with small arrows. See Figure 4.3.2.	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 0.7*R* 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 0.8*R*.	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 is not to 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

Load	Definition	Use of the load in design process
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.

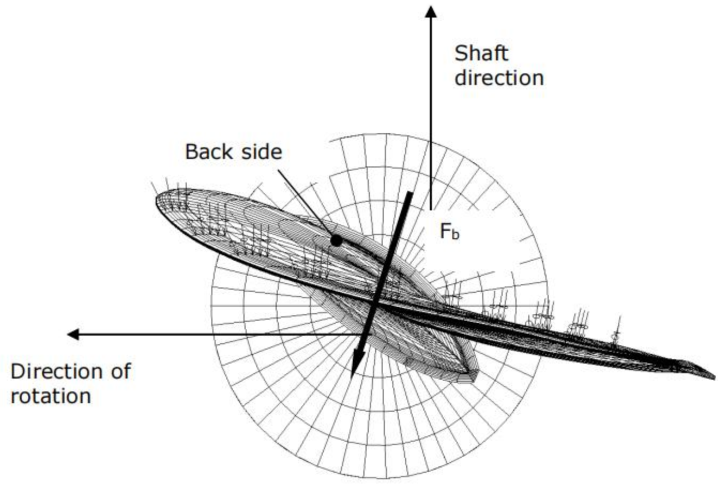


Figure 4.3.2 Direction of the backward blade force

4.4 Ice interaction loads

4.4.1 Propeller/ice interaction

The Rules cover open and ducted type propellers situated at the stern of a ship having controllable pitch or fixed pitch blades. Ice loads on bow-mounted propellers are to receive special consideration. 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. The 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 are to be estimated on a case by case basis, however are not included within the following section.

The loads given in section 4.4 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.

4.4.2 Ice class factors

The dimensions of the considered design ice block are $H_{ice} \times 2H_{ice} \times 3H_{ice}$. The design ice block H_{ice} and ice strength index S_{ice} used for the estimation of propeller ice loads are defined for each Ice class in Table 4.4.2 below.

Design ice block and ice strength index **Table 4.4.2**

Icebreaker*	H_{ice}[m]	S_{ice}[-]
PC1	4.0	1.2
PC2	3.5	1.1

4.4.3 Design ice loads for open propellers

(1) Maximum backward blade force F_b

When $D < D_{limit}$:

$$F_b = -27 C_1 S_{ice} \left(\frac{EAR}{Z}\right)^{0.3} (nD)^{0.7} D^2 \qquad \text{kN}$$

When $D \geq D_{limit}$:

$$F_b = -23 C_1 S_{ice} \left(\frac{EAR}{Z}\right)^{0.3} (nD)^{0.7} H_{ice}^{1.4} D \qquad \text{kN}$$

Where: $D_{limit} = 0.85 H_{ice}^{1.4}$;

C_1 —— coefficient, to be taken as 1.1 for a ship with icebreaker class notation;

S_{ice} and H_{ice} are selected in accordance with Table 4.4.2;

n —— 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), in r/s.

(2) Maximum forward blade force F_f

When $D < D_{limit}$:

$$F_f = 250 \frac{EAR}{Z} D^2 \qquad \text{kN}$$

When $D \geq D_{limit}$:

$$F_f = 500 \left(\frac{1}{1 - \frac{d}{D}}\right) H_{ice} \frac{EAR}{Z} D \qquad \text{kN}$$

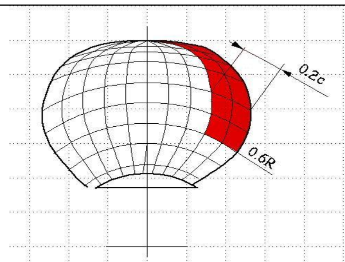
Where: $D_{limit} = \left(\frac{2}{1 - \frac{d}{D}}\right) H_{ice}$, in m;

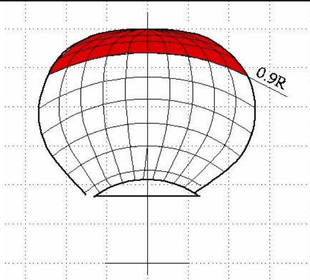
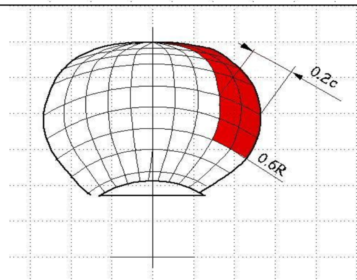
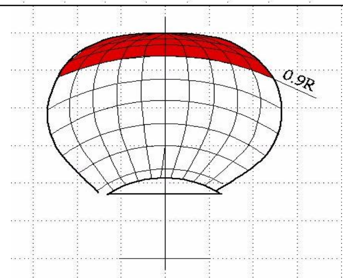
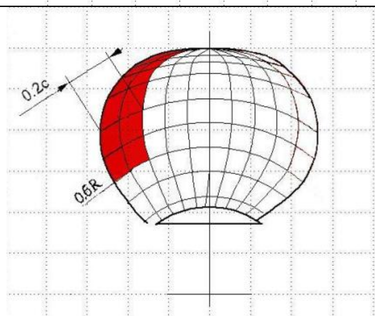
H_{ice} is selected in accordance with Table 4.4.2.

(3) Load cases

Load cases 1-4 are to be covered, as given in Table 4.4.3(3), for CP and FP propellers. In order to obtain blade ice loads for a reversing propeller, load case 5 is also to be covered for propellers where reversing is possible.

Load cases for open propellers **Table 4.4.3(3)**

Load case	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	Force	Loaded area	Right-handed propeller blade seen from behind
Load case 2	50 % 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%F_f	Uniform pressure applied on propeller face (pressure side) on the propeller tip area outside 0.9R radius.	
Load case 5	60%F_f or 60%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	

4.4.4 Design ice loads for ducted propellers

(1) Maximum backward blade force F_b

When $D < D_{limit}$:

$$F_b = -9.5 C_1 S_{ice} \left(\frac{EAR}{Z}\right)^{0.3} (nD)^{0.7} D^2 \qquad \text{kN}$$

When $D \geq D_{limit}$:

$$F_b = -66 C_1 S_{ice} \left(\frac{EAR}{Z}\right)^{0.3} (nD)^{0.7} H_{ice}^{1.4} D^{0.6} \qquad \text{kN}$$

Where: $D_{limit} = 4\ H_{ice}$, in m;

C_1 —— coefficient, to be taken as 1.1 for a ship with icebreaker class notation;

S_{ice} and H_{ice} are selected in accordance with Table 4.4.2;

n is the same as 4.4.3(1).

(2) Maximum forward blade force F_f

When $D < D_{limit}$:

$$F_f = 250 \frac{EAR}{Z} D^2 \qquad \text{kN}$$

When $D \geq D_{limit}$:

$$F_f = 500 \left(\frac{1}{1 - \frac{d}{D}} \right) H_{ice} \frac{EAR}{Z} D \qquad \text{kN}$$

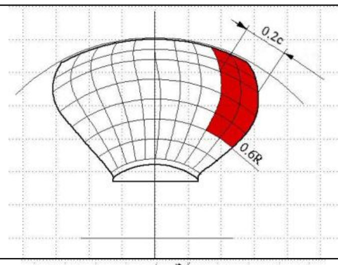
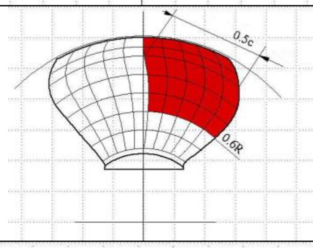
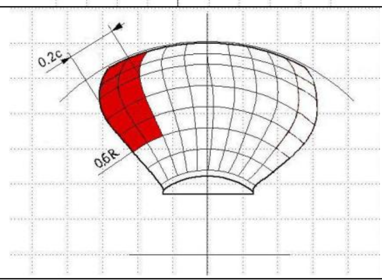
where: $D_{limit} = \left(\frac{2}{1 - \frac{d}{D}} \right) H_{ice}$, m;

H_{ice} is selected in accordance with Table 4.4.2.

(3) Load cases

Load cases 1 and 3 are to be covered as given in Table 4.4.4(3) for all propellers. In order to obtain blade ice loads for a reversing propeller, load case 5 is also to be covered for propellers, where reversing is possible.

Load cases for ducted propellers **Table 4.4.4(3)**

Load case	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 0.6*R* 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 0.6*R* to the tip and from the leading edge to 0.5 times the chord length.	
Load case 5	60%F_f or 60%F_b, whichever is greater	Uniform pressure applied on propeller face (pressure side) to an area from 0.6*R* to the tip and from the trailing edge to 0.2 times the chord length.	

4.4.5 Design ice loads for both open and ducted propellers

(1) Maximum blade spindle torque Q_{smax} for CP mechanism designing

The spindle torque Q_{smax} around the axis of the blade fitting is to be determined both for the maximum backward blade force F_b and forward blade force F_f, which are applied as per Table 4.4.3(3) and Table 4.4.4(3). If the above method gives a value which is less than the default value given by the formula below, the default value is to be used.

$$Q_{smax} = 0.25 F c_{0.7} \qquad \text{kNm}$$

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 4.4.6.

(2) Load distributions (spectra) for blade loads

The Weibull-type distribution (probability that F_{ice} exceeds $(F_{ice})_{\max}$), as given in Figure 4.4.5(2) is used for the fatigue design of the blade.

$$P\left(\frac{F_{ice}}{(F_{ice})_{\max}} \geqslant \frac{F}{(F_{ice})_{\max}}\right) = e^{\left(-\left(\frac{F}{(F_{ice})_{\max}}\right)^k \ln(N_{ice})\right)}$$

where: k —— shape parameter of the spectrum, for open propeller, k=0.75; for ducted propeller, k=1.0;

N_{ice} —— number of load cycles in the spectrum, see 4.4.5(3);

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:

$$(\sigma_{ice})_A(N) = (\sigma_{ice})_{A\max}\left\{1 - \frac{\log(N)}{\log(N_{ice})}\right\}^{\frac{1}{k}}$$

where: $(\sigma_{ice})_{A\max} = \dfrac{(\sigma_{ice})_{f\max} - (\sigma_{ice})_{b\max}}{2}$.

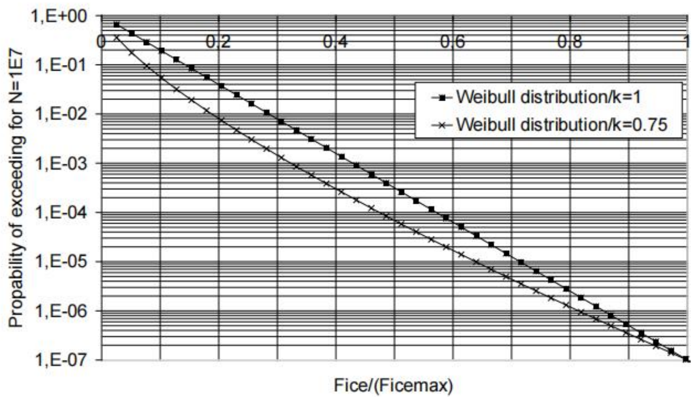


Figure 4.4.5(2) The Weibull-type distribution (probability of exceedance) that is used for fatigue design

(3) Number of ice loads

Number of load cycles in the load spectrum per blade is to be determined according to the formula:

$$N_{ice} = k_1 k_2 N_{class} n$$

where: N_{class} —— reference number of impacts per propeller revolution for each ice class (Table 4.4.5(3));

k_1 —— propeller position coefficient, to be taken as 1 for centre propeller; 2 for wing propeller; 3 for pulling propeller (wing and centre);

k_2 —— immersion coefficient, to be determined as follows:

When $f < 0$: $k_2 = 0.8 - f$;

When $0 \leqslant f \leqslant 1$: $k_2 = 0.8 - 0.4f$;

When $1 < f \leqslant 2.5$: $k_2 = 0.6 - 0.2f$;

When $f > 2.5$: $k_2 = 0.1$;

where: $f = \dfrac{h_0 - H_{ice}}{D/2} - 1$, if h_0 is not known, h_0 may be taken as $D/2$.

For ships with the icebreaker class notation, the above-mentioned number of load cycles N_{ice} is to 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).

Reference number of impacts **Table 4.4.5(3)**

Ice class	PC1	PC2
N_{class}	21×10^6	17×10^6

4.4.6 Blade failure load for both open and ducted propellers

(1) Bending force F_{ex}

The minimum load required resulting in blade failure through plastic bending. It is to be calculated iteratively along the radius of the blade from blade root to 0.5R using the equation below with the ultimate load assumed to be acting at 0.8R in the weakest direction.

$$F_{ex} = \frac{0.3ct^2\sigma_{ref1}}{0.8D-2r}\times10^3 \quad \text{kN}$$

where: 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 failure load may be calculated alternatively 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.

(2) Spindle torque Q_{sex}

The maximum spindle torque due to a blade failure load acting at 0.8R is to 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 is to be defined by an appropriate stress analysis or using equation below.

$$Q_{sex} = \max(c_{LE0.8}, 0.8c_{TE0.8})C_{spex}F_{ex}$$

where: $C_{spex} = C_{sp}C_{fex} = 0.7\left[1-\left(4\frac{EAR}{Z}\right)^3\right]$, to be taken not less than 0.3;

C_{sp} —— non-dimensional parameter taking into account the spindle arm;

C_{fex} —— non-dimensional parameter taking into account the reduction of blade failure force at the location of maximum spindle torque;

Figure 4.4.6(2) illustrates the spindle torque values due to blade failure loads across the whole chord length.

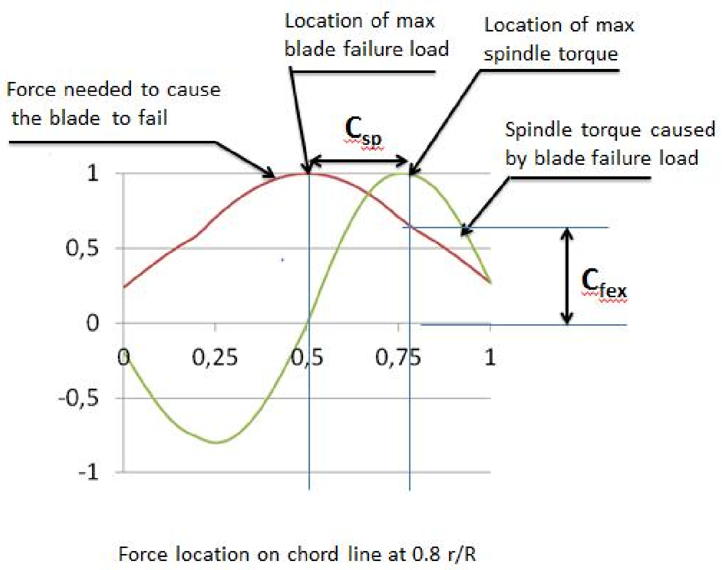


Figure 4.4.6(2) Schematic figure showing blade failure load and related spindle torque when the force acts at different location on the chord line at radius 0.8*R*

4.4.7 Axial design loads acting on open and ducted propellers

(1) Maximum ice thrust on propeller

Forward thrust T_f:

$$T_f = 1.1 F_f \qquad \text{kN}$$

Backward thrust T_b:

$$T_b = 1.1 F_b \qquad \text{kN}$$

However, the load models within this Section do not include propeller/ice interaction loads where an ice block hits the propeller hub of a pulling propeller.

(2) Design thrust along the propulsion shaft line for 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 is to 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:

$$T_r = T + 2.2 T_f \qquad \text{kN}$$

In a backward direction:

$$T_r = 1.5 T_b \qquad \text{kN}$$

If the hydrodynamic bollard thrust, T, is not known, T is to be taken in accordance with Table 4.4.7(2):

Value of *T* **Table 4.4.7(2)**

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)	$0.75\ T_n$

Note: T_n is the nominal propeller thrust at MCR in the free running open water condition, in kN.

For pulling type propellers, ice interaction loads on propeller hub are to be be considered in addition to the above, and submitted to CCS for special consideration.

4.4.8 Torsional design loads acting on open and ducted propellers

(1) Design ice torque on propeller for open propellers

When $D < D_{limit}$:

$$Q_{\max} = k_{open}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}(nD)^{0.17}D^{3} \quad \text{kNm}$$

When $D \geq D_{limit}$:

$$Q_{\max} = 1.9k_{open}H_{ice}^{1.1}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}(nD)^{0.17}D^{1.9} \quad \text{kNm}$$

where: $D_{limit} = 1.8\ H_{ice}$, in m;

H_{ice} is selected in accordance with Table 4.4.2;

k_{open} —— taken as14.7;

$P_{0.7}$ —— propeller pitch at $0.7R$, in m; (For CP propellers, $P_{0.7}=P_{0.7b}$; if $P_{0.7b}$ is not known, $P_{0.7}=0.7P_{0.7n}$)

n —— the rotational propeller speed in r/s in bollard condition. If not known, n is to be taken as follows:

Value of *n* **Table 4.4.8(1)**

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

Note: n_n is the nominal propeller rotational speed at MCR in the free running open water condition, in r/s.

(2) Design ice torque on propeller for ducted propellers

When $D < D_{limit}$:

$$Q_{\max} = k_{ducted}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}(nD)^{0.17}D^{3} \quad \text{kNm}$$

When $D \geq D_{limit}$:

$$Q_{\max} = 1.9k_{ducted}H_{ice}^{1.1}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}(nD)^{0.17}D^{1.9} \quad \text{kNm}$$

where: $D_{limit} = 1.8\ H_{ice}$, in m;

H_{ice} is selected in accordance with Table 4.4.2;

k_{ducted} —— taken as 10.4;

$P_{0.7}$ and n are selected as the same as 4.4.8(1).

(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 are to be calculated at the least favourable phase angle for ice versus engine excitation, when calculated in time domain. The engine firing pulses are to 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 are to cover the rotational speeds up to 105% of MCR speed.
See also Guidelines for calculations given in 4.4.9.
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:

When $\varphi = 0 \ldots a_i$:

$$Q(\varphi) = C_q Q_{max} sin[\varphi(180 / a_i)] \quad \text{kNm}$$

When $\varphi = a_i \ldots 360$:

$$Q(\varphi) = 0 \quad \text{kNm}$$

where: φ —— rotation angle starting when the first impact occurs.

α_i —— duration of interaction between propeller blades and ice measured by rotation angle;

Parameters C_q and α_i for different blade numbers are given in Table 4.4.8(3)①.

Ice impact magnification C_q and duration α_i factors for different blade numbers

Table 4.4.8(3)①

Torque excitation	Propeller/ ice interaction	C_q	α_i (°)			
			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°/(2Z)）	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°/Z.
At the beginning and at the end of the milling sequence (within calculated duration) linear ramp functions are to be used to increase C_q to its maximum within one propeller revolution and vice versa to decrease it to zero (see Figure 4.4.8(3)①).
The number of propeller revolutions during a milling sequence is to be obtained from the formula:

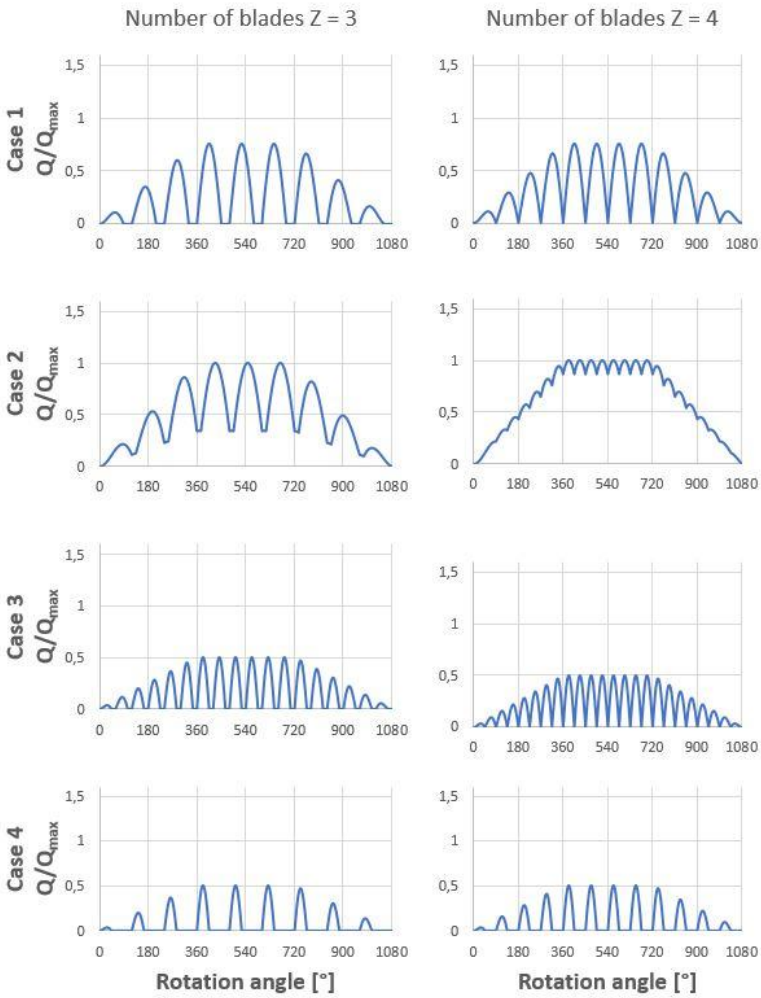
$$N_Q = 2H_{ice}$$

The number of impacts for blade order excitation:

$$ZN_Q$$

where: H_{ice} is selected in accordance with Table 4.3.2.
An illustration of all excitation cases for different blade numbers is given in Figure 4.4.8(3)①.
The dynamic simulation is to 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 is also to 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 is to be applied. On these cases, the excitation is to follow the shaft speed.



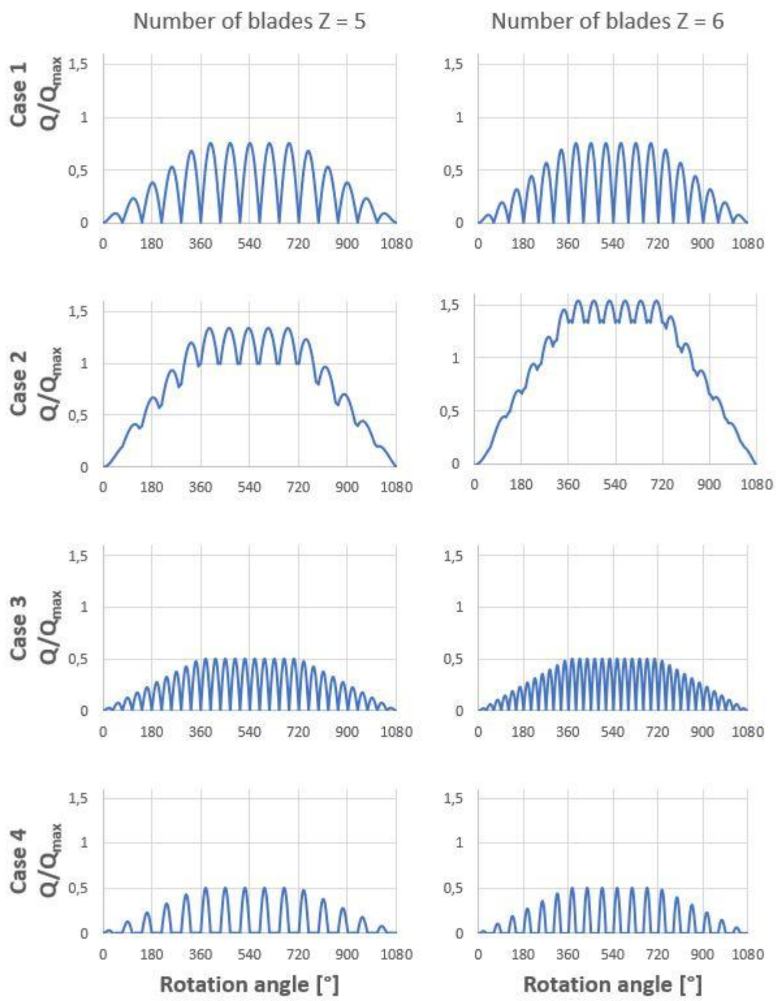


Figure 4.4.8(3)① Ice block torque excitation cases for 3~6 blade propellers

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.

$$Q_F(\varphi) = Q_{\max} \cdot \left[C_{q0} + C_{q1} \cdot \sin(Z \cdot E_0 \cdot \varphi + \alpha_1) + C_{q2} \cdot \sin(2Z \cdot E_0 \cdot \varphi + \alpha_2) \right] \quad \text{kNm}$$

where: φ —— angle of rotation;

C_{q0} —— mean torque component;

C_{q1} —— first blade order excitation amplitude;

C_{q2} —— second blade order excitation amplitude;

E_0 —— number of ice blocks simultaneously participating in the impact.

Coefficients for simplified excitation torque estimation are given in Table 4.4.8(3)②.

Coefficients for simplified excitation torque estimation   Table 4.4.8(3)②

Torque excitation	Z=3					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Case 1	0.375	0.375	-90	0	0	1
Case 2	0.7	0.33	-90	0.05	-45	1
Case 3	0.25	0.25	-90	0	0	2
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
Case 1	0.45	0.36	-90	0.06	-90	1
Case 2	0.9375	0	-90	0.0625	-90	1
Case 3	0.25	0.251	-90	0	0	2
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
Case 1	0.45	0.36	-90	0.06	-90	1
Case 2	1.19	0.17	-90	0.02	-90	1
Case 3	0.3	0.25	-90	0.048	-90	2
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
Case 1	0.45	0.375	-90	0.05	-90	1
Case 2	1.435	0.1	-90	0	0	1
Case 3	0.3	0.25	-90	0.048	-90	2
Case 4	0.2	0.25	0	0.05	-90	1

Torsional vibration responses are to 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}, see Figure 4.4.8(3)②.
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}, see Figure 4.4.8(3)②.
Q_{Amax} can by determined by:

$$Q_{A\max} = \left[\max(Q_r(t)) - \min(Q_r(\mathrm{t}))\right] / 2 \qquad \text{kNm}$$

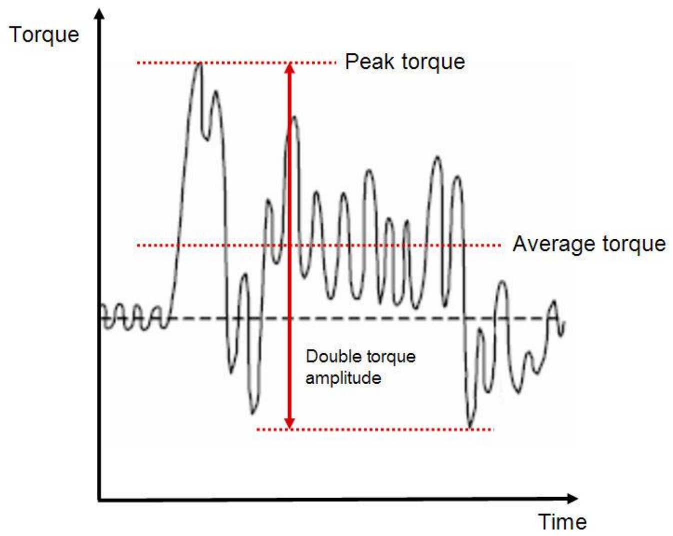


Figure 4.4.8(3)② Interpretation of different torques in a measured curve, as example

(4) Design torque along shaft line

① 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, the following estimation of the maximum response torque can be used to calculate the design torque along the propeller shaft line.

$$Q_r = Q_{e\max} + Q_{\text{vib}} + Q_{\max}\frac{I}{I_t} \qquad \text{kNm}$$

The above equation is to be applied for directly coupled two stroke diesel engines without flexible coupling. For all other plants:

$$Q_r = Q_{e\max} + Q_{\max}\frac{I}{I_t} \qquad \text{kNm}$$

where: all the torques and the inertia moments are to be reduced to the rotation speed of the component being examined.

If the maximum torque $Q_{e\max}$ is not known, it is to be taken as follows:

Value of $Q_{e\max}$ **Table 4.4.9(1)**

Propeller type	$Q_{e\max}$
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	$0.75Q_n$

Note: Q_{motor} is the electric motor peak torque.

② 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 is to 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.

4.4.9 Torsional vibration calculation

(1) 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 is to be applied.

(2) For time domain analysis the model is to 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 are to 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.

(3) For frequency domain calculations the load is to be estimated as a Fourier component analysis of the continuous sequence of half sine load peaks. The first and second order blade components are to be used for excitation. The calculation is to 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:

$$Q_{peak} = Q_{e\max} + Q_{\mathrm{opw}} + Q_{\mathrm{ice}} \qquad \text{kNm}$$

where: Q_{emax} —— the maximum engine torque at considered rotational speed;

Q_{opw} —— the maximum open water response of engine excitation at considered shaft speed and determined by frequency domain analysis;

Q_{ice} —— 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.

4.5 Design

4.5.1 Design principle

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

The propulsion line components are to 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.

4.5.2 Fatigue design in general

The design loads are to be based on the ice excitation and where necessary (shafting) dynamic analysis, described as a sequence of blade impacts (4.4.8(3)①). The shaft response torque is to be determined according 4.4.8(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.

$$D_{mds} = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \cdots + \frac{n_k}{N_k} \le 1 \qquad \text{or} \qquad D_{mds} = \sum_{J=1}^{j=k} \frac{n_j}{N_j} \le 1$$

where: k —— the number of stress levels;

$N_{1\ldots k}$ —— the number of load cycles to failure of the individual stress level class;

$n_{1\ldots k}$ —— the accumulated number of load cycles of the case under consideration, per class;

D_{mds} —— Miner damage sum.

The stress distribution is to 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 σ_{ref2} for propeller blades and yield strength for all other components. The load distribution (spectrum) is to be in accordance with the Weibull distribution.

4.5.3 Propeller blades

(1) Calculation of blade stresses due to static loads

The blade stresses (equivalent and principal stresses) are to be calculated for the design loads given in 4.4.3 and 4.4.4. Finite element analysis (FEA) is to be used for stress analysis as part of the final approval for all propeller blades. The von Mises stresses, taken as σ_{st}, are to comply with

criteria in (2).
Alternatively, the following simplified equation can be used in estimating the blade stresses for all propellers in the root area (r/R < 0.5) for final approval.

$$\sigma_{st} = A_1 \frac{M_{BL}}{100 \times ct^2} \qquad \text{MPa}$$

where: constant A_1 is the $\frac{actual\ stress}{stress\ obtained\ with\ beam\ equation}$. If the actual value is not available, A_1 is to be taken as 1.6.

$M_{BL} = (0.75 - r/R)RF$, for relative radius r/R<0.5, where F is the maximum of F_b and F_f, whichever is greater.

(2) Acceptability criterion for static loads

The following criterion for calculated blade stresses is to be fulfilled:

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

where: σ_{st} —— calculated stress for the design loads. If FE analysis is used in estimating the stresses, von Mises stresses are to be used.

(3) Fatigue design of propeller blade

For materials with a two slope S-N curve the fatigue calculations defined in this chapter are not required if the following criterion is fulfilled.

$$\sigma_{\exp} \geqslant B_1 \cdot \sigma_{ref2}^{B_2} \cdot \log\left(N_{ice}\right)^{B_3}$$

Where: B_1, B_2 and B_3 are coefficients for propellers given in Table 4.5.3(3).

Values of B_1、 B_2 and B_3 Table 4.5.3(3)

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

Where the above criterion is not fulfilled the fatigue requirements defined below are to 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 is to be calculated according to Miner's rule and the acceptability criterion for fatigue is to be fulfilled as given in this Section. The equivalent stress is normalised for 1 × 10^8 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 4.4.
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.

$$\Delta\sigma_{\max} = 2 \cdot \sigma_{A\max} = \left|(\sigma_{ice})_{f\max}\right| + \left|(\sigma_{ice})_{b\max}\right| \qquad \text{MPa}$$

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 4.5.3(3)-1. One slope S-N curve (the slope can be chosen),

see Figure 4.5.3(3)-2.
The type of the S-N-curve is to 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 is to be used.

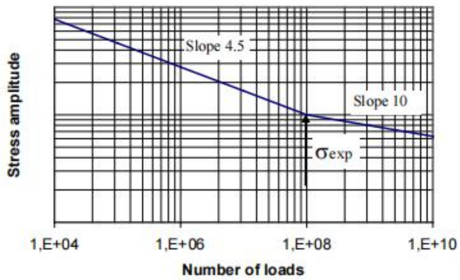


Figure 4.5.3(3)-1 Two-slope *S-N* curve

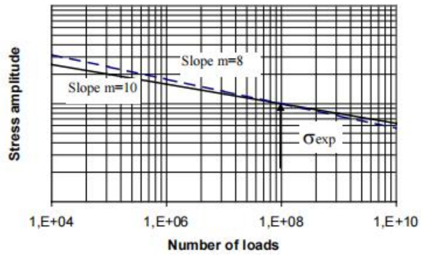


Figure 4.5.3(3)-2 Constant-slope *S-N* curve

① Equivalent fatigue stress
A more general method of determining the equivalent fatigue stress of propeller blades is described in 4.5.5, where the principal stresses are considered according to 4.4.3 and 4.4.4 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:

$$\sigma_{fat} = \rho \cdot \left(\sigma_{ice}\right)_{\max} \qquad \text{MPa}$$

where: $(\sigma_{ice})_{\max}$ —— mean value of the principal stress amplitudes resulting from design forward and backward blade forces at the location being studied, to be determined as follows:

$$\left(\sigma_{ice}\right)_{\max} = 0.5 \cdot \left(\left(\sigma_{ice}\right)_{f\max} - \left(\sigma_{ice}\right)_{b\max}\right) \qquad \text{MPa}$$

Where: $(\sigma_{ice})_{f\max}$ is principal stress resulting from forward load, $(\sigma_{ice})_{b\max}$ is 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 $(\sigma_{ice})_{b\max}$. Case 5 is excluded from the fatigue analysis.
② Calculation of parameter ρ for two-slope S-N curve
When $5 \times 10^6 \le N_{ice} \le 1\times10^8$, the error of the following method to determine the parameter ρ is sufficiently small.
The parameter ρ relates the maximum ice load to the distribution of ice loads according to the regression formula.

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

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 4.5.3(3)④.
The coefficients C_1, C_2, C_3 and C_4 are given in Table 4.5.3(3)②.

Values of C_1、C_2、C_3 and C_4 值 **Table 4.5.3(3)②**

Coefficient	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

③ Calculation of parameter ρ for constant-slope S-N curve

For materials with a constant-slope S-N curve, see Figure 4.5.3(3)-2, the factor ρ is to be calculated from the following formula:

$$\rho = \left(G \frac{N_{ice}}{N_R} \right)^{1/m} \left[\ln(N_{ice}) \right]^{-1/k}$$

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);

G —— Values for the parameter G are given in Table 4.5.3(3) ③ below. Linear interpolation may be used to calculate the value of G for m/k ratios other than those given in the Table 4.5.3③.

Value for the parameter *G* for different *m/k* ratios Table 4.5.3(3)③

m/k	G	*m/k*	G	*m/k*	G	*m/k*	G
3	6	5.5	287.9	8	40320	10.5	11.899E6
3.5	11.6	6	720	8.5	119292	11	39.917E6
4	24	6.5	1871	9	362880	11.5	136.843E6
4.5	52.3	7	5040	9.5	1.133E6	12	479.002E6
5	120	7.5	14034	10	3.623E6		

④ Acceptability criterion for fatigue

The equivalent fatigue stress σ_{fat} at all locations on the blade is to fulfil the following acceptability criterion:

$$\frac{\sigma_{fl}}{\sigma_{fat}} \geqslant 1.5$$

where: $\sigma_{fl} = \gamma_{\varepsilon 1} \cdot \gamma_{\varepsilon 2} \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp}$, in MPa;

$\gamma_{\varepsilon 1}$—— reduction factor due to scatter (equal to one standard deviation);

$\gamma_{\varepsilon 2}$—— reduction factor for test specimen size effect;

σ_{exp} in Table 4.5.3(3)④ 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 4.5.3(3)④ may be used, provided the values are determined under conditions approved by CCS.

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%

Table 4.5.3(3)④

Bronze and brass（a=1.0）		Stainless steel（a=0.05）	
Mn-Bronze, Cu1 (high tensile brass)	84 MPa	Ferritic (12Cr 1Ni)	144 MPa①
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

Note: ① This value may be used, provided a perfect galvanic protection is active. Otherwise a reduction of about 30 MPa is to be applied.

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 10^8 load cycles.

The geometrical size factor ($\gamma_{\varepsilon 2}$) is:

$$\gamma_{\varepsilon 2} = 1 - a \cdot \ln(\frac{t}{0.025})$$

where: a —— as given in Table 4.5.3(3)④;

t —— the maximum blade thickness at the considered point.

The mean stress effect γ_m is:

$$\gamma_m = 1.0 - \left(\frac{1.4 \cdot \sigma_{mean}}{R_m}\right)^{0.75}$$

The following values are to be used for the reduction factors if actual values are not available:

$\gamma_{\varepsilon 1} = 0.85$, $\gamma_v = 0.75$, $\gamma_m = 0.75$.

4.5.4 Blade bolts, propeller hub and CP mechanism

(1) General

The blade bolts, CP mechanism, propeller boss and the fitting of the propeller to the propeller shaft are to be designed to withstand the maximum static and fatigue design loads (as applicable), as defined in 4.4.3, 4.4.4, 4.4.5 and 4.5.3. The safety factor against yielding due to static loads and against fatigue is to be greater than 1.5, if not stated otherwise. The safety factor for loads, resulting from propeller blade failure as defined in 4.4.6 is to 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 tensile strength of respective materials, detailed fatigue analysis is not required. In all other cases components are to be analysed for cumulative fatigue. An approach similar to that used for shafting assessment may be applied (4.5.5).

(2) Blade bolts

Blade bolts are to 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:

$$M_{bolt} = S \cdot F_{ex}(0.8\frac{D}{2} - r_{bolt}) \qquad \text{kNm}$$

where: r_{bolt}—— radius to the bolt plane (measured from the center line of propeller shaft);

S —— 1.0 safety factor.

Blade bolt pre-tension is to be sufficient to avoid separation between mating surfaces when the maximum forward and backward ice loads defined in 4.4.3 and 4.4.4 (open and ducted propellers respectively) are applied. For conventional arrangements, the following formula may be applied:

$$d_{bb} = 41 \cdot \sqrt{\frac{F_{ex}(0.8D - d) \cdot S \cdot \alpha}{R_{p0.2} \cdot z_{bb} \cdot PCD}} \qquad \text{mm}$$

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, in mm;

z_{bb} —— number of blade bolts;

S —— 1.0 safety factor.

(3) CP mechanism

Separate means, e.g. dowel pins, are to be provided in order to withstand the spindle torque resulting from blade failure Q_{sex} (see 4.4.6(2)) or ice interaction Q_{smax} (see 4.4.5(1)) whichever is greater. Other components of the CP mechanism are not to 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 between the blade and blade carrier can be calculated using the formula:

$$d_{fp} = 66 \cdot \sqrt{\frac{Q_s - Q_{fr}}{PCD \cdot z_{pin} \cdot R_{p0.2}}} \qquad \text{mm}$$

where: Q_s —— max($S \cdot Q_{s\max}$; $S \cdot Q_{sex}$), in kNm;

S —— 1.3 for $Q_{s\max}$, 1.0 for Q_{sex} ;

Q_{fr} —— friction between connected surfaces,= $0.33 \cdot Q_s$, in kNm. CCS 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:

$$\sigma_{vMises} = \sqrt{\left(\frac{F_m \cdot \frac{h_{pin}}{2}}{\frac{\pi \cdot d_{pin}^{\ 3}}{32}}\right)^2 + 3 \cdot \left(\frac{F_m}{\frac{\pi}{4} \cdot d_{pin}^{\ 2}}\right)^2} \qquad \text{MPa}$$

where: F_m —— $\left(Q_s - Q_{fr}\right) / l_m$, in kN;

l_m —— distance pitching centre of blade – axis of pin, in m;

h_{pin} —— height of actuating pin, in mm;

d_{pin} —— diameter of actuating pin, in 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, in kNm.

The blade failure spindle torque Q_{sex} is not to 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:

$$Q_{sa\max} = \frac{Q_{sb} + Q_{sf}}{2} \qquad \text{kNm}$$

where: Q_{sb} —— spindle torque due to $|F_b|$, in kNm;

Q_{sf} —— spindle torque due to $|F_f|$, in kNm.

(4) Servo pressure

The design pressure for the servo system is to 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 is in any case not to be less than relief valve set pressure.

4.5.5 Propulsion line components

The ultimate load resulting from total blade failure F_{ex} as defined in 4.4.6 is to 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 are to be designed to withstand the operational propeller/ice interaction loads as given in 4.4.

The given loads are not intended to be used for shaft alignment calculation.

Cumulative fatigue calculations are to 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 $k = 1$), with Figure 4.5.5(1) as an example:

$$Q_A(\mathrm{N}) = Q_{A\max} \cdot \left[1 - \frac{\log(N)}{\log(Z \cdot N_{ice})}\right]$$

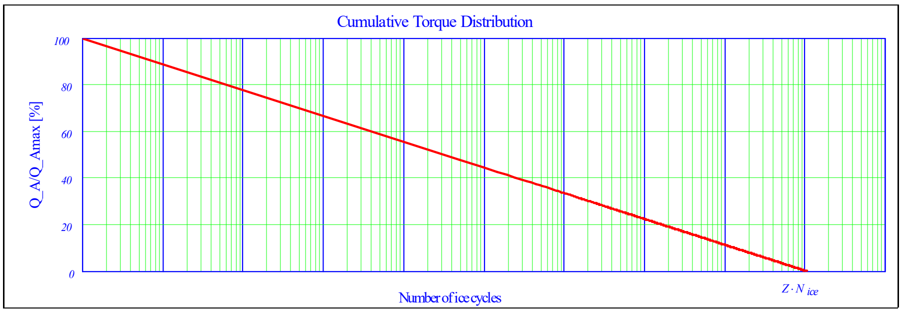


Figure 4.5.5(1) Cumulative torque distribution

The number of load cycles in the load spectrum is defined as $Z \cdot N_{ice}$.

The Weibull exponent is to 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 is to 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.

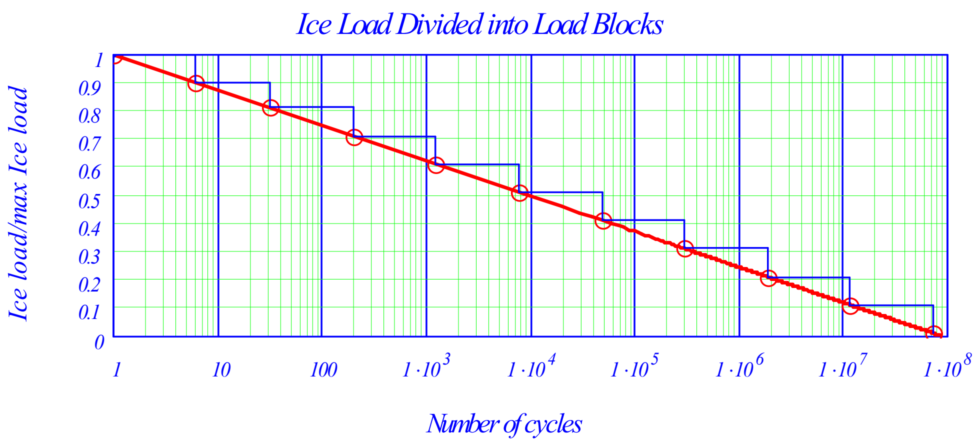


Figure 4.5.5(2) Example of ice load distribution (spectrum) for the shafting (k = 1)

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.

$$n_i = N_{ice}^{1-\left(1-\frac{i}{n_{bl}}\right)^k} - \sum_{i=1}^{i} n_{i-1}$$

where: i —— single load block, n_{bl} is the number of load blocks.

(1) Propeller fitting to the shaft

Keyless cone mounting or flange mounting may be generally used for propeller fitting to the shaft. Key mounting alone is not permitted.

① Keyless cone mounting

The friction capacity (at 0 ℃) is to be at least S = 2.0 times the highest peak torque Q_{peak} as determined in 4.4.9 without exceeding the permissible hub stresses.

The necessary surface pressure can be determined as:

$$p_{0^\circ C} = \frac{2 \cdot S \cdot Q_{peak}}{\pi \cdot \mu \cdot D_S^2 \cdot L \cdot 10^3} \qquad \text{MPa}$$

where: μ —— 0.15 for steel-steel; 0.13 for steel-bronze;

D_S —— the shrinkage diameter at the mid-length of the taper, in m;

L —— the effective length of taper, in m.

Above friction coefficients may be increased by 0.04 if glycerine is used in wet mounting.

② Flange mounting

The flange thickness is to be at least 25% of the required aft end shaft diameter.

Any additional stress raisers such as recesses for bolt heads are not to 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 is to be calculated according to the following equation:

$$d_{pin} = 66 \cdot \sqrt{\frac{Q_{peak} \cdot S}{PCD \cdot z_{pin} \cdot R_{p0.2}}} \qquad \text{mm}$$

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} (4.4.6) in backward direction does not cause yielding of the bolts. The following equation is to be applied:

$$d_b = 41 \cdot \sqrt{\frac{F_{ex} \cdot (0.8 \cdot D / PCD + 1) \cdot \alpha}{R_{p0.2} \cdot z_b}} \qquad \text{mm}$$

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, in mm;

z_b —— number of flange bolts.

(2) Propeller shaft

The propeller shaft is to be designed to fulfil the following:

① The blade failure load F_{ex} (4.4.6) applied parallel to the shaft (forward or backwards) is not to cause yielding. The bending moment need not to be combined with any other loads.

The diameter in way of the aft stern tube bearing is not to be less than:

$$d_p = 160 \cdot \sqrt[3]{\frac{F_{ex} \cdot D}{R_{p0.2} \cdot \left(1 - \frac{d_i^4}{d_p^4}\right)}} \qquad \text{mm}$$

where: d_p —— propeller shaft diameter, in mm;

d_i —— propeller shaft inner diameter, in mm, 0 for solid shaft.

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.

② The stresses due to the peak torque Q_{peak} are to 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.

Considering peak torque, the minimum shaft diameter is not to be less than that calculated from the following formulae:

Plain shaft:
$$d_p = 210 \cdot \sqrt[3]{\frac{Q_{peak} \cdot S}{R_{p0.2} \cdot \left(1 - \frac{d_i^4}{d_p^4}\right)}} \quad \text{mm}$$

Notched shaft:
$$d_p = 210 \cdot \sqrt[3]{\frac{Q_{peak} \cdot S \cdot \alpha_t}{R_{p0.2} \cdot \left(1 - \frac{d_i^4}{d_p^4}\right)}} \quad \text{mm}$$

where: α_t —— local stress concentration factor in torsion.

Notched shaft diameter is in any case not to be less than the required plain shaft diameter.

③ The torque amplitudes (4.4.8(4)) with the corresponding number of load cycles are to 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 is also to be considered.

④ The fatigue strengths σ_F and τ_F (3×10^6 cycles) of shaft materials may be assessed on the basis of the material's yield or 0.2% proof strength as:

$$\sigma_F = 0.436 \cdot R_{p0.2} + 77 = \tau_F \cdot \sqrt{3} \quad \text{MPa}$$

This is valid for small polished specimens (no notch) and reversed stresses, see "VDEH 1983 Report No.abf11 calculation of Wöhlerlinien for components made of steel".

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 3×10^6 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.

(3) Intermediate shaft

The intermediate shafts are to be designed to fulfil the requirements of 4.5.5(2)②~④.

(4) Shaft connections

Shaft connections are to satisfy the following requirements. Keyed connections (except for splined shaft) are not to be used.

① Shrink fit couplings (keyless)

See 4.5.5(1)①. A safety factor of S = 1.8 is to be applied.

② Flange mounting

The flange thickness is to be at least 20% of the required shaft diameter.

Any additional stress raisers such as recesses for bolt heads are not to 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.

The diameter of ream fitted (light press fit) bolts is to be chosen so that the peak torque is transmitted with a safety factor of 1.9. This accounts for a prestress. Pins are to transmit the peak torque with a safety factor of 1.5 against yielding (see the formula in 4.5.5(1)②) .

The bolts are to be designed so that the blade failure load F_{ex} (4.4.6) in backward direction does not cause yielding.

③ 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 peak torque is to be applied.

(5) Gear transmissions

① Shafts

Shafts in gear transmissions are to meet the same safety level as intermediate shafts, but where relevant, bending stresses and torsional stresses are to be combined (e.g. by von Mises for static loads). Maximum permissible deflection in order to maintain sufficient tooth contact pattern is to be considered for the relevant parts of the gear shafts.

② Gearing

The gearing is to fulfil following three acceptance criteria: tooth root stresses, pitting of flanks, and 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 4.5.5) 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 Pt. 1-6, provided that "method B" is used. Equivalent requirements in Appendix 1, Chapter 10, PART THREE of the Rules for Classification of Sea-going Steel Ships can also be applied.

For Bevel Gears the methods or standards used or acknowledged by CCS can be applied provided that they are properly calibrated.

Tooth root safety is to 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 Appendix 1, Chapter 10, PART THREE of the Rules for Classification of Sea-going Steel Ships).

The safety against pitting is to be assessed in the same way as tooth root stresses, but with a minimum resulting safety factor of 1.2. (Ref ISO 6336 Pt 1, 2 and 6 and Appendix 1, Chapter 10, PART THREE of the CCS Rules for Classification of Sea-going Steel Ships).

The scuffing safety (flash temperature method – ref. ISO/TR 13989-1 and ISO/TR 13989-2) based on the peak torque is to 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 CCS. (It is to 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.)

③ Bearings

See 4.5.5(9).

④ Gear wheel shaft connections

The torque capacity is to be at least 1.8 times the highest peak torque Q_{peak} (at considered rotational speed) as determined in 4.4.9 without exceeding the permissible hub stresses of 80% yield.

(6) Clutches

Clutches are to 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 is to be made possible within reasonably short time. If this is arranged by bolts, it is to be on the engine side of the clutch in order to ensure access to all bolts by turning the engine.

(7) Elastic couplings

There is to be a separation margin of at least 20% between the peak torque and permissible torque (valid for at least a single load cycle), expressed as follows:

$$Q_{peak} < 0.8 \cdot T_{k\max}(N=1) \qquad \text{kNm}$$

A sufficient fatigue strength is to 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 4.5.5(1)):

$$\frac{Q_r(N=x)}{Q_r(N=1)} = 1 - \frac{\log(x)}{\log(Z \cdot N_{ice})} \qquad \text{and}$$

$$\frac{Q_A(N=x)}{Q_A(N=1)} = 1 - \frac{\log(x)}{\log(Z \cdot N_{ICE})}$$

where $Q_r(N=1)$ corresponds to Q_{peak} and $Q_A(N=1)$ to Q_{Amax}.

$$Q_r(\mathrm{N} = 5 \times 10^4) \cdot S < T_{k\max}(\mathrm{N} = 5 \times 10^4) \qquad \text{kNm}$$

$$Q_r(\mathrm{N} = 1 \times 10^6) \cdot S < T_{kv} \qquad \text{kNm}$$

$$Q_A(\mathrm{N} = 5 \times 10^4) \cdot S < \Delta T_{k\max}(\mathrm{N} = 5 \times 10^4) \qquad \text{kNm}$$

where $Q_r(N=1)$ corresponds to Q_{peak} and $Q_A(N=1)$ to Q_{Amax}.

S is the general safety factor for fatigue, equal to 1.5.

See figures below when using the above formulae:

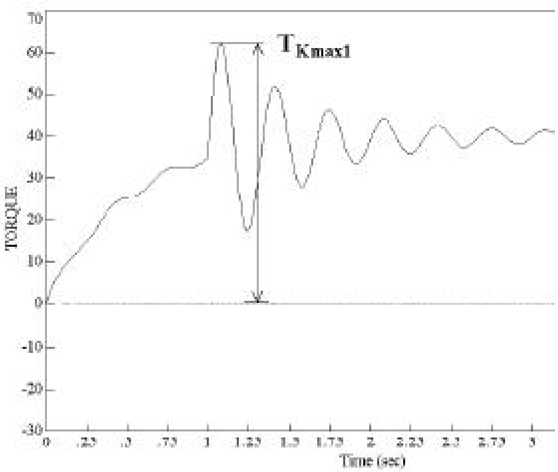


Figure 4.5.5(7)-1

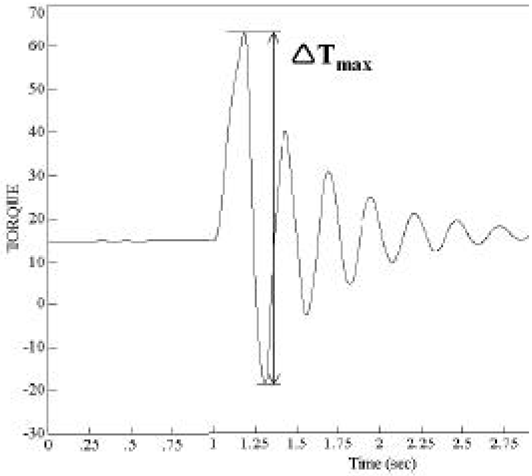


Figure 4.5.5(7)-2

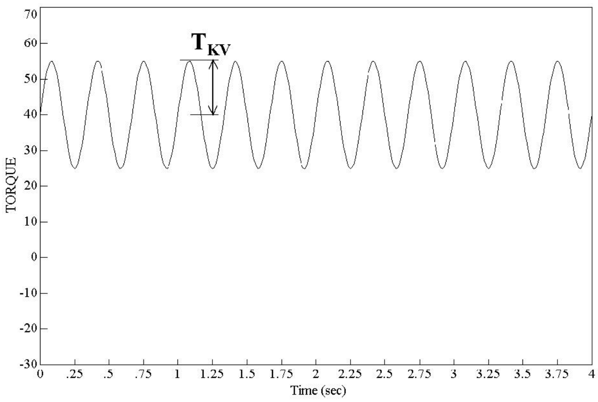


Figure 4.5.5(7)-3

The torque amplitude (or range Δ) is not to 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 5×10^4 cycles and T_{kv} refer to 10^6 cycles. See illustration in below Figures 4.5.5(7)-1, 4.5.5(7)-2 and 4.5.5(7)-3.

$$T_{k\max1}(\mathrm{N}=5\times10^4)\geq Q_r(\mathrm{N}=5\times10^4) \qquad \text{kNm}$$

(8) 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).

(9) Bearings

The aft stern tube bearing as well as the next shaft line bearing are to withstand F_{ex} as given in 4.4.6, 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. Thrust bearings and their housings are to be designed to withstand with the maximum response thrust specified in 4.4.7 and the axial force resulting from the blade failure load specified in 4.4.6. 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.

(10) 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 by CCS.

4.6 Prime movers

4.6.1 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.

4.6.2 The arrangement of compressed air for mechanical starting arrangements (including capacity of air receivers and charging capacity) is to satisfy relevant requirements of 9.5.1, Chapter 9, PART THREE of the Rules for Classification of Sea-going Steel Ships. In addition, if the propulsion engine has to be reversed for going astern, the charging time required by 9.5.1.2, Chapter 9, PART THREE of the Rules for Classification of Sea-going Steel Ships is not to exceed 0.5h.

4.6.3 Provisions are to be made for heating arrangements to ensure ready starting from cold of the emergency power units at an ambient temperature applicable to the ice class of the ship. Emergency power units are to 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 is to 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 is to be provided for an additional three starts within 30 min., unless manual starting can be demonstrated to be effective.

4.7 Equipment fastening loading accelerations

4.7.1 Essential equipment and supports are to be suitable for the accelerations as indicated in the following paragraphs. Accelerations are to be considered as acting independently.

4.7.2 Longitudinal Impact Accelerations, a_l

Maximum longitudinal impact acceleration at any point along the hull girder is calculated by the following formula:

$$a_l = (F_{IB} / \Delta)\ \{[1.1\ tan(\gamma + \varphi)] + [7\frac{H}{L}]\} \qquad \text{m/s}^2$$

where: φ —— maximum friction angle between steel and ice, normally taken as 10 °;

γ —— bow stem angle at waterline, in °;

Δ —— displacement, in kt;

L —— length between perpendiculars, in m;

H —— distance in meters from the water line to the point being considered, in m;

F_{IB} —— vertical impact force, defined in 2.4.2.1.

4.7.3 Vertical Accelerations, a_v

Combined vertical impact acceleration at any point along the hull girder is calculated by the following formula:

$$a_v = 2.5\ (F_{IB} / \Delta) F_X \qquad \text{m/s}^2$$

where: F_{IB} and Δ are the same as 4.7.2;

F_X—— 1.3 at FP, 0.2 at midships, 0.4 at AP, 1.3 at AP for ships conducting ice breaking astern. Intermediate values are to be interpolated linearly.

4.7.4 Transverse impact acceleration, a_t

Combined transverse impact acceleration at any point along hull girder is calculated by the following formula:

$$a_t = 3F_i \frac{F_X}{\Delta} \qquad \text{m/s}^2$$

where: Δ is the same as 4.7.2;

F_i —— total force normal to shell plating in the bow area due to oblique ice impact, defined in 2.1.2.2(3);

F_X —— 1.5 at FP, 0.25 at midships, 0.5 at FP, 1.5 at AP for ships conducting ice breaking astern. Intermediate values are to be interpolated linearly.

4.8 Auxiliary systems

4.8.1 Machinery is to be protected from the harmful effects of ingestion or accumulation of ice or snow. Where continuous operation is necessary, means are to be provided to purge the system

of accumulated ice or snow.
4.8.2 Means are to be provided to prevent damage to tanks containing liquids due to freezing.
4.8.3 Vent pipes, intake and discharge pipes and associated systems are to be designed to prevent blockage due to freezing or ice and snow accumulation.

4.9 Ballast tanks

4.9.1 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.

4.10 Ventilation systems

4.10.1 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. Notwithstanding the above, multiple air intakes are to be provided for the emergency generating set and are to be as far apart as possible.
4.10.2 The temperature of the inlet air is to be suitable for the safe operation of the machinery and the thermal comfort in the accommodation. Accommodation and ventilation air intakes are to be provided with means of heating, if needed.

4.11 Sea inlets and cooling water systems

4.11.1 Cooling water systems for machinery that is essential for the propulsion and safety of the ship, including sea chest inlets, are to be designed for the environmental conditions applicable to the ice class.
4.11.2 At least two sea chests are to be arranged as ice boxes (sea chests for water intake in severe ice conditions) for icebreakers. The calculated volume for each of the ice boxes is to be at least 1 m^3 for every 750 kW of the totally installed power.
4.11.3 Ice boxes are to be designed for an effective separation of ice and venting of air.
4.11.4 Sea inlet valves are to be secured directly to the ice boxes. The valve is to be a full bore type.
4.11.5 Ice boxes and sea bays are to have vent pipes and are to have shut off valves connected directly to the shell.
4.11.6 Means are to be provided to prevent freezing of sea bays, ice boxes, ship side valves and fittings above the load water line.
4.11.7 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.
4.11.8 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.
4.11.9 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. The means of clearing is to be of a type using low pressure steam. Clearing pipes are to be provided with screw-down type non return valves.

4.12 Azimuthing main propulsors

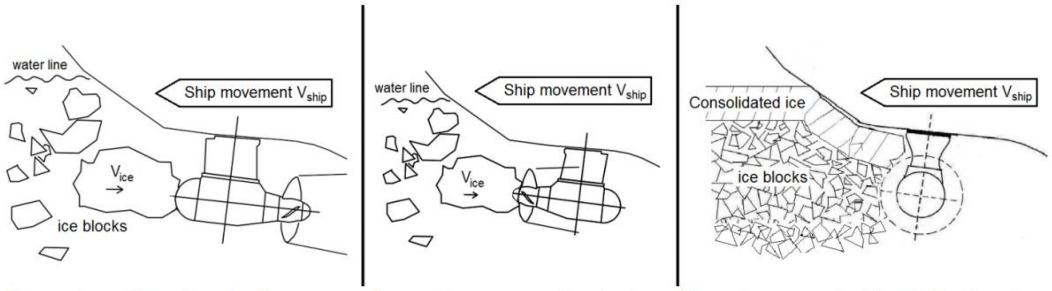
4.12.1 Design principle

In addition to the above requirements for propeller blade dimensioning, azimuthing thrusters are also to 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. Three main ice load scenarios have been selected for defining the extreme ice loads. Examples of loads are illustrated in Figure 4.12.1. In addition, blade order thruster body vibration responses may be estimated for propeller excitation. The following load scenario types are considered:

① Ice block impact on the thruster body or propeller hub;

② Thruster penetration into an ice ridge that has a thick consolidated layer;

③ Vibratory response of the thruster at blade order frequency.



Impact on thruster body | Impact on propeller hub | Thruster penetration into the ice ridge

Figure 4.12.1 Examples of load scenario types

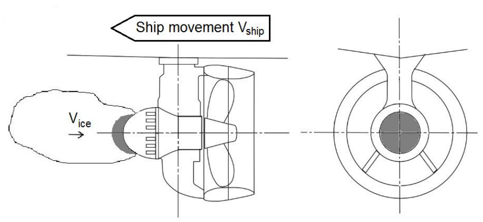
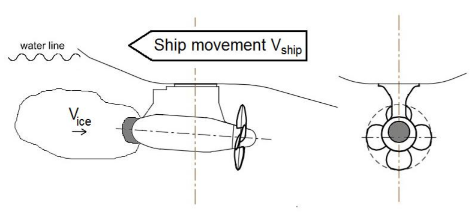
The steering mechanism, the fitting of the unit, and the body of the thruster are to be designed to withstand the plastic bending of a blade without damage. The loss of a blade is to be taken into account for the propeller blade orientation causing the maximum load on the component being studied. Top-down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

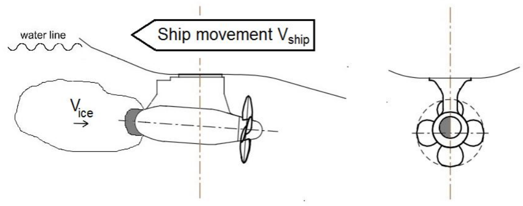
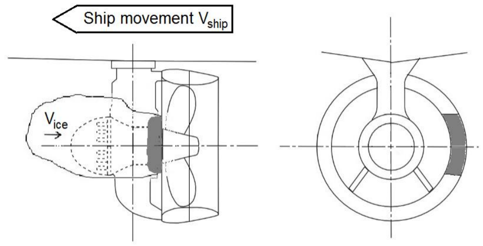
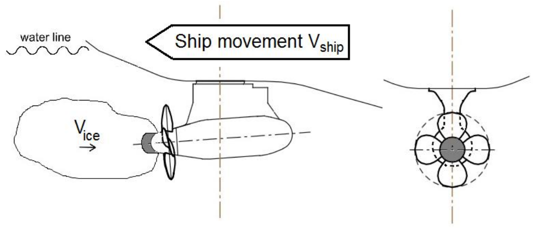
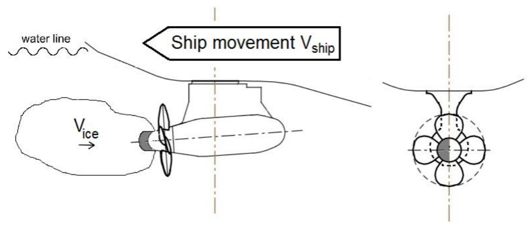
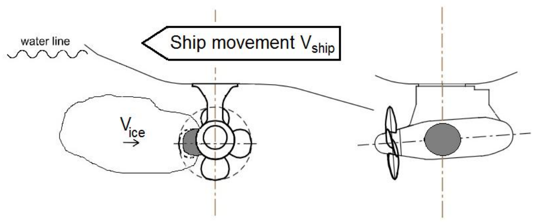
4.12.2 Extreme ice impact loads

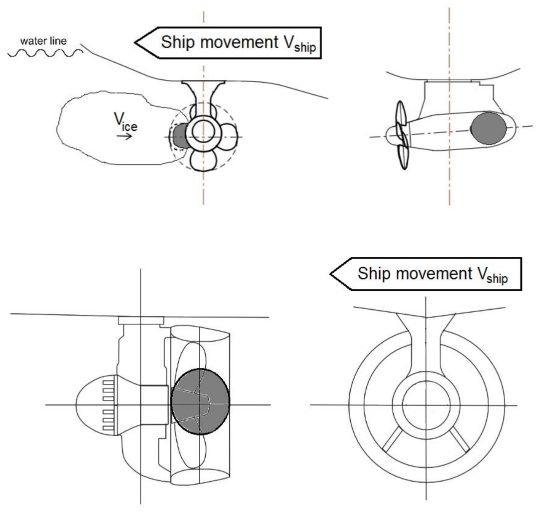
When the ship is operated in ice conditions, ice blocks formed in channel side walls or from the ridge consolidated layer may impact on the thruster body and the propeller hub. Exposure to ice impact is very much dependent on the ship size and ship hull design, as well as the location of the thruster.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

The thruster is to be capable of withstanding the loads occurring when the design ice block defined in Table 4.4.2 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 4.12.2(1). 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 is to 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.

Load cases for azimuthing thruster ice impact loads Table 4.12.2(1)

Load case	force	Loaded area	Illustration
Load case T1a Symmetric longitudinal ice impact on thruster	F_{ti}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area	 

Load case	force	Loaded area	Illustration
Load case T1b Non-symmetric longitudinal ice impact on thruster	50% 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_{iced})	
Load case T2a Symmetric longitudinal 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 propeller hub	50% 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	

Load case	force	Loaded area	Illustration
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 is to be taken from the nozzle length (L_{nz})	

The ice impact contact load is to be calculated using the following formula. The related parameter values are given in Table 4.12.2(2). The design operation speed in ice can be derived from Tables 4.12.2(3) and 4.12.2(4), or the ship in question's actual design operation speed in ice can be used. The longitudinal impact speed in Tables 4.12.3(3) and 4.12.2(4) 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.

$$F_{ti} = 34.5 C_{ice} C_{DM1} R_c^{0.5} (m_{ice} v_s^2)^{0.333} \qquad \text{kN}$$

where: R_C —— the impacting part sphere radius, in m, see Figure 4.12.2;

m_{ice} —— the ice block mass, in kg;

v_s —— the ship speed at the time of contact, in m/s;

C_{DM1} —— the dynamic magnification factor for impact loads;

C_{ice}—— coefficient of ice strength.

C_{DM1} is to be taken from Table 4.12.2(2) if unknown.

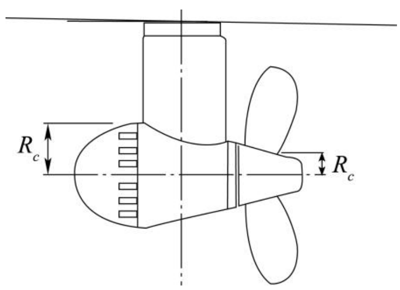


Figure 4.12.2 Dimensions used for R_c

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

$$R_{ceq} = \sqrt{\frac{A}{\pi}} \qquad \text{m}$$

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

Parameter values for ice dimensions and dynamic magnification Table 4.12.2(2)

Icebreaker*	PC1	PC2
Thickness of the design ice block impacting thruster (2/3 H_{ice}), in m	2.67	2.33
Extreme ice block mass (m_{ice}), in kg	103600	69400
C_{DMI} (if not known)	1.3	1.3
C_{ice} (if not known)	2.37	2.04

Impact speeds for aft centerline thruster Table 4.12.2(3)

Icebreaker*	PC1	PC2
Longitudinal impact in main operational direction, in m/s	6	6
Longitudinal impact in reversing direction (pushing unit propeller hub or pulling unit cover end cap impact), in m/s	4	4
Transversal impact in bow first operation, in m/s	3	3
Transversal impact in stern first operation (double acting ship), in m/s	4	4

Impact speeds for aft wing, bow centerline and bow wing thrusters Table 4.12.2(4)

Icebreaker*	PC1	PC2
Longitudinal impact in main operational direction, in m/s	6	6
Longitudinal impact in reversing direction (pushing unit propeller hub or pulling unit cover end cap impact), in m/s	4	4
Transversal impact, in m/s	4	4

4.12.3 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.

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 is to be estimated for the load cases shown in Table 4.12.3(1), using the following equation. The parameter values for calculations are given in Table 4.12.3(2) and Table 4.12.3(3). The loads are to be applied as uniform distributed load or uniform pressure over the thruster surface. The design operation speed in ice can be derived from Table 4.12.3(2) or Table 4.12.3(3). Alternatively, the actual design operation speed in ice of the ship in question can be used.

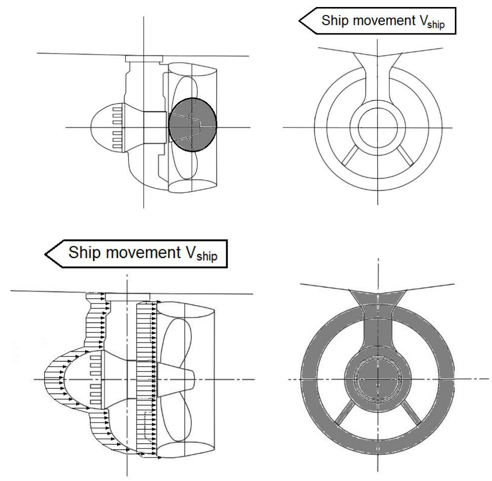
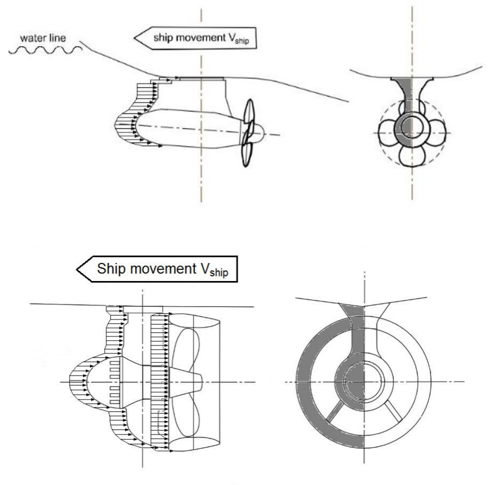
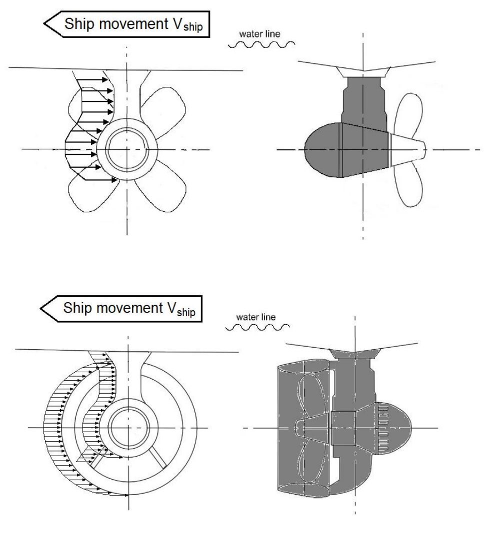
$$F_{tr} = 32 v_s^{0.66} H_r^{0.9} A_t^{0.74} \quad \text{kN}$$

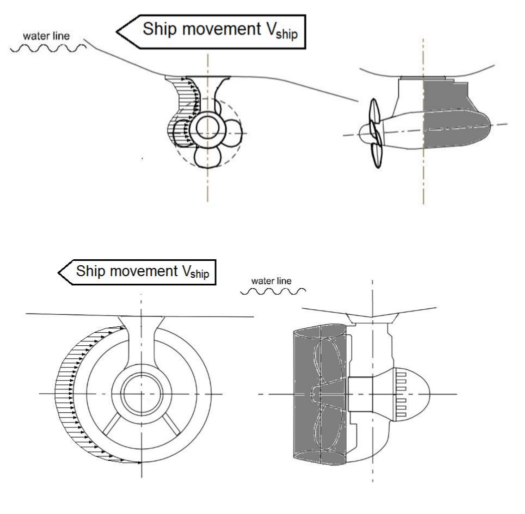
where: v_s —— ship speed, in m/s;

H_r —— design ridge thickness (the thickness of the consolidated layer is 18% of the total ridge thickness), in m;

A_t —— the projected area of the thruster, in m^2.

Load cases for ridge ice loads **Table 4.12.3(1)**

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

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 4.12.3.

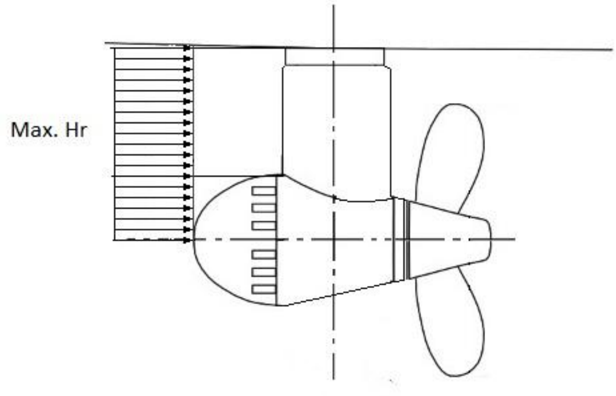


Figure 4.12.3 Schematic figure showing the reduction of the contact area by the maximum ridge thickness

Parameters for calculating maximum loads when the thruster penetrates an ice ridge (Aft thrusters. Bow first operation) Table 4.12.3(2)

Icebreaker*	PC1	PC2
Thickness of the design ridge consolidated layer, in m	4	3.5
Total thickness of the design ridge, H_r, in m	21	18
Initial ridge penetration speed (longitudinal loads), in m/s	4	4
Initial ridge penetration speed (transversal loads), in m/s	2	2

Parameters for calculating maximum loads when the thruster penetrates an ice ridge (Thruster first mode such as double acting ships) Table 4.12.3(3)

Icebreaker*	PC1	PC2
Total thickness of the design ridge, H_r, in m	21	18

Initial ridge penetration speed (longitudinal loads), in m/s	6	6
Initial ridge penetration speed (transversal loads), in m/s	3	3

4.12.4 Strength calculation criterion for static loads

The stresses on the thruster are to be calculated for the extreme once-in-a-lifetime loads described in 4.12. The nominal von Mises stresses on the thruster body are to have a safety margin of 1.3 against the yielding strength of the material. At areas of local stress concentrations, stresses are to have a safety margin of 1.0 against yielding. The slewing bearing, bolt connections and other components are to be able to maintain operability without incurring damage that requires repair when subject to the loads given in 4.12.2 and 4.12.3 multiplied by a safety factor of 1.3.

4.12.5 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 is to 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 are to be taken into account. In addition to this, the effect of ship attachment stiffness is to be modelled.

4.13 Alternative design

4.13.1 As an alternative, a comprehensive design study on the relevant mechanical devices can be submitted, and is to be requested to validate by an agreed test programme.

Section 5 STEERING SYSTEMS AND EQUIPMENT

5.1 General requirements

5.1.1 When installing two or more rudder devices, they are to be mechanically independent of each other.

5.1.2 At least two rudder pins are to be recommended to provide in order to transmit the load on the rudder blades.

5.1.3 Suitable rudder blade fastening devices (such as rudder stops/locking pins) are to be installed to ensure that the rudder blades are parallel to the centerline of the ship when astern running.

5.1.4 An ice knife is to be fitted at the rear of the rudder to avoid direct contact between the rudder (rudder stock and upper part of the rudder) and ice. The ice knife is to extend below LIWL, and be connected to the hull structure. Ice knife is to be designed to withstand the forces acting on its connected hull structural parts of the ship or the positions within the ship's area.

5.1.5 The non-conventional propulsion steering devices, such as rudder propellers, pods, etc., are to meet the requirements of 5.4.

5.2 Rudders

5.2.1 The dimensions of the rudder post, rudder stock and rudder pin are to comply with the relevant provisions of Chapter 3, Section 1 of PART TWO of the CCS Rules for Classification of Sea-going Steel Ships. When calculating rudder force, the ship speed is to be taken as the actual maximum service speed of the ship or the minimum ship speed corresponding to the ice class in Table 5.2.1, whichever is the greater. When using the minimum ship speed in Table 5.2.1, the coefficient K_2 (selected based on the type of rudder and profile shape) for calculating rudder force can be taken as 1.1.

Table 5.2.1

Icebreaker*	Calculated minimum ship speed (kn)
PC1	42
PC2	38

5.2.2 For astern case, the ship speed is to be taken as the actual maximum astern speed of the ship or half of the minimum ship speed corresponding to the ice class in Table 5.2.1, whichever is the greater.

5.3 Rudder actuators

5.3.1 The rudder actuators are to meet the following requirements:

(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:

Table 5.3.1(1)

Icebreaker*	PC1	PC2
factor	5	5

(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 Table 5.3.1(1).

5.3.2 The rudder actuator is to be protected by torque relief arrangements. 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 5.3.2.

Table 5.3.2

Icebreaker*	PC1&PC2
Turning speed（deg/s）	40

5.4 Non-conventional propulsion steering devices

5.4.1 In addition to the applicable requirements of Chapter 13, Section 1 of PART THREE of CCS Rules for Classification of Sea-going Steel Ships,the non-conventional propulsion steering devices are to meet the requirements of 5.4.2 and 5.4.3.

5.4.2 The design torque Q_{SG} of the steering device is not to be less than the calculated value using the following formula:

$$Q_{SG} = 0.75 \frac{Q_{\max}}{R} l \qquad \text{kNm}$$

where: Q_{max} —— the maximum ice torque acting on the propeller, see 4.4.8 of Section 4;

R —— radius of propeller, in m;

l —— distance from the rotation axis of the steering device to the rotation plane of the propeller, in m.

5.4.3 The steering device is to be equipped with effective protective measures to avoid the harmful effects of the following extreme torques, which may cause passive steering of the device:

(1) Ice block torque exceeding the design torque of the steering device;

(2) The torque caused by plastic bending of a certain blade in an unfavorable position.

5.4.4 The steering device is to be able to quickly return to normal working condition after effectively releasing the maximum torque specified in 5.4.2.

Appendix 1 DIRECT PREDICTION OF ICE LOADS BASED ON OPERATIONAL SCENARIOS

1 General requirements

1.1 With the development of innovative and large-scale polar ships, diverse operating modes and more complex ship-ice effects, the numerical calculation method of ice load based on ice mechanics analysis can effectively predict the ice load effects on the ship under various operating modes. Therefore, the numerical simulation analysis technology of ice loads is an essential supplementary means for the design of polar ships, such as icebreakers, etc..

1.2 Followed by the development of technology, numerical analysis techniques such as discrete element, finite element, peridynamics, circumferential crack method and smoothed particle hydrodynamics have provided a foundation for direct prediction of ice loads.

1.3 In order to meet the innovative design of polar equipment such as icebreakers, the requirements are formulated to guide the direct prediction of ice loads.

2 Operational Scenarios

2.1 Due to the complexity of polar sea ice environments and the diversity of ship ice operations, a reasonable operating scenario is the basis for ice load prediction.

2.2 In generally, operational scenarios are to be selected based on the expected goals and functions of the ships. The operational scenarios listed in 2.3~2.5 are for reference.

2.3 The typical operational scenarios are to include the followings:

(1) continuous ice breaking
(2) impacting ice
(3) steering in ice region
(4) navigating in ice channel
(5) navigating in crushed ice.

2.4 The special operational scenarios are to include the followings:

(1) penetrating ice ridge
(2) both port and starboard of the ship compressed by ice
(3) obliquely breaking ice
(4) shallow water operations in ice regions.

2.5 The extreme operational scenarios are to include the followings:

(1) Iceberg collision.

2.6 If necessary, additional scenarios beyond the above-mentioned can also be increased.

3 Analogue simulation

3.1 General requirements

3.1.1 Generally, ships are considered as rigid bodies for simulating ship-ice loads in order to obtain the spatiotemporal distribution of ice loads on hull.

3.1.2 The coupling analysis between structure and sea ice can also be used to obtain the spatiotemporal distribution of ice loads on hull and the response of the structure.

3.1.3 The effect of water on ship-ice is to be taken into consideration.

3.2 Sea ice models

3.2.1 A reasonable sea ice mechanics model is to be used to predict ship-ice loads, which can simulate the main sea ice failure modes, such as sea ice fracture, crushing,etc..

3.2.2 The physical and mechanical properties of sea ice generally include elastic modulus, compressive strength, bending strength, etc.. The physical and mechanical properties of sea ice in Table 3.2.2 are for reference.

Physical and mechanical properties of sea ice Table 3.2.2

Ice class	Mechanical properties of sea ice		
	Elastic modulus/GPa	Compressive strength/MPa	Bending strength/MPa
PC1	2.8	6.0	1.4
PC2	2.6	4.2	1.3

3.2.3 The ice ridge model is to include the keel, consolidation layer and dragon sail. The idealized cross-section is shown in Figure 3.2.3.

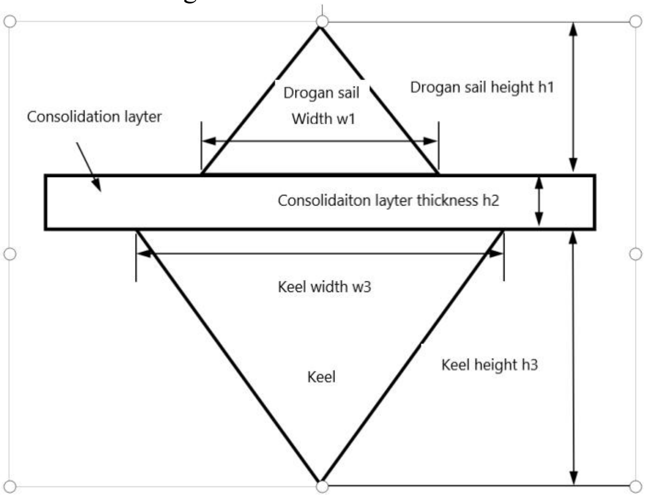


Figure 3.2.3 Idealized cross-section of ice ridges

3.3 Ship models

3.3.1 The wet surface models are to be able to reflect the linear characteristics of the ship, and model grids are to be evenly distributed.
3.3.2 When using self-propelled simulation, the models are also to include ship mass distribution, propulsion system bollard thrust, rudder parameters, etc..
3.3.3 When analyzing the coupling between the structure and sea ice, the models are also to include the internal hull structures.

3.4 Ice field construction

3.4.1 The ice field is to be constructed in combination with operational scenarios and design indicators, and the main considerations are generally as follows:
(1) The layer ice field is to include the range (length, width) of the field, sea ice thickness, ice speed (if necessary), etc.
(2) The crushed ice field is to include the range (length, width) of the field, sea ice density, ice block size and thickness, ice speed (if necessary), etc.
(3) The ice breaking waterway is to include the range (length, width) of the field, sea ice thickness and width of the ice breaking waterway, shape of the waterway edge, density of sea ice inside the waterway, ice block size and thickness, etc.
(4) When the model ship crosses an ice ridge or collides with an iceberg, the ice field is to include an ice ridge or iceberg model.
3.4.2 The range of the ice field is to be large enough to reduce the influence of boundary conditions on calculations. For rated speed and self-propelled ships, the width of the ice filed is not to be less than three times the width of the ship. For steering in ice regions, the width of the ice block is not to be less than three times the turning radius.

3.5 Simulation of ship-ice actions

3.5.1 According to the operational scenarios, mooring, rated speed, self-propelled, etc. for the ship can be selected. Usually, rated speed mode is used for continuous ice breaking, steering in ice region, etc. Self-propelled mode is used for impacting ice, crossing ice ridges, iceberg collisions, etc. Ship mooring mode can be used for both port and starboard compressed by ice.

3.5.2 For rated speed and self-propelled modes, sufficient interaction between the hull and sea ice is to be achieved, and the calculated period is to be so set to ensure that the ship navigation distance in the ice region is not to be less than 1.5 times the ship length. Simulation of crossing ice ridges is to ensure that the ship crosses the ice ridges completely. Simulation of iceberg collision is to ensure that the ship is in contact with the iceberg.

3.5.3 For the mooring mode, the calculated period is to ensure that the ice loads on the hull reaches a stable state.

4 Application

4.1 The time-history results of ice force at any position on the ship's waterline is to be available, and if necessary, it is to be possible to obtain the time-history results of ice force at any position onboard the ship below the waterline.

4.2 Further processing of the time-history results of ice forces is to be carried out to obtain ice loads that meet engineering evaluation requirements, generally including:

(1) ice resistance

(2) ice pressure

(3) ice bending moment

(4) time-history of ice force.

4.3 The criteria for applying the above-mentioned loads for strength analysis are to be given special consideration and approved by CCS.