TÜRK LOYDU

Chapter 33 - Polar Class Ships

2013

This latest edition incorporates all rule changes. This rule is totally revised. Changes after the publication of the rule are written in red colour.

Unless otherwise specified, these Rules apply to ships for which the date of contract for construction as defined in IACS PR No.29 is on or after 01st March 2013. New rules or amendments entering into force after the date of contract for construction are to be applied if required by those rules. See Rule Change Notices on TL website for details.

"General Terms and Conditions" of the respective latest edition will be applicable (see Rules for Classification and Surveys).

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Polar Class Ships

SECTION 1

POLAR CLASS DESCRIPTIONS AND APPLICATION

Page

A. Application

1. The Rules for Polar Ships apply to ships constructed of steel and intended for navigation in ice-infested polar waters, except ice breakers (see 3.).

2. Ships that comply with the Section 2 and Section 3 can be considered for a Polar Class notation as listed in Table 1.1. The requirements of Section 2 and Section 3 are in addition to the open water Türk Loydu Rules requirements. If the hull and machinery are constructed such as to comply with the requirements of different polar classes, then both the hull and machinery are to be assigned the lower of these classes in the classification certificate. Compliance of the hull or machinery with the requirements of a higher polar class is also to be indicated in the classification certificate or an appendix thereto.

3. Ships that are also to receive an “ICE-BREAKER” notation may have additional requirements and are to receive special consideration. “ICE-BREAKER" refers to any ship having an operational profile that includes escort or ice management functions, having powering and dimensions that allow it to undertake aggressive operations in ice-covered waters, and having a class certificate endorsed with this notation.

Notes:
Reference is made to the IMO Guidelines for Ships Operating in Arctic Ice-Covered Waters (MSC / Circ. 1056, MEPC / Circ.399, 23 December 2002).

B. Polar Classes

1. The Polar Class (PC) notations and descriptions are given in Table 1.1. It is the responsibility of the Owner to select an appropriate Polar Class. The descriptions in Table 1.1 are intended to guide owners, designers and administrations in selecting an appropriate Polar Class to match the requirements for the ship with its intended voyage or service.

2. The Polar Class notation is used throughout the IACS Unified Requirements for Polar Ships to convey the differences between classes with respect to operational capability and strength.

C. Upper and Lower Ice Waterlines

1. The upper and lower ice waterlines upon which the design of the vessel has been based is to be indicated in the classification certificate. The upper ice waterline (UIWL) is to be defined by the maximum draughts fore, amidships and aft. The lower ice waterline (LIWL) is to be defined by the minimum draughts fore, amidships and aft.

2. The lower ice waterline is to be determined with due regard to the vessel’s ice-going capability in the ballast loading conditions (e.g. propeller submergence).

Table 1.1 - Polar Class Descriptions

Polar Class	Ice Description (based on WMO Sea Ice Nomenclature)
PC 1	Year-round operation in all Polar waters
PC 2	Year-round operation in moderate multi-year ice conditions
PC 3	Year-round operation in second-year ice which may include multi- year ice inclusions.
PC 4	Year-round operation in thick first-year ice which may include old ice inclusions
PC 5	Year-round operation in medium first-year ice which may include old ice inclusions
PC 6	Summer/autumn operation in medium first-year ice which may include old ice inclusions
PC 7	Summer/autumn operation in thin first-year ice which may include old ice inclusions

SECTION 2

STRUCTURAL REQUIREMENTS FOR POLAR CLASS SHIPS

Page

A. Scope

1. The contents of this Section provide requirements for the structural arrangements of Polar Class ships. Each area of the hull and all appendages shall be strengthened to resist the global and local ice loads, as well as temperature, characteristics of their Polar Class.

Notes:

Reference is made to the IMO Guidelines for Ships Operating in Arctic Ice-Covered Waters (MSC / Circ. 1056, MEPC / Circ.399, 23 December 2002).

B. Hull Areas

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

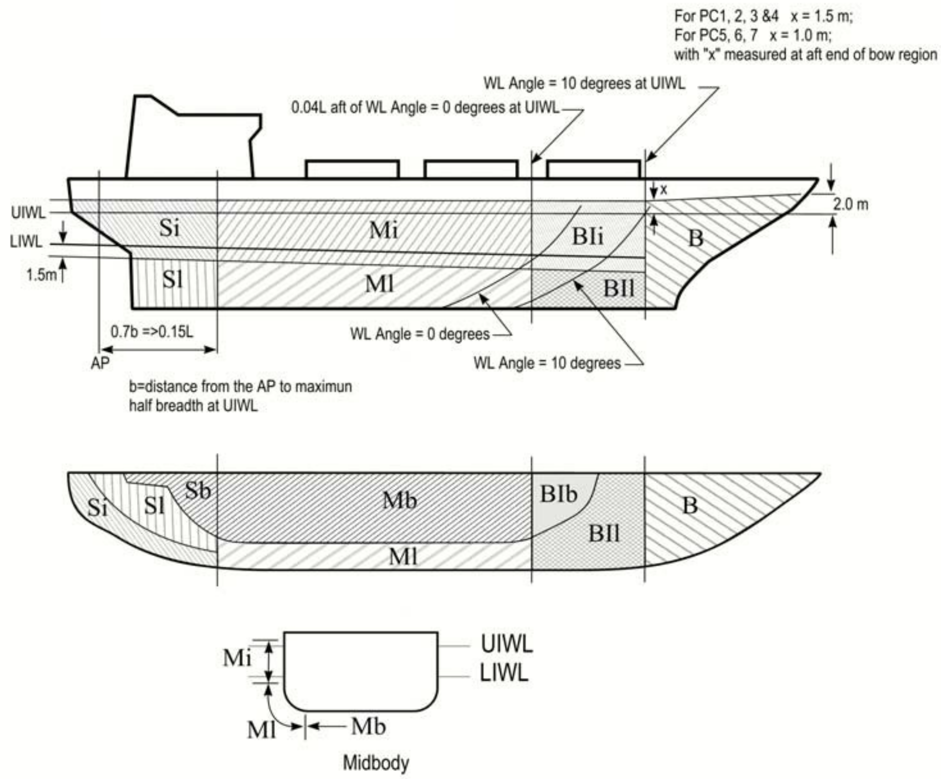


Figure 2.1 - Hull Area Extents

2. The upper ice waterline (UIWL) and lower ice waterline (LIWL) are as defined in Section 1, C.

3. Figure 2.1 notwithstanding, at no time is the boundary between the Bow and Bow Intermediate regions to be forward of the intersection point of the line of the stem and the ship baseline.

4. Figure 2.1 notwithstanding, the aft boundary of the Bow region need not be more than 0.45 L aft of the forward perpendicular (FP).

L = ship length as defined in Chapter 1, Hull, Section 1, H.2.1, but measured on the upper ice waterline (UIWL) [m]

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

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

7. All hull areas, including the locations of the UIWL and LIWL, are to be clearly indicated on the shell expansion submitted for approval.

C. Design Ice Loads

1. General

1.1 For ships of all Polar Classes, a glancing impact on the bow is the design scenario for determining the scantlings required to resist ice loads.

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).

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

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

1.5 Design ice forces calculated according to C.2 are only valid for vessels with icebreaking forms. Design ice forces for any other bow forms are to be specially considered.

1.6 Ship structures that are not directly subjected to ice loads may still experience inertial loads of stowed cargo and equipment resulting from ship/ice interaction. These inertial loads, calculated according to the design accelerations specified in Section 3, J.2 to J.4, are to be considered in the design of these structures.

2. Glancing Impact Load Characteristics

The parameters defining the glancing impact load characteristics are reflected in the Class Factors listed in Table 2.1.

Table 2.1 - Class Factors

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

2.1 Bow Area

2.1.1 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.

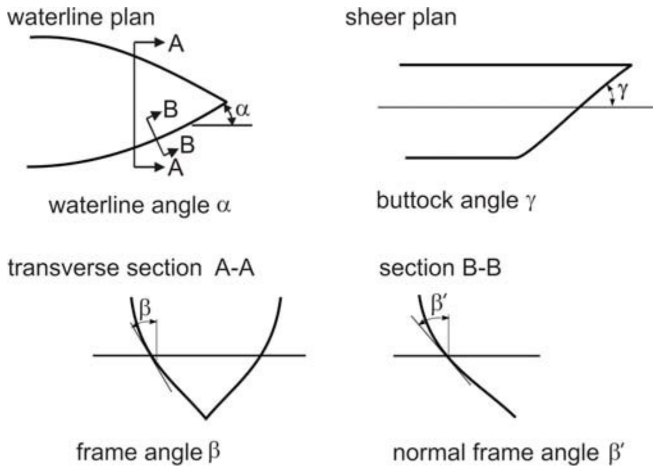


Figure 2.2 – Definition of Hull Angles

Note:

β' = normal frame angle at upper ice waterline [deg]

α = upper ice waterline angle [deg]

γ = buttock angle at upper ice waterline (angle of buttock line measured from horizontal) [deg]

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

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

2.1.2 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).

2.1.3 The Bow area load characteristics are determined as follows:

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

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

where

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

$$fa_{i,2} = 1.2 \cdot CF_F \;/\; (\sin(\beta'_i) \cdot CF_C \cdot D^{0.64}) \qquad (3)$$

$fa_{i,3}$ = 0.60 (4)

i = sub-region considered

L = ship length as defined in Chapter 1, Hull, Section 1, H.2.1, but measured on the upper ice waterline (UIWL) [m]

x = distance from the forward perpendicular (FP) to station under consideration [m]

α = waterline angle [deg], see Figure 2.2

β'_i = normal frame angle [deg], see Figure 2.2

D = ship displacement [kt], not to be taken less than 5 kt

CF_C = Crushing Failure Class Factor from Table 2.1

CF_F = Flexural Failure Class Factor from Table 2.1

(b) Force, F:

$F_i = fa_i \cdot CF_C \cdot D^{0.64}$ [MN] (5)

(c) Load patch aspect ratio, AR:

$AR_i = 7.46 \cdot \sin(\beta'_i) \geq 1.3$ (6)

(d) Line load, Q:

$Q_i = F_i^{0.61} \cdot CF_D / AR_i^{0.35}$ [MN/m] (7)

where

CF_D = Load Patch Dimensions Class Factor from Table 2.1

(e) Pressure, P:

$P_i = F_i^{0.22} \cdot CF_D^{2} \cdot AR_i^{0.3}$ [MPa]
(8)

2.2 Hull Areas Other Than the Bow

2.2.1 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 determined as follows:

(a) Force, F_{NonBow}:

$F_{NonBow} = 0.36 \cdot CF_C \cdot DF$ [MN] (9)

where

CF_C = Crushing Force Class Factor from Table 2.1

DF = ship displacement factor

= $D^{0.64}$ if $D \le CF_{DIS}$

= $CF_{DIS}^{0.64} + 0.10 \cdot (D - CF_{DIS})$ if $D > CF_{DIS}$

D = ship displacement [kt], not to be taken less than 10 kt

CF_{DIS} = Displacement Class Factor from Table 2.1

(b) Line Load, Q_{NonBow}:

$$Q_{NonBow} = 0.639 \cdot F_{NonBow}^{0.61} . CF_D \ \ [MN/m] \qquad (10)$$

where

F_{NonBow} = force from (9) [MN]

CF_D = Load Patch Dimensions Class Factor from Table 2.1

3. Design Load Patch

3.1. In the Bow area, and the Bow Intermediate Icebelt area for ships with class notation PC6 and PC7, the design load patch has dimensions of width, w_{Bow}, and height, b_{Bow}, defined as follows:

$$w_{Bow} = F_{Bow} / Q_{Bow} \ [m] \qquad (11)$$

$$b_{Bow} = Q_{Bow} / P_{Bow} \ [m] \qquad (12)$$

where

F_{Bow} = maximum force F_i in the Bow area from (5) [MN]

Q_{Bow} = maximum line load Q_i in the Bow area from (7) [MN/m]

P_{Bow} = maximum pressure P_i in the Bow area from (8) [MPa]

3.2 In hull areas other than those covered by C.3.1, the design load patch has dimensions of width, w_{NonBow}, and height, b_{NonBow}, defined as follows:

$$w_{NonBow} = F_{NonBow} / Q_{NonBow} \ [m] \qquad (13)$$

$$b_{NonBow} = w_{NonBow} / 3.6 \ [m] \qquad (14)$$

where

F_{NonBow} = force determined using (9) [MN]

Q_{NonBow} = line load determined using (10) [MN/m]

4. Pressure Within the Design Load Patch

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

$$P_{avg} = F / (b \cdot w) \text{ [MPa]} \quad (15)$$

where

F = F_{Bow} or F_{NonBow} as appropriate for the hull area under consideration [MN]

b = b_{Bow} or b_{NonBow} as appropriate for the hull area under consideration [m]

w = w_{Bow} or w_{NonBow} as appropriate for the hull area under consideration [m]

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

Table 2.2 - Peak Pressure Factors

Structural Member		**Peak Pressure Factor (PPF_i)**
Plating	Transversely-Framed	PPF_p = (1.8 - s) ≥ 1.2
	Longitudinally-Framed	PPF_p = (2.2 - 1.2 · s) ≥ 1.5
Frames in Transverse Framing Systems	With Load Distributing Stringers	PPF_t = (1.6 - s) ≥ 1.0
	With No Load Distributing Stringers	PPF_t = (1.8 - s) ≥ 1.2
Load Carrying Stringers Side and Bottom Longitudinals Web Frames		PPF_s = 1, if S_w ≥ 0.5 · w PPF_s = 2.0 - 2.0 · S_w / w, if S_w < (0.5 · w)
where: s = frame or longitudinal spacing [m] S_w = web frame spacing [m] w = ice load patch width [m]		

5. Hull Area Factors

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 2.3.

5.2 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.

5.3 Due to their increased manoeuvrability, ships having propulsion arrangements with azimuthing thruster(s) or "podded" propellers shall have specially considered Stern Icebelt (S_i) and Stern Lower (S_l) hull area factors.

Table 2.3 - Hull Area Factors (AF)

Hull Area		Area	Polar Class						
			PC1	PC2	PC3	PC4	PC5	PC6	PC7
Bow (B)	All	B	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Bow Intermediate (BI)	Icebelt	BI_i	0.90	0.85	0.85	0.80	0.80	1.00*	1.00*
	Lower	BI_l	0.70	0.65	0.65	0.60	0.55	0.55	0.50
	Bottom	BI_b	0.55	0.50	0.45	0.40	0.35	0.30	0.25
Midbody (M)	Icebelt	M_i	0.70	0.65	0.55	0.55	0.50	0.45	0.45
	Lower	M_l	0.50	0.45	0.40	0.35	0.30	0.25	0.25
	Bottom	M_b	0.30	0.30	0.25	**	**	**	**
Stern (S)	Icebelt	S_i	0.75	0.70	0.65	0.60	0.50	0.40	0.35
	Lower	S_l	0.45	0.40	0.35	0.30	0.25	0.25	0.25
	Bottom	S_b	0.35	0.30	0.30	0.25	0.15	**	**

Notes:
* *See C.1.3.*
** *Indicates that strengthening for ice loads is not necessary.*

D. Shell Plate Requirements

1. The required minimum shell plate thickness, t, is given by:

$$t = t_{net} + t_s \text{ [mm]} \quad (16)$$

where

t_{net} = plate thickness required to resist ice loads according to C [mm]

t_s = corrosion and abrasion allowance according to K.2 [mm]

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 ($\Omega \geq 70$ deg), including all bottom plating, i.e. plating in hull areas BI_b, M_b and S_b, the net thickness is given by:

$$t_{net} = 500 \cdot s \cdot ((AF \cdot PPF_p \cdot P_{avg}) / \sigma_y)^{0.5} / (1 + s / (2 \cdot b)) \text{ [mm]} \quad (17a)$$

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

$$t_{net} = 500 \cdot s \cdot ((AF \cdot PPF_p \cdot P_{avg}) / \sigma_y)^{0.5} / (1 + s / (2 \cdot l)) \text{ [mm]} \quad (17b)$$

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

$$t_{net} = 500 \cdot s \cdot ((AF \cdot PPF_p \cdot P_{avg}) / \sigma_y)^{0.5} \cdot (2 \cdot b / s - (b / s)^2)^{0.5} / (1 + s / (2 \cdot l)) \text{ [mm]} \quad (17c)$$

In the case of obliquely-framed plating (70 deg > Ω > 20 deg), linear interpolation is to be used.

where

Ω = smallest angle between the chord of the waterline and the line of the first level framing as illustrated in Figure 2.3 [deg].

s = transverse frame spacing in transversely-framed ships or longitudinal frame spacing in longitudinally-framed ships [m]

AF = Hull Area Factor from Table 2.3

PPF_p = Peak Pressure Factor from Table 2.2

P_{avg} = average patch pressure according to (15) [MPa]

σ_y = minimum upper yield stress of the material [N/mm^2]

b = height of design load patch [m], where b ≤ (*l* – s/4) in the case of (17a)

l = distance between frame supports, i.e. equal to the frame span as given in E.5, but not reduced for any fitted end brackets [m]. When a load-distributing stringer is fitted, the length *l* need not be taken larger than the distance from the stringer to the most distant frame support.

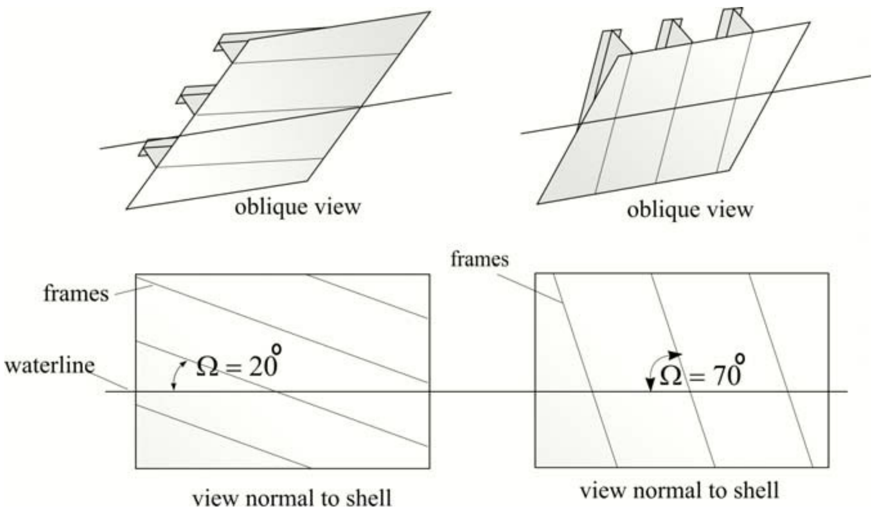


Figure 2.3 - Shell Framing Angle Ω

E. Framing - General

1. Framing members of Polar class ships are to be designed to withstand the ice loads defined in C.

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 2.1. Where load-distributing stringers have been fitted, the arrangement and scantlings of these are to be specially considered. In general, load-distributing stringers shall be located at or close to mid-span of transverse frames, have a web height not less than 80% of transverse frames and have at least the same web net thickness.

3. The strength of a framing member is dependent upon the fixity that is provided at its supports. Fixity can be assumed where framing members are either continuous through the support or attached to a supporting section with a connection bracket. In other cases, simple support is to be assumed unless the connection can be demonstrated to provide significant rotational restraint. Fixity is to be ensured at the support of any framing which terminates within an ice-strengthened area. See also P.1.

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 specially considered.

5. The design span of a framing member is to be determined on the basis of its moulded length. If brackets are fitted, the design span may be reduced in accordance with the Figure 2.4. Brackets are to be configured to ensure stability in the elastic and post-yield response regions.

6. When calculating the section modulus and shear area of a framing member, net thicknesses of the web, flange (if fitted) and attached shell plating are to be used. The shear area of a framing member may include that material contained over the full depth of the member, i.e. web area including portion of flange, if fitted, but excluding attached shell plating.

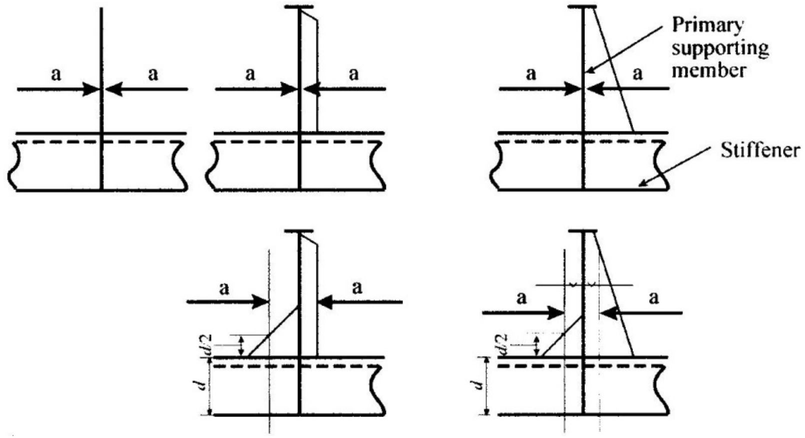


Figure 2.4 - Design span of framing member

7. The actual net effective shear area, A_w, of a framing member is given by:

$$A_w = h \cdot t_{wn} \cdot \sin \varphi_w / 100 \ [cm^2] \qquad (18)$$

h = height of stiffener [mm], see Figure 2.5

t_{wn} = net web thickness [mm]

= $t_w - t_c$

t_w = as built web thickness [mm], see Figure 2.5

t_c = corrosion deduction [mm] to be subtracted from the web and flange thickness (not less than t_s as required by K.3)

φ_w = smallest angle between shell plate and stiffener web, measured at the midspan of the stiffener, see Figure 2.5. The angle φ_w may be taken as 90 degrees provided the smallest angle is not less than 75 degrees.

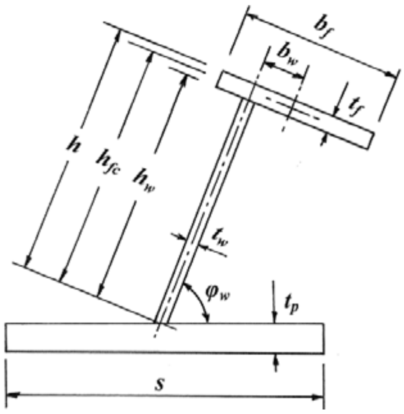


Figure 2.5 - Stiffener geometry

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, is given by:

$$Z_p = A_{pn}.t_{pn}/20 + \frac{h_w^2 . t_{wn} .\sin\varphi_w}{2000} + A_{fn}.\left(h_{fc}.\sin\varphi_w - b_w.\cos\varphi_w\right)/10 \quad \left[cm^3\right] \tag{19}$$

h, t_{wn}, t_c, and φ_w are as given in E.7 and s as given in D.2.

A_{pn} = net cross-sectional area of the local frame [cm^2]

t_{pn} = fitted net shell plate thickness [mm] (shall comply with t_{net} as required by D.2)

h_w = height of local frame web [mm], see Figure 2.5

A_{fn} = net cross-sectional area of local frame flange [cm^2]

h_{fc} = height of local frame measured to centre of the flange area [mm], see Figure 2.5

b_w = distance from mid thickness plane of local frame web to the centre of the flange area [mm], see Figure 2.5

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:

$$z_{na} = (100 \cdot A_{fn} + h_w \cdot t_{wn} - 1000 \cdot t_{pn} \cdot s)/(2 \cdot t_{wn}) \text{ [mm]} \tag{20}$$

and the net effective plastic section modulus, Z_p, is given by:

$$Z_p = t_{pn}.s.\left(z_{na} + t_{pn}/2\right).\sin\varphi_w + \frac{\left[\left(h_w - z_{na}\right)^2 + z_{na}^2\right]\cdot t_{wn}\cdot\sin\varphi_w}{2000} + \frac{A_{fn}\cdot\left[\left(h_{fc} - z_{na}\right)\cdot\sin\varphi_w - b_w\cdot\cos\varphi_w\right]}{10} \text{ [cm}^3\text{]} \tag{21}$$

9. In the case of oblique framing arrangement (70 deg > Ω > 20 deg, where Ω is defined as given in D.2), linear interpolation is to be used.

F. Framing - Transversely Framed Side Structures and Bottom Structures

1. The local frames in transversely-framed side structures and in bottom structures (i.e. hull areas BI_b, 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.

2. The actual net effective shear area of the frame, A_w, as defined in E.7, is to comply with the following condition: $A_w \geq A_t$, where:

$$A_t = 100^2 \cdot 0.5 \cdot LL \cdot s \cdot (AF \cdot PPF_t \cdot P_{avg}) / (0.577 \cdot \sigma_y) \ [cm^2] \quad (22)$$

where

LL = length of loaded portion of span

= lesser of a and b [m]

a = frame span as defined in E.5 [m]

b = height of design ice load patch according to (12) or (14) [m]

s = transverse frame spacing [m]

AF = Hull Area Factor from Table 2.3

PPF_t = Peak Pressure Factor from Table 2.2

P_{avg} = average pressure within load patch according to (15) [MPa]

σ_y = minimum upper yield stress of the material [N/mm²]

3. The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in E.8, is to comply with the following condition: $Z_p \geq Z_{pt}$, where Z_{pt} is to be the greater calculated on the basis of two load conditions: a) ice load acting at the midspan of the transverse frame, and b) the ice load acting near a support. The A_1 parameter in (23) reflects the two conditions:

$$Z_{pt} = 100^3 \cdot LL \cdot Y \cdot s \cdot (AF \cdot PPF_t \cdot P_{avg}) \cdot a \cdot A_1 / (4 \cdot \sigma_y) \ [cm^3] \quad (23)$$

where

A_F, PPF_t, P_{avg}, LL, b, s, a and σ_y are as given in F.2.

$Y = 1 - 0.5 \cdot (LL / a)$

A_1 = maximum of

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

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

j = 1 for framing with one simple support outside the ice-strengthened areas

= 2 for framing without any simple supports

$a_1 = A_t / A_w$

A_t = minimum shear area of transverse frame as given in F.2 [cm^2]

A_w = effective net shear area of transverse frame (calculated according to E.7) [cm^2]

$k_w = 1 / (1 + 2 \cdot A_{fn} / A_w)$ with A_{fn} as given in E.8

$k_z = z_p / Z_p$ in general

= 0.0 when the frame is arranged with end bracket

z_p = sum of individual plastic section moduli of flange and shell plate as fitted [cm^3]

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

b_f = flange breadth [mm], see Figure 2.5

t_{fn} = net flange thickness [mm]

= $t_f – t_c$ (t_c as given in K.3)

t_f = as-built flange thickness [mm], see Figure 2.5

t_{pn} = the fitted net shell plate thickness [mm] (not to be less than t_{net} as given in D.2)

b_{eff} = effective width of shell plate flange [mm]

= $500 \cdot s$

Z_p = net effective plastic section modulus of transverse frame (calculated according to E.8) [cm^3]

4. The scantlings of the frame are to meet the structural stability requirements of I.

G. Framing - Side Longitudinals (Longitudinally Framed Ships)

1. Side longitudinals 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. The actual net effective shear area of the frame, A_w, as defined in E.7, is to comply with the following condition: $A_w \geq A_L$, where:

$$A_L = 100^2 \cdot (AF \cdot PPF_s \cdot P_{avg}) \cdot 0.5 \cdot b_1 \cdot a / (0.577 \cdot \sigma_y) \ [cm^2] \qquad (24)$$

where

AF = Hull Area Factor from Table 2.3

PPF_s = Peak Pressure Factor from Table 2.2

P_{avg} = average pressure within load patch according to (15) [MPa]

$b_1 = k_o \cdot b_2$ [m]

$k_o = 1 - 0.3 / b'$

$b' = b / s$

b = height of design ice load patch from (12) or (14) [m]

s = spacing of longitudinal frames [m]

$b_2 = b \cdot (1 - 0.25 \cdot b')$ [m], if $b' < 2$

$= s$ [m], if $b' \geq 2$

a = longitudinal design span as given in E.5 [m]

σ_y = minimum upper yield stress of the material [N/mm²]

3. The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in E.8, is to comply with the following condition: $Z_p \geq Z_{pL}$, where:

$$Z_{pL} = 100^3 \cdot (AF \cdot PPF_s \cdot P_{avg}) \cdot b_1 \cdot a^2 \cdot A_4 / (8 \cdot \sigma_y) \ [cm^3] \qquad (25)$$

where

AF, PPF_s, P_{avg}, b_1, a and σ_y are as given in G.2

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

$a_4 = A_L / A_w$

A_L = minimum shear area for longitudinal as given in G.2 [cm²]

A_w = net effective shear area of longitudinal (calculated according to E.7) [cm^2]

k_{wl} = $1 / (1 + 2 \cdot A_{fn} / A_w)$ with A_{fn} as given in E.8

4. The scantlings of the longitudinals are to meet the structural stability requirements of I.

H. Framing - Web Frame and Load-Carrying Stringers

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

2. Web frames and load-carrying stringers are to be dimensioned such that the combined effects of shear and bending, nowhere exceed the minimum upper yield stress of the material σ_y. Where these members form part of a structural grillage system, appropriate methods of analysis are to be used. Where the structural configuration is such that members do not form part of a grillage system, the appropriate peak pressure factor (PPF) from Table 2.2 is to be used.

3. Special attention is to be paid to the shear capacity in way of lightening holes and cut-outs in way of intersecting members.

4. The scantlings of web frames and load-carrying stringers are to meet the structural stability requirements of I.2.

I. Framing - Structural Stability

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: $h_w / t_{wn} \leq 282 / (\sigma_y)^{0.5}$

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

where

h_w = web height

t_{wn} = net web thickness

σ_y = minimum upper yield stress of the material [N/mm^2]

2. Framing members for which it is not practicable to meet the requirements of I.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:

$$t_{wn} = 2.63x10^{-3}.c_1\sqrt{\varphi_y/(5.34 + 4.(c_1/c_2)^2)} \quad \text{[mm]} \qquad (26)$$

where

$c_1 = h_w - 0.8 \cdot h$ [mm]

h_w = web height of stringer / web frame [mm] (see Figure 2.6)

h = height of framing member penetrating the member under consideration (0 if no such framing member) [mm] (see Figure 2.6)

c_2 = spacing between supporting structure oriented perpendicular to the member under consideration [mm] (see Figure 2.6)

σ_y = minimum upper yield stress of the material [N/mm^2]

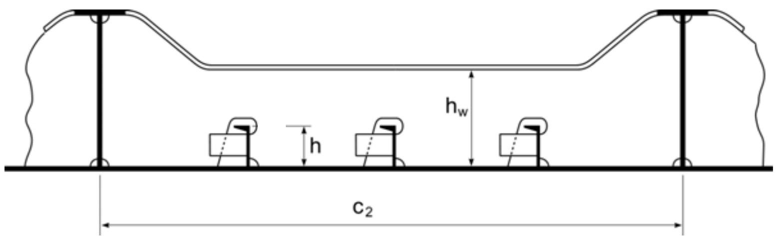


Figure 2.6 - Parameter Definition for Web Stiffening

3. In addition, the following is to be satisfied:

$t_{wn} \geq 0.35 \cdot t_{pn} \cdot (\sigma_y / 235)^{0.5}$

where

σ_y = minimum upper yield stress of the shell plate in way of the framing member [N/mm^2]

t_{wn} = net thickness of the web [mm]

t_{pn} = net thickness of the shell plate in way of the framing member [mm]

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

(i) The flange width, b_f [mm], shall not be less than five times the net thickness of the web, t_{wn}.

(ii) The flange outstand, b_{out} [mm], shall meet the following requirement:

$b_{out} / t_{fn} \leq 155 / (\sigma_y)^{0.5}$

$b_{out} = b_f / 2 + b_w - t_w / 2$ [mm] (see Figure 2.5)

where

t_{fn} = net thickness of flange [mm]

σ_y = minimum upper yield stress of the material [N/mm^2]

J. Plated Structures

1. Plated structures are those stiffened plate elements in contact with the hull and subject to ice loads. These requirements are applicable to an inboard extent which is the lesser of:

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

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

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.

3. The stability of the plated structure is to adequately withstand the ice loads defined in C.

K. Corrosion/Abrasion Additions and Steel Renewal

1. Effective protection against corrosion and ice-induced abrasion is recommended for all external surfaces of the shell plating for all Polar ships.

2. The values of corrosion/abrasion additions, t_s, to be used in determining the shell plate thickness for each Polar Class are listed in Table 2.4.

3. Polar ships are to have a minimum corrosion/abrasion addition of t_s = 1.0 mm applied to all internal structures within the ice-strengthened hull areas, including plated members adjacent to the shell, as well as stiffener webs and flanges.

Table 2.4 - Corrosion/Abrasion Additions for Shell Plating

Hull Area	t_s [mm]					
	With Effective Protection			Without Effective Protection		
	PC1 - PC3	PC4 & PC5	PC6 & PC7	PC1 - PC3	PC4 & PC5	PC6 & PC7
Bow; Bow Intermediate Icebelt	3.5	2.5	2.0	7.0	5.0	4.0
Bow Intermediate Lower; Midbody & Stern Icebelt	2.5	2.0	2.0	5.0	4.0	3.0
Midbody & Stern Lower; Bottom	2.0	2.0	2.0	4.0	3.0	2.5

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

L. Materials

1. Plating materials for hull structures are to be not less than those given in Tables 2.6 and 2.7 based on the as-built thickness of the material, the Polar ice class notation assigned to the ship and the Material Class of structural members according to L.2.

Table 2.5 - Material Classes for Structural Members of Polar Ships

Structural Members	Material Class
Shell plating within the bow and bow intermediate icebelt hull areas (B, B_{Ii})	II
All weather and sea exposed SECONDARY and PRIMARY, as defined in Chapter 1, Hull, Section 3, Table 3.2, structural members outside 0.4L amidships	I
Plating materials for stem and stern frames, rudder horn, rudder, propeller nozzle, shaft brackets, ice skeg, ice knife and other appendages subject to ice impact loads	II
All inboard framing members attached to the weather and sea-exposed plating, including any contiguous inboard member within 600 mm of the plating	I
Weather-exposed plating and attached framing in cargo holds of ships which by nature of their trade have their cargo hold hatches open during cold weather operations	I
All weather and sea exposed SPECIAL, as defined in Chapter 1, Hull, Section 3, Table 3.2, structural members within 0.2L from FP	II

2. Material classes specified in Chapter 1, Hull, Section 3, Table 3.2 are applicable to polar ships regardless of the ship's length. In addition, material classes for weather and sea exposed structural members and for members attached to the weather and sea exposed plating are given in Table 2.5. Where the material classes in Table 2.5 and those in Chapter 1, Hull, Section 3, Table 3.2 differ, the higher material class is to be applied.

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 2.7, are to be obtained from Chapter 1, Hull, Section 3, Table 3.7 based on the Material Class for Structural Members in Table 2.5 above, regardless of Polar Class.

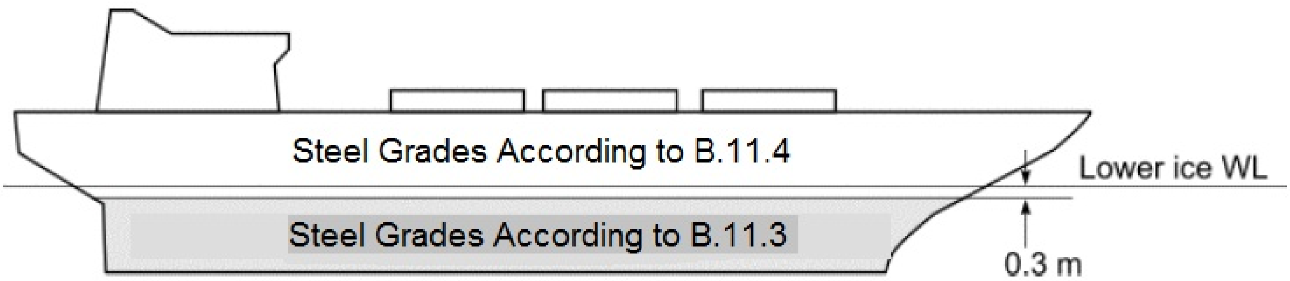


Figure 2.7 - Steel Grade Requirements for Submerged and Weather Exposed Shell Plating

4. Steel grades for all weather exposed plating of hull structures and appendages situated above the level of 0.3 m below the lower ice waterline, as shown in Figure 2.7, are to be not less than given in Table 2.6.

Table 2.6 - Steel Grades for Weather Exposed Plating

Thickness, t [mm]	Material Class I				Material Class II				Material Class III					
	PC1-5		PC6&7		PC1-5		PC6&7		PC1-3		PC4&5		PC6&7	
	MS	HT	MS	HT	MS	HT	MS	HT	MS	HT	MS	HT	MS	HT
t ≤ 10	B	AH	B	AH	B	AH	B	AH	E	EH	E	EH	B	AH
10 < t ≤ 15	B	AH	B	AH	D	DH	B	AH	E	EH	E	EH	D	DH
15 < t ≤ 20	D	DH	B	AH	D	DH	B	AH	E	EH	E	EH	D	DH
20 < t ≤ 25	D	DH	B	AH	D	DH	B	AH	E	EH	E	EH	D	DH
25 < t ≤ 30	D	DH	B	AH	E	EH2	D	DH	E	EH	E	EH	E	EH
30 < t ≤ 35	D	DH	B	AH	E	EH	D	DH	E	EH	E	EH	E	EH
35 < t ≤ 40	D	DH	D	DH	E	EH	D	DH	F	FH	E	EH	E	EH
40 < t ≤ 45	E	EH	D	DH	E	EH	D	DH	F	FH	E	EH	E	EH
45 < t ≤ 50	E	EH	D	DH	E	EH	D	DH	F	FH	F	FH	E	EH

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.*

5. Steel grades for all inboard framing members attached to weather exposed plating are to be not less than given in Table 2.7. This applies to all inboard framing members as well as to other contiguous inboard members (e.g. bulkheads, decks) within 600 mm of the exposed plating.

Table 2.7 - Steel Grades for Inboard Framing Members Attached to Weather Exposed Plating

Thickness t, mm	PC1 - PC5		PC6 & PC7	
	MS	HT	MS	HT
t ≤ 20	B	AH	B	AH
20 < t ≤ 35	D	DH	B	AH
35 < t ≤ 45	D	DH	D	DH
45 < t ≤ 50	E	EH	D	DH

6. Castings are to have specified properties consistent with the expected service temperature for the cast component.

M. Longitudinal Strength

1. Application

Ice loads need only 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. Design Vertical Ice Force at the Bow

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

$$F_{IB} = \text{minimum}\,(F_{IB,1};\, F_{IB,2})\ [MN] \tag{27}$$

where

$$F_{IB,1} = 0.534 \cdot K_I^{0.15} \cdot \sin^{0.2}(\gamma_{stem}) \cdot (D \cdot K_h)^{0.5} \cdot CF_L\ [MN] \tag{28}$$

$$F_{IB,2} = 1.20 \cdot CF_F\ [MN] \tag{29}$$

K_I = indentation parameter = K_f / K_h

a) for the case of a blunt bow form

$K_f = (2 \cdot C \cdot B^{1-eb} / (1 + e_b))^{0.9} \cdot \tan(\gamma_{stem})^{-0.9 \cdot (1 + eb)}$

b) for the case of wedge bow form (α_{stem} < 80 deg), e_b =1 and the above simplifies to

$K_f = (\tan(\alpha_{stem}) / \tan^2(\gamma_{stem}))^{0.9}$

K_h = 0.01 · A_{wp} [MN/m]

CF_L = Longitudinal Strength Class Factor from Table 2.1

e_b = bow shape exponent which best describes the waterplane (see Figures 2.8 and 2.9)

= 1.0 for a simple wedge bow form

= 0.4 to 0.6 for a spoon bow form

= 0 for a landing craft bow form

An approximate e_b determined by a simple fit is acceptable.

γ_{stem} = stem angle to be measured between the horizontal axis and the stem tangent at the upper ice waterline [deg] (buttock angle as per Figure 2.2 measured on the centreline)

α_{stem} = waterline angle measured in way of the stem at the upper ice waterline (UIWL) [deg] (see Figure 2.8)

$C = 1 / (2 \cdot (L_B / B)^{eb})$

B = greatest ship moulded breadth [m]

L_B = bow length used in the equation $y = B / 2 \cdot (x/L_B)^{eb}$ [m] (see Figures 2.8 and 2.9)

D = ship displacement [kt], not to be taken less than 10 kt

A_{wp} = ship waterplane area [m^2]

CF_F = Flexural Failure Class Factor from Table 2.1

Where applicable, draught dependent quantities are to be determined at the waterline corresponding to the loading condition under consideration.

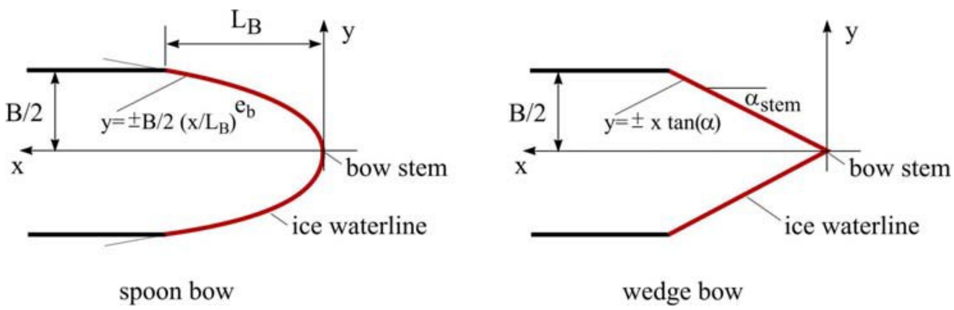


Figure 2.8 - Bow Shape Definition

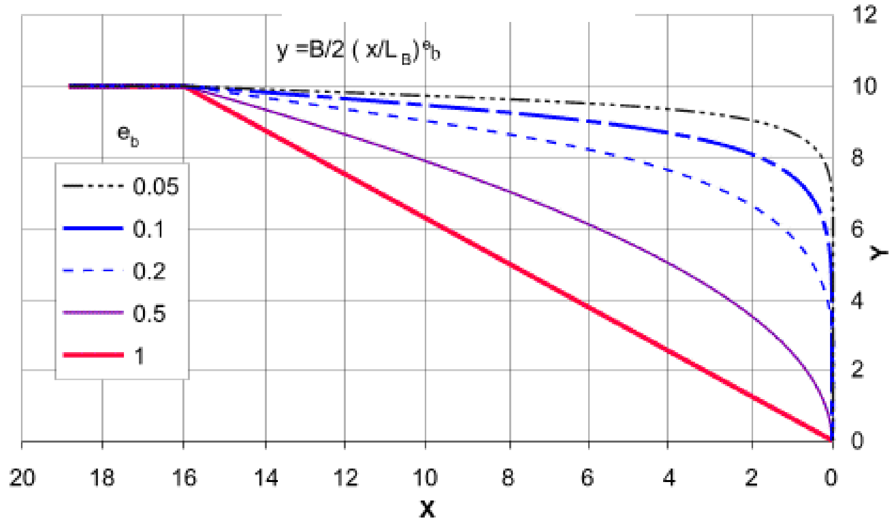


Figure 2.9 - Illustration of e_b Effect on the Bow Shape for B = 20 and L_B =16

3. Design Vertical Shear Force

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

$F_I = C_f \cdot F_{IB}$ [MN] (30)

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 and 0.6L from aft

C_f = 1.0 between 0.9 L from aft and the forward end of L

(b) Negative shear force

C_f = 0.0 at the aft end of L

C_f = -0.5 between 0.2 L and 0.6L from aft

C_f = 0.0 between 0.8 L from aft and the forward end of L

Intermediate values are to be determined by linear interpolation

3.2 The applied vertical shear stress, τ_a, is to be determined along the hull girder in a similar manner as in Chapter 1, Hull, Section 3, Table 3.24 by substituting the design vertical ice shear force for the design vertical wave shear force.

4. Design Vertical Ice Bending Moment

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

$$M_I = 0.1 \cdot C_m \cdot L \cdot \sin^{-0.2}(\gamma_{stem}) \cdot F_{IB} \text{ [MNm]} \qquad (31)$$

where

L = ship length as defined in Chapter 1, Hull, Section 1, H.2.1, but measured on the upper ice waterline [UIWL] [m]

γ_{stem} is as given in M.2.

F_{IB} = design vertical ice force at the bow [MN]

C_m = longitudinal distribution factor for design vertical ice bending moment to be taken as follows:

C_m = 0.0 at the aft end of L

C_m = 1.0 between 0.5L and 0.7L from aft

C_m = 0.3 at 0.95L from aft

C_m = 0.0 at the forward end of L

Intermediate values are to be determined by linear interpolation.

Where applicable, draught dependent quantities are to be determined at the waterline corresponding to the loading condition under consideration.

4.2 The applied vertical bending stress, σ_a, is to be determined along the hull girder in a similar manner as in Chapter 1, Hull, Section 3, Table 3.23a and 3.23b, 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 maximum sagging moment.

5. Longitudinal Strength Criteria

5.1 The strength criteria provided in Table 2.8 are to be satisfied. The design stress is not to exceed the permissible stress.

Table 2.8 - Longitudinal Strength Criteria

Failure Mode	Applied Stress	Permissible Stress when $\sigma_y / \sigma_u \le 0.7$	Permissible Stress when $\sigma_y / \sigma_u > 0.7$
Tension	σ_a	$\eta \cdot \sigma_y$	$\eta \cdot 0.41\ (\sigma_u + \sigma_y)$
Shear	τ_a	$\eta \cdot \sigma_y / (3)^{0.5}$	$\eta \cdot 0.41\ (\sigma_u + \sigma_y) / (3)^{0.5}$
Buckling	σ_a	σ_c for plating and for web plating of stiffeners $\sigma_c / 1.1$ for stiffeners	
	τ_a	τ_c	

where

σ_a = applied vertical bending stress [N/mm^2]

τ_a = applied vertical shear stress [N/mm^2]

σ_y = minimum upper yield stress of the material [N/mm^2]

σ_u = ultimate tensile strength of material [N/mm^2]

σ_c = critical buckling stress in compression, according to Chapter 1, Hull, Section 3, Table 3.19 [N/mm^2]

τ_c = critical buckling stress in shear, according to Chapter 1, Hull, Section 3, Table 3.20 [N/mm^2]

η = 0.8

N. Stem and Stern Frames

1. The stem is to be shaped in such a way that it can break ice effectively. The thickness of the stem plating is not to be less than 1.3 times the thickness of the adjacent shell plating.

2. The stern frame is to be shaped in such a way that it can displace broken ice effectively.

3. For Polar Class ships requiring **ICE-B3** or **ICE-B4** equivalency (see Chapter 1, Hull, Section 14.A), the requirements of Chapter 1, Hull, Section 14.D.7 to D.10 need also to be observed.

O. Appendages

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. All manoeuvring arrangements, e.g. rudder stocks, rudder couplings, rudder bearings, rudder bodies, ice horns, propeller nozzles, podded propulsors, azimuth thrusters etc., are to be dimensioned to withstand the design ice force defined in C.2.2.1, adjusted by the appropriate hull area factor in Table 2.3. Alternative design ice force definitions, including reduced design ice forces below the lower ice waterline (LIWL) and longitudinal design ice forces (where applicable), may be agreed with **TL**.

3. The design ice force shall be applied at locations where the capacity of these structural members under the combined effects of bending, shear and torsion (where applicable) is minimised. A stress analysis shall demonstrate that equivalent stresses in the structure nowhere exceed the minimum upper yield stress of the material σ_y.

4. The thickness of rudder and nozzle plating is to be determined according to D.

5. Rudders and rudder stocks shall be protected from ice loads with an ice horn which is fitted directly abaft the rudder and which extends a minimum distance of 1.5 CF_D [m] below the lower ice waterline (LIWL) defined in Section 1, C. When dimensioning the ice horn and the uppermost part of the rudder, it may be assumed that the design ice patch is acting over both structures, i.e. the design ice force defined in O.2 may be distributed between them.

6. When bilge keels are fitted, it is required that they be divided into several independent lengths to limit possible damage to the shell.

P. Local Details

1. The intersection and termination of framing members at supporting structures, i.e. stringers, web frames, decks or bulkheads, shall be arranged to enable the transfer of ice-induced loads (bending moments and shear forces), generally by means of direct welding, collar plates, lugs, connection brackets or heel stiffeners.

2. The loads carried by a member in way of cut-outs are not to cause instability. Where necessary, the structure is to be stiffened.

Q. Direct Calculations

1. Direct calculations are not to be utilised as an alternative to the analytical procedures prescribed in these Rules.

2. Where direct calculations are used to check the strength of structural arrangements (e.g. arrangements which may need to be specially considered), the load patch specified in C shall be applied at locations where the capacity of these systems under the combined effects of bending and shear is minimized.

3. The calculations shall demonstrate that equivalent stresses in the structure, under the combined effects of shear and bending, nowhere exceed the minimum upper yield stress of the material σ_y.

R. Welding

1. All welding within ice-strengthened areas is to be of the double continuous type.

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

SECTION 3

MACHINERY REQUIREMENTS FOR POLAR CLASS SHIPS

Page

A. General

1. Scope

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

The vessel operating conditions are defined in Section 1.

The requirements herein are additional to those applicable for the basic class.

Note:
Reference is made to Part A (Construction Provisions) of the IMO Guidelines for Ships Operating in Arctic Ice-Covered Waters (MSC/Circ. 1056, MEPC/Circ. 399, 23 December 2002) for additional guidance concerning machinery arrangements.

2. Definitions

The following main parameters are used:

CP = Controllable pitch propeller

d = Propeller hub diameter [m]

D = Diameter of propeller [m]

EAR = Expanded blade area ratio [-]

FP = Fixed pitch propeller

LIWL = Minimum ballast waterline in ice

n = Rotational propeller speed [rps]

N = Number of loads

R = Radius of the propeller [m]

S = Safety factor [-]

z = Number of propeller blades

ϕ = Propeller rotation angle [degrees]

μ = Friction coefficient [-]

3. Drawings and particulars to be submitted

(i) Details of the environmental conditions and the required ice class for the machinery, if different from ship's ice class.

(ii) Detailed drawings of the main propulsion machinery. Description of the main propulsion, steering, emergency and essential auxiliaries are to include operational limitations. Information on essential main propulsion load control functions.

(iii) Description detailing how main, emergency and auxiliary systems are located and protected to prevent problems from freezing, ice and snow and evidence of their capability to operate in intended environmental conditions.

(iv) Calculations and documentation indicating compliance with the requirements of this Section.

B. Materials

1. Materials exposed to sea water

Materials exposed to sea water, such as propeller blades, propeller hub and blade bolts shall have an elongation not less than 15% on a test piece the length of which is five times the diameter.

Charpy V impact test shall be carried out for other than bronze and austenitic steel materials. Test pieces taken from the propeller castings shall be representative of the thickest section of the blade. An average impact energy value of 20 J taken from three Charpy V tests is to be obtained at minus 10 ºC.

2. Materials exposed to sea water temperature

Materials exposed to sea water temperature shall be of steel or other approved ductile material. An average impact energy value of 20 J taken from three tests is to be obtained at minus 10 ºC. This requirement applies to blade bolts, CP-mechanisms, shaft bolts, strut-pod connecting bolts, etc. This does not apply to surface hardened components, such as bearings and gear teeth. For definition of structural boundaries exposed to sea water temperature see Section 2, Figure 2.7.

3. Material exposed to low air temperature

Materials of essential components exposed to low air temperature shall be of 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. This does not apply to surface hardened components, such as bearings and gear teeth. For definition of structural boundaries exposed to sea water temperature see Section 2, Figure 2.7.

C. Design Principles

1. General

All components and systems shall be designed such that the task of the ship in the relevant ice and weather conditions can be fulfilled with reasonable safety. The principle of the pyramid of strength has to be followed.

2. Ship operation in case of damage

Single screw vessels classed PC1 to PC5 inclusive shall have means provided to ensure sufficient vessel operation in the case of propeller damage including CP-mechanism (i.e. pitch control mechanism). Sufficient ship operation means that the ship shall be able to reach safe harbour (safe location) where repair can be undertaken in case of propeller damage. This may be achieved either by a temporary repair at sea, or by towing assistance provided the availability can be demonstrated (condition for approval, to be mentioned in Class Certificate).

3. Propulsion line components

The strength of the propulsion line shall be designed:

a) For maximum loads in D.2. (for open and ducted propellers respectively) and E.1.;

b) Such that the plastic bending of a propeller blade shall not cause damages in other propulsion line components;

c) With sufficient fatigue strength as determined in e.g. D.2.2, E.2.2 and F.1.2.3.

4. Reverse operation of propellers

Means shall be provided to free a stuck propeller by turning backwards. This means that a plant intended for unidirectional rotation is to be equipped at least with a sufficient turning gear that is capable of turning the propeller in reverse direction.

5. Drainage

Systems, subject to damage by freezing, shall be drainable.

D. Propeller

1. General

These Rules cover open and ducted type propellers situated at the stern of a vessel having controllable pitch or fixed pitch blades. Ice loads on bow propellers and pulling type propellers shall receive special consideration. The given loads are expected, single occurrence, maximum values for the whole ships service life for normal operational conditions. These loads do not cover off-design operational conditions, for example when a stopped propeller is dragged through ice. These Rules apply also for azimuthing (geared and podded) thrusters considering loads due to propeller ice interaction. However, ice loads due to ice impacts on the body of azimuthing thrusters are not covered by Section 3.

The loads given herein are total loads (unless otherwise stated) during ice interaction and are to be applied separately (unless otherwise stated) and are intended for component strength calculations only. The different loads given here are to be applied separately.

2. Propeller blades

2.1. Design ice loads

F_b is a force bending a propeller blade backwards when the propeller mills an ice block while rotating ahead. F_f is a force bending a propeller blade forwards when a propeller interacts with an ice block while rotating ahead.

2.1.1. Ice Class Factors

The Table 3.1 lists the design ice thickness and ice strength index to be used for estimation of the propeller ice loads.

Table 3.1 Definition of ice strength index

Ice Class	H_{ice} [m]	S_{ice} [-]	S_{qice} [-]
PC1	4.0	1.2	1.15
PC2	3.5	1.1	1.15
PC3	3.0	1.1	1.15
PC4	2.5	1.1	1.15
PC5	2.0	1.1	1.15
PC6	1.75	1	1
PC7	1.5	1	1

H_{ice} Ice thickness for machinery strength design

S_{ice} Ice strength index for blade ice force

S_{qice} Ice strength index for blade ice torque

2.1.2 Design Ice Loads for Open Propellers

2.1.2.1 Maximum Backward Blade Force, F_b

When $D < D_{limit}$;

$$F_b = -27\ S_{ice}\left[nD\right]^{0.7}\left[\frac{EAR}{Z}\right]^{0.3}\left[D\right]^2 \quad \text{[kN]} \qquad (1)$$

when $D \geq D_{limit}$;

$$F_b = -23\ S_{ice}\left[nD\right]^{0.7}\left[\frac{EAR}{Z}\right]^{0.3}\left[H_{ice}\right]^{1.4}\left[D\right] \quad \text{[kN]} \qquad (2)$$

Where;

$$D_{limit} = 0.85\ .\ (H_{ice})^{1.4} \quad \text{[m]} \qquad (3)$$

n is the nominal rotational speed (at MCR free running condition) for CP-propeller and 85% of the nominal rotational speed (at MCR free running condition) for a FP-propeller (regardless driving engine type).

F_b is to be applied as a uniform pressure distribution to an area on the back (suction) side of the blade for the following load cases:

a) Load case 1: from 0.6R to the tip and from the blade leading edge to a value of 0.2 chord length.

b) Load case 2: a load equal to 50% of the F_b is to be applied on the propeller tip area outside of 0.9R.

c) Load case 5: for reversible propellers a load equal to 60% of the F_b is to be applied from 0.6R to the tip and from the blade trailing edge to a value of 0.2 chord length.

See load cases 1, 2 and 5 in Table 1 of Appendix.

2.1.2.2 Maximum Forward Blade Force F_f

When D < D_{limit};

$$F_f = 250\left[\frac{EAR}{Z}\right]D^2 \quad [kN] \tag{4}$$

when D ≥ D_{limit};

$$F_f = 500\left[\frac{1}{1-\frac{d}{D}}\right]H_{ice}\left[\frac{EAR}{Z}\right][D] \quad [kN] \tag{5}$$

where;

$$D_{limit} = \left(\frac{2}{1-\frac{d}{D}}\right)H_{ice} \tag{6}$$

F_f is to be applied as a uniform pressure distribution to an area on the face (pressure) side of the blade for the following loads cases:

a) Load case 3: from 0.6R to the tip and from the blade leading edge to a value of 0.2 chord length.

b) Load case 4: a load equal to 50% of the F_f is to be applied on the propeller tip area outside of 0.9R.

c) Load case 5: for reversible propellers a load equal to 60% F_f is to be applied from 0.6R to the tip and from the blade trailing edge to a value of 0.2 chord length.

See load cases 3, 4 and 5 in Table 1 of Appendix.

2.1.3 Design Ice Loads for Ducted Propellers

2.1.3.1 Maximum Backward Blade Force F_b

When D < D_{limit};

$$F_b = -9.5\,S_{ice}\left[\frac{EAR}{Z}\right]^{0.3}[nD]^{0.7}\,D^2 \quad [kN] \tag{7}$$

when $D \geq D_{limit}$;

$$F_b = -66\ S_{ice} \left[\frac{EAR}{Z}\right]^{0.3} [nD]^{0.7}\ D^{0.6}\ [H_{ice}]^{1.4} \quad [kN] \tag{8}$$

where $D_{limit} = 4.H_{ice}$

n shall be taken as in D.2.1.2.1.

F_b is to be applied as a uniform pressure distribution to an area on the back side for the following load cases (see Table 2 of Appendix):

a) Load case 1: On the back of the blade from 0.6R to the tip and from the blade leading edge to a value of 0.2 chord length.

b) Load case 5: For reversible rotation propellers a load equal to 60% of F_b is applied on the blade face from 0.6R to the tip and from the blade trailing edge to a value of 0.2 chord length.

2.1.3.2 Maximum Forward Blade Force F_f

When $D \leq D_{limit}$;

$$F_f = 250 \cdot \left[\frac{EAR}{Z}\right] \cdot D^2 \quad [kN] \tag{9}$$

when $D > D_{limit}$;

$$F_f = 500 \cdot \left[\frac{EAR}{Z}\right] \cdot D \cdot \frac{1}{\left(1 - \frac{d}{D}\right)} \cdot H_{ice} \quad [kN] \tag{10}$$

where;

$$D_{limit} = \frac{2}{\left(1 - \frac{d}{D}\right)} \cdot H_{ice} \quad [m] \tag{11}$$

F_f is to be applied as a uniform pressure distribution to an area on the face (pressure) side for the following load case (see Table 2 Appendix):

a) Load case 3: On the blade face from 0.6R to the tip and from the blade leading edge to a value of 0.5 chord length.

b) Load case 5: A load equal to 60% F_f is to be applied from 0.6R to the tip and from the blade leading edge to a value of 0.2 chord length.

2.1.4 Blade Failure Load for Both Open and Ducted Propeller F_{ex}

The force F_{ex} is acting at 0.8R in the weakest direction of the blade and at a spindle arm of 2/3 of the distance of axis of blade rotation of leading and trailing edge whichever is the greatest.

The blade failure load is:

$$F_{ex} = \frac{0.3 \cdot c \cdot t^2 \cdot \sigma_{ref}}{0.8 \cdot D - 2 \cdot r} 10^3 \quad [kN] \tag{12}$$

$$\sigma_{ref} = 0.6 \cdot \sigma_{0.2} + 0.4 \cdot \sigma_u \quad [MPa] \qquad (13)$$

where σ_u (maximum ultimate tensile strength) and $\sigma_{0.2}$ (maximum yield or 0.2 % proof strength) are representative values for the blade material. Representative in this respect means values for the considered section. These values may either be obtained by means of tests, or commonly accepted "thickness correction factors" approved by **TL**. If not available, maximum specified values shall be used.

c, t and r are respectively the actual chord length, thickness and radius of the cylindrical root section of the blade at the weakest section outside root fillet, and typically will be at the termination of the fillet into the blade profile.

2.1.5 Maximum Propeller Ice Torque applied to both, open and ducted propeller Q_{max}

When $D < D_{limit}$;

$$Q_{max} = 105 \times (1 - d/D) \times S_{qice} \times (P_{0.7} / D)^{0.16} \times (t_{0.7} / D)^{0.6} \times (nD)^{0.17} \times D^3 \quad [kNm] \qquad (14)$$

when $D \geq D_{limit}$;

$$Q_{max} = 202 \times (1 - d/D) \times S_{qice} \times H_{ice}^{1.1} \times (P_{0.7} / D)^{0.16} \times (t_{0.7} / D)^{0.6} \times (nD)^{0.17} \times D^{1.9} \quad [kNm] \qquad (15)$$

where;

$D_{limit} = 1.81 \cdot H_{ice}$ [m]

$P_{0.7}$ = Propeller pitch at 0.7 R [m]

$t_{0.7}$ = Max thickness at 0.7 radius

n is the rotational propeller speed, [rps], at bollard condition. If not known, n is to be taken as follows:

Table 3.2 Rotational speed at bollard condition

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

Where n_n is the nominal rotational speed at MCR, [rps], free running condition.

For CP propellers, propeller pitch, $P_{0.7}$ shall correspond to MCR in bollard condition. If not known, $P_{0.7}$ is to be taken as $0.7 \cdot P_{0.7n}$, where $P_{0.7n}$ is propeller pitch at MCR free running condition.

2.2 Dynamic analysis of blade-fatigue

2.2.1 Number of ice loads N_{ice}

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

$N_{ice} = k_1 \cdot k_2 \cdot N_{class} \cdot n$ (16)

N_{class} = Reference number of impacts per propeller rotation speed for each ice class, according to Table 3.3.

Table 3.3 Reference number of impacts

Ice Class	PC1	PC2	PC3	PC4	PC5	PC6	PC7
N_{class}	21×10^6	17×10^6	15×10^6	13×10^6	11×10^6	9×10^6	6×10^6

k_1 = 1 for centre propeller

= 2 for wing propeller

= 3 for pulling propeller (wing and centre)

k_2 = 0,8 - f when f < 0

= 0,8 – 0,4·f when 0 ≤f ≤ 1

= 0,6 – 0,2·f when 1< f ≤ 2,5

= 0,1 when f > 2,5

where the immersion function f is:

$$f = \frac{h_o - H_{ice}}{D/2} - 1 \tag{17}$$

h_0 = Depth of the propeller centreline at the minimum ballast waterline in ice (LIWL) of the ship [m]

2.2.2 Distribution of ice loads

The ice load spectrum is assumed to be of the Weibull type distribution and has the general form:

$$P\left[\frac{F_{ice}}{(F_{ice})_{maks}} \geq \frac{F}{(F_{ice})_{maks}}\right] = e^{\left[-\left(\frac{F}{(F_{ice})_{maks}}\right)^k \cdot \ln(N_0)\right]} \tag{18}$$

k = Shape parameter of the spectrum

N_0 = Number of load cycles in the spectrum

F_{ice} = Random variable for ice loads on the blade,

$0 \leq F_{ice} \leq (F_{ice})_{max}$

The Weibull distributions with shape parameters k = 0.75 and k = 1 are shown in Fig. 3.1.

It is suggested that the distribution with a shape parameter k = 0.75 is used for open propellers and k = 1 for ducted propellers.

2.2.3 Equivalent fatigue stress σ_{fat}

The equivalent fatigue stress for 100 million stress cycles which produces the same fatigue damage as the load distribution is:

$$\sigma_{fat} = \rho \cdot (\sigma_{ice})_{max} \quad (19)$$

$$(\sigma_{ice})_{max} = 0{,}5 \cdot [(\sigma_{ice})_{fmax} - (\sigma_{ice})_{bmax}] \quad (20)$$

$(\sigma_{ice})_{max}$ = Mean value of the principal stress amplitudes resulting from design forward and backward blade forces at the location being studied. [N/mm^2]

$(\sigma_{ice})_{fmax}$ = Principal stress resulting from forward load F_f (D.2.1.2.2 and D.2.1.3.2) [N/mm^2]

$(\sigma_{ice})_{bmax}$ = Principal stress resulting from backward load F_b (D.2.1.2.1 and D.2.1.3.1) [N/mm^2]

2.2.4 Calculation of ρ-parameter for reduction of ice load spectrum

For calculation of equivalent fatigue stress two types of S-N curves are available.

a) Two slope S-N curve (slopes 4.5 and 10), see Figure 3.2.

b) One slope S-N curve(the slope can be chosen), see Figure 3.3.

The type of the S-N-curve shall be selected to correspond to the material properties of the blade. If the SN- curve is not known the two slope S-N curve shall be used, see Figure 3.2.

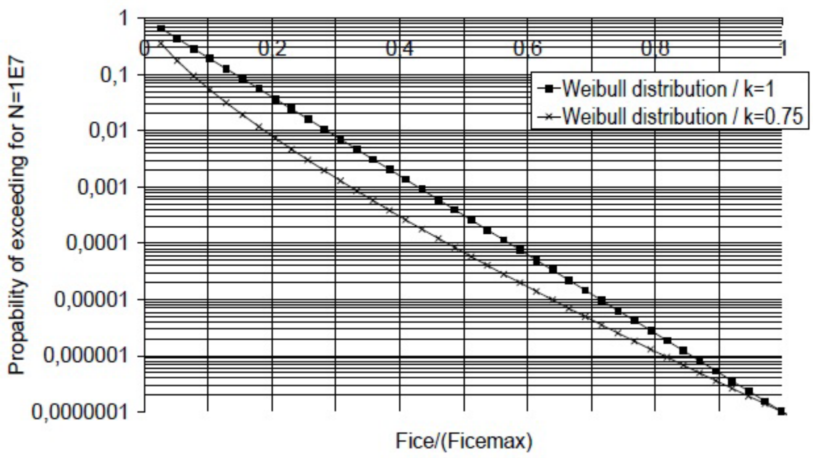


Figure 3.1 – The rainflow distribution of blade bending moment for Gudingen and Weibull distributions with shape parameters 0.75 and 1

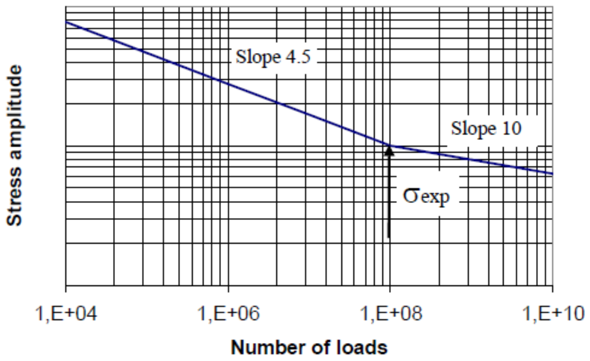


Figure 3.2 – Two Slope S-N Curve

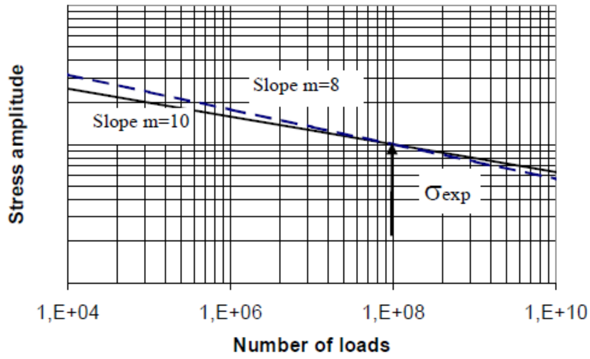


Figure 3.3 – Constant Slope S-N Curve

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

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

$$\rho = C_1 \cdot (\sigma_{ice})_{max}^{c_2} \cdot \sigma_n^{c_3} \cdot \log (N_{ice})^{c_4} \tag{21}$$

$$\sigma_{fl} = \gamma_\varepsilon \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp} \tag{22}$$

γ_ε = Reduction factor for scatter and test specimen size effect,

γ_v = Reduction factor for variable amplitude loading,

γ_m = Reduction factor for mean stress,

σ_{exp} = Mean fatigue strength of the blade material at 10^8 cycles to failure in seawater. The following values should be used for the reduction factors if actual values are not available: γ_ε = 0,67, γ_v = 0,75, and γ_m = 0,75. [N/mm^2],

The coefficients C_1, C_2, C_3 and C_4 are given in Table 3.4.

Table 3.4 Definition of Calculation coefficients

Coefficients	Open propeller	Ducted propeller
C_1	0,000711	0,000509
C_2	0,0645	0,0533
C_3	-0,0565	-0,0459
C_4	2,22	2,584

Table 3.5 Characteristic fatigue strengths for cast propeller materials at zero and 60 MPa mean stress

Material	Experimental fatigue strength [MPa] at 1x10^8 cycles in sea water	σ_{fl} [MPa] at zero mean stress	σ_{fl} [MPa] at 60 Mpa mean stress
Ni Al Bronze	110	55	41
Ni Mn Bronze	80	40	30
High tensile brass	72	36	27
Ferritic stainless steel	50	25	19

b) Calculation of ρ parameter for constant-slope S-N curve

For materials with a constant-slope S-N curve, see Fig 3.3, the ρ factor shall be calculated with the following formula:

$$\rho = \left(G \frac{N_{ice}}{N_R} \right)^{1/m} \cdot \left[\ln \left(N_{ice} \right) \right]^{-1/k} \tag{23}$$

where k is the shape parameter of the Weibull distribution k = 1.0 for ducted propellers and k= 0.75 for open propellers and m is the slope of the S-N curve.

N_R = Reference number of load cycles (10^8)
m = Slope parameter

Values for the G parameter are given in Table 3.6.

Table 3.6 Value for the G parameter for different m/k ratios

m/k	G	m/k	G	m/k	G
3	6	**5,5**	287,9	**8**	40320
3.5	11,6	**6**	720	**8.5**	119292
4	24	**6,5**	1871	**9**	362880
4.5	52,3	**7**	5040	**9.5**	1,133E6
5	120	**7,5**	14034	**10**	3,623E6

2.3 Acceptability of blades

2.3.1 Maximum Blade Stresses σ_{calc}

Blade stresses are to be calculated using the backward and forward loads given in D.2.1.2 and D.2.1.3. The stresses shall be calculated with recognised and well documented FE-analysis or other acceptable alternative method. The stresses on the blade shall not exceed the allowable stresses for the blade material given below.

Calculated blade stress for maximum ice load shall comply with the following:

$\sigma_{calc} < \sigma_{ref} / S$ [MPa] (24)

S = Safety factor

= 1.5

σ_{ref} = Reference stress, defined as:

$\sigma_{ref} = 0.7 \cdot \sigma_u$ [MPa] (25)

or

$\sigma_{ref} = 0.6 \cdot \sigma_{0.2} + 0.4 \cdot \sigma_u$ [MPa] (26)

whichever is less.

Where σ_u and $\sigma_{0.2}$ are representative values for the blade material.

2.3.2 Blade Tip and Edge Thickness $t_{1.0\,PC}$, t_{EPC}

The blade edges and tip have to be designed such that during normal operation, ice contact and ice milling, no essential damage can be expected.

The blade tip thickness has to be greater than $t_{1,0\,PC}$ given by the following formula:

$$t_{1,0\,PC} = \left(t_{1,0\,B} + 2 \cdot D\right)\sqrt{\frac{500}{\sigma_{ref}}} \text{ [mm]} \tag{27}$$

The tip thickness $t_{1.0\,PC}$ has to be measured at a distance x_{th} perpendicular to the contour edge, above 0.975 R. It needs to be demonstrated that the thickness is smoothly interpolated between lower bound leading edge thickness at 0.975 R, tip and lower bound trailing edge at 0.975 R. The basic tip thickness $t_{1.0B}$ has to be chosen according to Table 3.7.

Table 3.7 Basic tip thickness for propeller blades

Ice Class	PC1	PC2	PC3	PC4	PC5	PC6	PC7
$t_{1.0\,B}$ [mm]	30	28	26	24	22	19	16

$x_{th} = \text{Min}(0.025\, c_{0.975\,R}\, ;\, 45)$ [mm] (28)

x_{th} = Distance from the blade edge [mm]

The blade edge thickness t_{EPC} measured at a distance of x_{th} along the cylindrical section at any radius up to 0.975 R has to be not less than 50 % of the tip thickness. This requirement is not applicable to the trailing edge of non reversible propellers.

2.3.3 Acceptability criterion for fatigue

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

$$\frac{\sigma_{fl}}{\sigma_{fat}} \geq 1{,}5 \quad (29)$$

$$\sigma_{fl} = \gamma_\varepsilon \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp} \quad (30)$$

γ_ε = The reduction factor for scatter and test specimen size effect

γ_v = Reduction factor for variable amplitude loading

γ_m = Reduction factor for mean stress

σ_{exp} = Mean fatigue strength of the blade material at 10^8 cycles to failure in sea water. The following values should be used for the reduction factors if actual values are not available: γ_ε = 0.67, γ_v = 0.75 and γ_m = 0.75 [N/mm^2]

2.4 Propeller blade mounting

2.4.1 Loads

Blade flanges and bolts are to be designed to withstand the blade failure force F_{ex} given in D.2.1.4.

Separate means, e.g. dowel pins, have to be provided in order to withstand the maximum spindle torque Q_{smax} (see D.3.1).

2.4.2 Acceptability of bolts and pins

Blade bolts shall have following minimum section modulus (based on minimum diameter of shank or thread core) around bolt pitch circle, or an other relevant axis for non circular joints, parallel to considered root section:

$$W_{bolt} = \frac{S \cdot F_{ex} \cdot \left(0{,}8D - 2r_{bolt}\right)}{2 \cdot \sigma_{0,2}} \cdot 10^6 \ [\mathrm{mm}^3] \quad (31)$$

r_{bolt} = Radius to the bolt plan [m]

S = Safety factor

= 1.5

$\sigma_{0,2}$ = Minimum specified yield strength of bolt material [N/mm^2]

Blade bolt pre-tension shall be sufficient to avoid separation between mating surfaces with maximum forward and backward ice loads as defined in D.2.1.2 and D.2.1.3 (open and ducted propeller respectively). Usually 60 %-70 % of bolt yield strength is sufficient.

A safety factor of S = 1.5 is required to withstand the spindle torque load Q_{smax} (1). The diameter of dowel pins may be calculated by using the following formula:

$$d = 10^3 \sqrt{\frac{S \cdot Q_{smax} \cdot 8 \cdot \sqrt{3}}{PCD \cdot i \cdot \pi \cdot \sigma_{0,2}}} \quad [mm] \qquad (32)$$

i = Number of pins

Q_{smax} = According to formula (35)

PCD = Pitch circle diameter [mm]

S = Safety factor

= 1.5

$\sigma_{0,2}$ = Minimum specified yield strength of bolt material [N/mm^2]

3. Pitching mechanism

3.1 Loads due to Maximum Blade Spindle Torque for open and ducted propellers Q_{smax}

Spindle torque Q_{smax} around the spindle axis of the blade fitting shall be calculated both for the load cases described in D.2.1.3 and D.2.1.4 for F_b and F_f. If these spindle torque values are less than the default value given below, the default minimum value shall be used.

Default Value:

$$Q_{spindle} = 0.25 \cdot F \cdot c_{0.7} \quad [kNm] \qquad (33)$$

where

$c_{0.7}$ = Length of the blade chord at 0.7 R radius [m]

F is either F_b or F_f which ever has the greater absolute value.

$$Q_{sex} = F_{ex} \cdot \frac{2}{3} \cdot L_{ex} \quad [kNm] \qquad (34)$$

Additionally the spindle torque caused by the blade breaking force F_{ex} (D.2.1.4) has to be calculated:

L_{ex} = Maximum of distance from spindle axis to the leading or trailing edge at radius 0.8 R [m]

The maximum spindle torque can be determined by:

$$Q_{smax} = MAX\,(Q_{spindle};\, Q_{sex};\, Q_{sf};\, Q_{sb}) - Q_{fr1} - Q_{fr2} \quad [kNm] \qquad (35)$$

Q_{sf} and Q_{sb} are the spindle torques due to the blade forward and backward acting ice load, F_f and F_b respectively, as given in the load cases D.2.1.2 and D.2.1.3.

Q_{fr1} = friction torque in blade bearings caused by reaction forces due to F_{ex} [kNm]

Q_{fr2} = friction between connected surfaces resulting from blade bolt pretension forces [kNm]

In calculating Q_{fr} a friction coefficient = 0.15 may normally be applied.

3.2 Dynamic loads for fatigue analysis Q_{samax}

Fatigue strength is to be considered for parts transmit- ting the spindle torque from blades to a servo system considering ice spindle torque acting on one blade. The maximum amplitude is defined as:

$$Q_{samax} = \frac{Q_{sb} + Q_{sf}}{2} \quad \text{[kNm]} \tag{36}$$

Q_{sf} and Q_{sb} see D.3.1.

3.3 Acceptability of pitching mechanism

Static calculations have to demonstrate that the components of CP mechanisms are to be designed to withstand the blade failure spindle torque Q_{sex} and maximum spindle torque Q_{smax}.

The maximum spindle torque Q_{smax} shall not lead to any consequential damages.

Provided that calculated stresses duly considering local stress concentrations are less than yield strength, or maximum 70 % of σ_u of respective materials, detailed fatigue analysis is not required. In opposite case components shall be analysed for cumulative fatigue, based on a maximum loading by Q_{samax} (see D.3.2). Similar approach as used for shafting (see E.2.2) may be applied.

3.4 Servo pressure

Minimum design pressure for servo system shall be taken as a pressure caused by Q_{samax} reduced by relevant friction losses in bearings caused by the respective ice loads.

4. Mounting of Propeller

4.1 Keyless cone mounting

The friction capacity shall be at least 2,0 times the highest peak torque Q_{peak} as determined in without exceeding 75 % (bronze) and 80 % (steel) respectively of yield strength of the hub in terms of von Mises stress.

The necessary surface pressure at 0 °C can be determined as:

$$p = \frac{\sqrt{\Theta^2 \cdot T^2 + f \cdot \left(Q^2 + T_r^2\right)} - \Theta \cdot T}{A \cdot f} \quad \text{[MPa]} \tag{37}$$

$$f = \left(\frac{\mu_o}{S}\right)^2 - \Theta^2 \tag{38}$$

Θ = Half conicity of the shaft [-]

T_r = Propeller response thrust [kN]

$Q = Q_{peak}$ according to E.1.1.2 [kNm]

A = Effective contact area of the shrink fit [mm^2]

μ_0 = 0.15 for steel-steel,

= 0.13 for steel-bronze

S = 2.0

The backward response thrust T_r for pushing propellers and the forward response thrust for pulling propellers respectively has to be inserted and a negative sign shall be used.

4.2 Key mounting

Key mounting is not permitted.

4.3 Flange mounting

a) The flange thickness is to be at least 25 % of the shaft diameter.

b) Any additional stress raisers such as recesses for bolt heads shall not interfere with the flange fillet.

c) The flange fillet radius is to be at least 10 % of the shaft diameter.

d) The diameter of ream fitted (light press fit) bolts shall be chosen so that the peak torque Q_{peak} does not cause shear stresses beyond 30 % of the yield strength of the bolts.

e) The bolts are to be designed so that the blade failure load F_{ex} in any direction (forward or backwards) does not cause yielding or flange opening.

E. Shafting

1. Design Loads

1.1 Torque

1.1.1 Torque due to a single blade impact Q(φ)

The propeller ice torque excitation for shaft line dynamic analysis shall be described by 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:

$$Q(\varphi) = C_q * Q_{max} * \sin(\varphi\,(180 / \alpha_i)) \quad [kNm] \quad \text{when } \varphi = 0...\alpha_i \qquad (39)$$

Q_{max} see D.2.1.5.

$$Q(\varphi) = 0 \qquad \text{when } \varphi = \alpha_i ....360 \qquad (40)$$

where C_q and α_i parameters are given in Table 3.8 below.

Table 3.8 – Parameters for torque excitation

Torque excitation	Propeller-ice interaction	C_q	α_i
Case 1	Single ice block	0.5	45
Case 2	Single ice block	0.75	90
Case 3	Single ice block	1.0	135
Case 4	Two ice blocks with 45 degree phase in rotation angle	0.5	45

The total ice torque is obtained by summing the torque of single blades taking into account the phase shift 360deg./z. The number of propeller revolutions n_Q during a milling sequence shall be obtained with the formula:

$$n_Q = 2 \cdot H_{ice} \tag{41}$$

The number of impacts is $z \cdot n_Q$.

See Figure 1 in Appendix.

Milling torque sequence duration is not valid for pulling bow propellers, which are subject to special consideration.

The response torque at any shaft component shall be analysed considering excitation torque at the propeller, the actual engine torque Q_e and mass elastic system.

Q_e = Actual maximum engine torque at considered speed

1.1.2 Response torque in the propulsion system Qr (t) and its maximum Qpeak

1.1.2.1 The maximum torque Q_{peak} may be calculated using one of the following three different approaches (a – c). With increasing simplification of method, the result Q_{peak} becomes higher.

a) Transient torsional vibration analysis

The response torque at any component in the propulsion system shall be analysed considering the above excitation torque at the propeller, the actual engine torque Q_e, and the mass elastic system. See Figure 3.4.

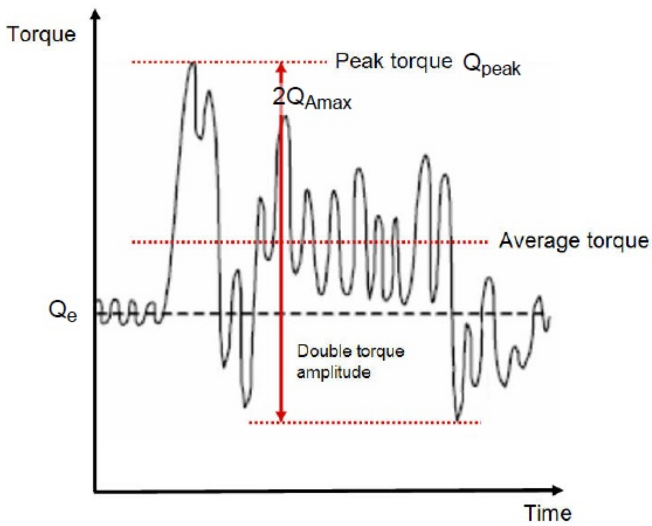


Figure 3.4 – Definition of torque

The response torque $Q_r(t)$ in all components shall be determined by means of transient torsional vibration analysis of the propulsion line. Calculations have to be carried out for all excitation cases given above (E.1.1.1) and the response has to be applied on top of the mean hydrodynamic torque in bollard condition at considered propeller rotational speed.

$$Q_r(t) = Q(\varphi) + Q_e \quad [\text{kNm}] \tag{42}$$

b) Steady state torsional vibration calculation

The response torque Q_r at any component of the propulsion system can be calculated by a steady state torsional vibration calculation (TVC) considering the ice excitation in E.1.1.1 by using excitation factors f_I and f_{II} for the 1st and 2nd propeller order as well as with a factor f_{static} for a static part on the basis of Table 3.9.

Table 3.9 – Parameters for torsional vibration calculation

No.of blades	Excitation case (acc. to Fig.1 of Appendix)	f_I	f_{II}	f_{static}
3	1	0,40	0	0,36
	2	0,32	0,05	0,72
	3	0	0,27	0,24
4	1	0,32	0,06	0,48
	2	0	0,05	0,95
	3	0	0,21	0,32
5	1	0,16	0	0,60
	2	0,17	0	1,20
	3	0	0,10	0,40
6	1	0	0,04	0,72
	2	0,10	0	1,43
	3	0	0	0,48
7	1	0,10	0	0,84
	2	0,15	0,02	1,67
	3	0	0,07	0,56
All values as fraction of Q_{max} (for use in TVC f_I and f_{II} are to be multiplied by the ratio of Q_{max} / Q_{nom})				

Q_{max} = Propeller ice torque according to D.2.1.5 [kNm]

Q_{nom} = Nominal engine torque at maximum continuous rating (MCR) [kNm]

The response torque Q_r for every single speed within the operating range for each ice excitation case (Fig.1 of Appendix) shall be calculated according to:

$$Q_r = Q_e + Q_{static} + Q_{dyn} \ [\text{kNm}] \tag{43}$$

Q_e = Actual mean torque at considered speed [kNm]

$Q_{static} = f_{static} \,.\, Q_{max}$ [kNm]
= static part of ice excitation torque according to Table 3.9

Q_{dyn} = result of TVC for alternating torque by using propeller excitation factors f_I and f_{II} acc. to Table 3.9 [kNm]

The highest peak torque Q_{peak} is equal to the highest response torque Q_r calculated over all excitation cases (see E.1.1.1) within the operating range.

c) Simple formula

If it can be demonstrated that no resonance of first or second blade order against any elastic element such as main and generator couplings (PTO) occur over the whole operating speed range, the highest peak torque in the propeller shaft Q_{peak} can be calculated by using the following formula:

$$Q_{peak} = 1.4\,(Q_{max} + Q_{nom})\ [\text{kNm}] \qquad (44)$$

or alternatively, if Q_{max} cannot be determined due to lack of propeller geometry information:

$$Q_{peak} = 3.Q_{nom}\ [\text{kNm}] \qquad (45)$$

For gear equipped propulsion systems the maximum torque for components on input side of the gear Q_{peak} in shall be calculated with the following formula based on the maximum torque Q_{peak} according to equations (44) respectively (45):

$$Q_{peak\,in} = \frac{1{,}3 \cdot I_H}{I_H \cdot \frac{1}{u^2} + I_I} \cdot Q_{peak} \quad [\text{kNm}] \qquad (46)$$

I_H = Moment of inertia for masses with engine speed [kgm^2]

I_I = Moment of inertia for masses with propeller speed [kgm^2]

u = Reduction ratio (engine speed / propeller speed) [-]

For all components on output side of the gear Q_{peak} applies.

1.1.2.2 The results of the three excitation cases are to be used in the following way for Q_{peak} and Q_{Amax}:

– The highest peak torque (between the various lumped masses in the system) is in the following referred to as peak torque Q_{peak}.

– The highest torque amplitude during a sequence of impacts is to be determined as half of the range from maximum to minimum torque and is referred to as Q_{Amax} (see Fig. 3.4).

1.2 Thrust

1.2.1 Maximum Propeller Ice Thrust applied to the shaft

$$T_f = 1.1\,.\,F_f \quad [\text{kN}] \qquad (47)$$

$T_b = 1.1 . F_b$ [kN] (48)

For F_f, F_b see D.2.1.2 and D.2.1.3.

1.2.2 Maximum Response Thrust T_r

Maximum thrust along the propeller shaft line is to be calculated with the formulae below. The factors 2.2 and 1.5 take into account the dynamic magnification due to axial vibration. Alternatively the propeller thrust magnification factor may be calculated by dynamic analysis.

Maximum Shaft Thrust Forwards: $T_r = T_n + 2.2 \times T_f$ [kN] (49)

Maximum Shaft Thrust Backwards: $T_r = 1.5 \times T_b$ [kN] (50)

T_n = Propeller bollard thrust [kN]
T_f = Maximum forward propeller ice thrust [kN]

T_b = Maximum backward propeller ice thrust [kN]

If hydrodynamic bollard thrust T_n is not known, T_n is to be taken as follows:

Table 3.10 – Propeller bollard thrust

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

T = Nominal propeller thrust at MCR at free running open water conditions

2. Dimensioning and acceptability

2.1 Propeller shaft

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

The blade failure load F_{ex} (see D.2.1.4) applied on the propeller blade at 0.8 R radius parallel to the shaft (forward or backwards) shall not cause yielding. The bending moment need not to be combined with any other load.

This requires a minimum diameter d_p in way of the aft stern tube bearing of:

$$d_p = 160 \cdot \sqrt[3]{\frac{F_{ex} \cdot D}{\sigma_y}} \quad \text{[mm]} \qquad (51)$$

σ_y = Minimum specified yield or 0.2 % proof strength of the propeller shaft material [MPa]

In front of the aft stern tube bearing the diameter may be reduced based on the assumption that the bending moment is linearly reduced to 20 % 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 (D.2.1.2 and D.2.1.3) has been disregarded, because the resulting stress levels are much below the stresses due to the blade failure load F_{ex}.

2.2. Propeller and intermediate shafts

2.2.1. The stresses due to the peak torque Q_{peak} (E.1.1.2) shall have a minimum safety factor of 1.25 against yielding in plain sections and 1.0 in way of stress concentrations in order to avoid bent shafts.

The minimum diameter is:

Plain shaft:

$$d_p = 225 \cdot \sqrt[3]{\frac{Q_{peak}}{\sigma_y}} \quad [\text{mm}] \tag{52}$$

Notched shaft:

$$d_p = 210 \cdot \sqrt[3]{\frac{Q_{peak} \cdot \alpha_t}{\sigma_y}} \quad [\text{mm}] \tag{53}$$

where α_t is the local stress concentration factor in torsion.

2.2.2 The torque amplitudes, based on Q_{peak}, with the foreseen number of cycles, as defined in F.1.2.3, shall be used in an accumulated fatigue evaluation where the safety factor is 1.5 compared with the 50 % survival probability curve. If the plant also has high engine excited torsional vibrations (e.g. direct coupled 2-stroke engines), this has also to be considered.

2.2.3 For plants with reversing direction of rotation the stress range $\Delta\tau \cdot \alpha_t$ resulting from forward $Q_{peakforward}$ to astern $Q_{peakastern}$ shall not exceed twice the yield strength (in order to avoid stress-strain hysteresis loop) with a safety factor of 1.25, i.e.:

$$\Delta\tau \cdot \alpha_t \le \frac{2 \cdot \sigma_y}{\sqrt{3} \cdot 1{,}25} \quad [\text{MPa}] \tag{54}$$

where α_t is the local stress concentration factor in torsion.

2.3 Shaft connections

2.3.1 Shrink fit couplings (keyless)

The friction capacity shall be at least 1.8 times the highest peak torque Q_{peak} as determined in E.1.1.2 without exceeding 80 % of yield strength (steel).

The necessary surface pressure can be determined according to equation (37).

2.3.2 Key mounting

Key mounting is not permitted.

2.3.3 Flange mounting

The following requirements have to be considered:

a) Any additional stress raisers such as recesses for bolt heads shall not interfere with the flange fillet

b) The diameter of ream fitted (light press fit) bolts or pins shall be chosen so that the peak torque Q_{peak} (see E.1.1.2) does not cause shear stresses be- yond 30 % of the yield strength of the bolts or pins.

c) The bolts are to be designed so that the blade failure load F_{ex} (see D.2.1.4) in backward direction does not cause yielding or flange mating surface separation. Depending on flange position, a reduction of bending load according to E.2.1 is permitted.

2.4 Bearings

2.4.1 General

All shaft bearings are to be designed to withstand the propeller blade ice interaction loads according to D.2. For the purpose of calculation, the shafts are assumed to rotate at rated speed. Reaction forces due to the response torque Q_r (φ) (e.g. in gear transmissions) are to be considered. Additionally the aft stern tube bearing as well as the next shaft line bearings are to withstand a bending moment caused by F_{ex} as given in D.2.1.4, in such a way that the ship can maintain operational capability. The pressures in these bearings are to be assessed based on the bending moment distribution given in E.2.1. For low operational propeller speeds (e.g. for drives with electric motors or for drives with several motors/ propellers and some of the propellers part-time in "wind milling") suitable measures for maintaining bearing lubrication (e.g. additional hydrostatic lubrication) are to be provided.

2.4.2 Thrust bearings

Thrust bearings and their housings are to be designed to withstand maximum response thrust T_r according to E.1.2.2 and the force resulting from the blade failure load F_{ex} in D.2.1.4. For the purpose of calculation, except for F_{ex}, the shafts are assumed to rotate at rated speed.

2.4.3 Roller bearings

Roller bearings are to have a L_{10h} lifetime of at least 40000 hours. The calculation of lifetime is to be based on reaction forces from the torque spectrum and principles given in F.1.2.3.

2.5 Seals

Seals are to be provided to prevent egress of pollutants and shall be suitable for the operating temperatures. Contingency plans for preventing the egress of pollutants under failure conditions are to be documented.

Seals are to be of proven design.

F. Gears, Flexible Couplings, Clutches

1. Gear transmissions

1.1 Calculation of maximum torque $Q_{peak\ g}$

For gear equipped propulsion systems it has to be demonstrated that the gear transmission withstands loads based on the maximum torque $Q_{peak\ g}$. The maximum torque can be calculated with the following formula:

$$Q_{peak\ g} = \frac{1{,}3 \cdot I_H \cdot u^2}{I_H \cdot u^2 + I_L} \cdot Q_{peak} \quad [kNm] \tag{55}$$

$Q_{peak\ g}$ = Maximum torque in gear mesh side [kNm]

I_H = Moment of inertia for masses with higher speed [kgm^2]

I_L = Moment of inertia for masses with lower speed [kgm^2]

u = Reduction ratio (input speed / output speed)

Q_{peak} = see E.1.1.2

1.2 Calculation of the load-bearing capacity of cylindrical and bevel gearing

1.2.1 General

The sufficient load capacity of the gear-tooth system is to be demonstrated by load capacity calculations while maintaining the required safety margins for the criteria stated below. Cylindrical gears can be assessed on the basis of the international standard ISO 6336 Pt. 1–6, provided that "methods B" are used. Other calculation methods may be accepted provided that they are reasonably equivalent.

It is recommended to assess bevel gears by equivalent methods. The use of ISO 10300 is only accepted within the given limitations of the ratio face width/module.

1.2.2 Load distribution factors

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 twin input single output gears).

1.2.3 Contact stress

The safety against pitting shall be assessed against the given load spectrum as well as the ordinary loads (open water running) by means of accumulated fatigue analyses (stated in ISO 6336 Pt 6) with a minimum resulting safety factor S_H of 1.2 (ref. ISO 6336 Pt 1, 2 and 6).

The ice load spectrum for the output gear is defined as:

100 % of $Q_{peak\ g}$ with n_Q cycles

80 % of $Q_{peak\ g}$ with $(z\ N_{ice})^{0.2}\ n_Q$ cycles

60 % of $Q_{peak\ g}$ with $(z\ N_{ice})^{0.4}\ n_Q$ cycles

40 % of $Q_{peak\ g}$ with $(z\ N_{ice})^{0.6}\ n_Q$ cycles

20 % of $Q_{peak\ g}$ with $(z\ N_{ice})^{0.8}\ n_Q$ cycles ($n_Q = 2\ H_{ice}$)

n_Q = see formula (41)

For pinions and wheels with higher speed the numbers of load cycles n_Q are found by multiplication with the gear ratios.

1.2.4 Tooth root stress

Tooth root safety shall be assessed in the same way as for contact stress but with a minimum safety factor S_F of 1.5 (ref. ISO 6336 Pt 1, 3 and 6).

1.2.5 Scuffing

The scuffing safety (flash temperature method according to DIN 3990, Part 4) based on the peak torque $Q_{peak\ g}$ shall be at least 1.2 when the FZG class of the oil is assumed one stage below specification.

1.2.6 Flank subsurface fatigue

Sub-surface fatigue is mainly influenced by material microstructure, surface hardness and hardness depth. Up to now there is no standardized calculation procedure available. Therefore a careful review of each parameter concerning subsurface fatigue is necessary. For case carburized gears the case depth should be within the recommended range stated in ISO 6336-5 clause 5.6.2/c. It should be noted that high overloads can initiate subsurface fatigue cracks that may lead to a premature failure. In lieu of reliable analyses UT inspection intervals may be used.

1.3 Shafts

Shafts in gear transmissions shall meet the same safety level as intermediate shafts, but where relevant, bending stresses and torsional stresses shall be combined (e.g. by von Mises).

1.4 Bearings

See E.2.4.

2. Flexible couplings

Couplings shall be designed such that frequent occurrence of peak torques Q_{peak} (E.1.1.2) will not lead to fatigue cracking, i.e. exceeding the permissible vibratory torques T_{Kmax1} or ΔT_{max} of the coupling. The permissible torque may be determined by interpolation in a log-log torque-cycle diagram where T_{Kmax1} respectively ΔT_{Kmax} refers to 50000 cycles, see illustration in Figure 3.5 and 3.6.

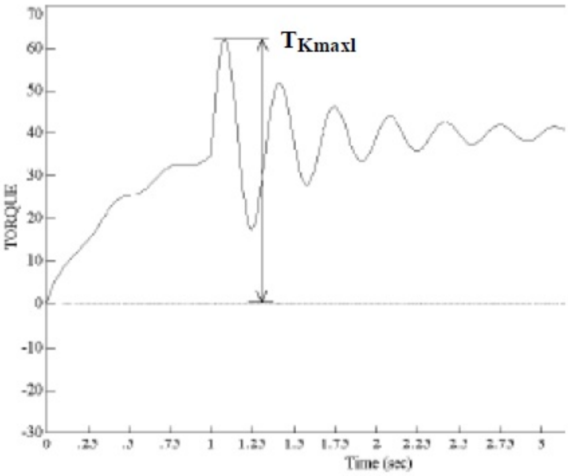


Figure 3.5 – Definition of torque amplitude T_{Kmax1}

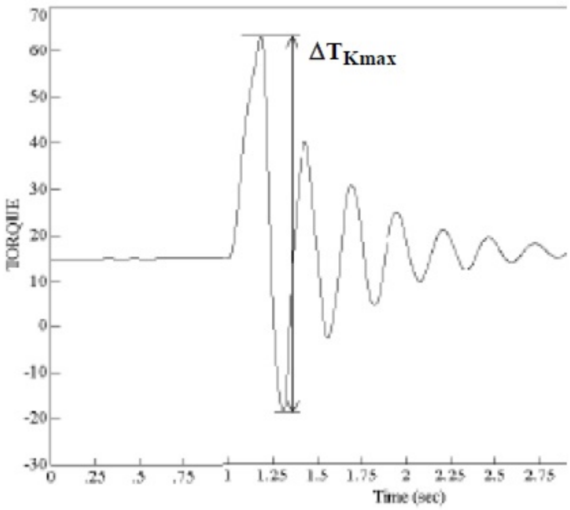


Figure 3.6 - Definition of torque amplitude ΔT_{Kmax}

There shall be a separation margin of at least 20 % between the peak torque Q_{peak} and the torque where any twist limitation is reached.

3. Clutches

Clutches shall have a static friction torque of at least 1.3 times the peak torque Q_{peak} and a dynamic friction torque of 2/3 of the static one.

Emergency operation of the clutch after failure, e.g. loss of operating pressure, shall be established within a reasonably short time. If this is arranged by bolts, they shall be situated on the engine side of the clutch in order to ensure access to all bolts by turning the engine.

G. Azimuth Propulsors

1. General

In addition to the above requirements special consideration shall be given to the loading cases which are extraordinary for propulsion units when compared with conventional propellers. Estimation of the loading cases must reflect the operational realities of the ship and the thrusters. In this respect, for example, the loads caused by impacts of ice blocks on the propeller hub of a pulling propeller must be considered. Also loads due to thrusters operating in an oblique angle to the flow must be considered.

2. Design Ice Loads

Azimuth propulsors shall be designed for the following loads. As far as appropriate, the loads have to be applied simultaneously.

2.1 Ice pressure on strut based on defined location area of the strut / ice interaction as per Section 2, O.2.

2.2 Ice pressure on pod based on defined location area of thruster body / ice interaction as per Section 2, O.2.

2.3 Plastic bending of one propeller blade F_{ex} (see D.2.1.4) in the worst position (typically top-down) or maximum response thrust T_r (see E.1.2.2).

2.4 Steering gear design torque Q_{SG} shall be at least 60 % of steering torque expected at propeller ice milling condition defined as Q_{max} (see D.2.1.5):

$$Q_{SG} = 0{,}6 \frac{Q_{max}}{0{,}8 \cdot R} \cdot \ell \qquad [\mathrm{kNm}] \qquad (56)$$

ℓ = distance from the propeller plane to steering (azimuth) axis [m]

2.5 Steering gear shall be protected by effective means limiting excessive torque caused by:

a) Ice milling torque exceeding design torque and leading to rotation of unit.

b) Torque caused by plastic bending of one propeller blade in the worse position (related to steering gear) and leading to rotation of the unit.

Steering gear shall be ready for operation after above loads a) or b) have disappeared.

3. Acceptability of azimuth thrusters

It has to be demonstrated that the individual components can withstand the loads given in G.2 and in its respective section with a safety factor as required for the individual component.

The housing has to have a safety of 1.0 with respect to yield.

For bolted connections the same safety factor as for the housing itself has to be demonstrated. However, opening of the mating surfaces is not permitted.

Slewing bearings of roller type are to have a L_{10h} lifetime of at least 40000 hours. The calculation of lifetime is to be based on reaction forces from the thrust spectrum given in G.2.

A safety of S = 2.5 against static loads given in G.2 has to be demonstrated.

H. Prime Movers

1. Propulsion Engines

1.1 General

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 in case with the CP system in full pitch as limited by mechanical stoppers.

1.2 Crankshafts

Special considerations apply for plants with large inertia (e.g. flywheel, tuning wheel or PTO) in the front of the engine (opposite to main power take off).

2. Emergency Power Units

Provisions shall be made for heating arrangements to ensure ready starting of the cold emergency power units at an ambient temperature applicable to the Polar class of the ship.

Emergency power units shall be equipped with starting devices with a stored energy capability of at least three consecutive starts at the above mentioned temperature. The source of stored energy shall be protected to preclude critical depletion by the automatic starting system, unless a second independent means of starting is provided. A second source of energy shall be provided for an additional three starts within 30 min., unless manual starting can be demonstrated to be effective.

I. Auxiliary Systems

1. General

In addition to the requirements for ice class ICE-B4 (see Chapter 4 – Machinery, Section 16) the following shall be observed.

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

1.2 Suitable material for low temperatures shall be used for the pipes, valves and fittings which are exposed to sea water or cold air.

1.3 Vent pipes, intake and discharge pipes and associated systems shall be designed to prevent blockage due to freezing or ice and snow accumulation.

1.4 Means should be provided to prevent damage due to freezing, to tanks containing liquids.

1.5 Systems subject to freezing shall be drainable.

1.6 Additional heating of lube oil may be needed for equipment located in machinery spaces.

1.7 Transverse thrusters (not used for propulsion) shall be designed to avoid self destruction in case propeller is blocked by ice.

2. Sea Inlets and cooling water systems

2.1 Cooling water systems for machinery that are essential for the propulsion and safety of the vessel, including sea chests inlets, shall be designed for the environmental conditions applicable to the ice class.

2.2 At least two sea chests are to be arranged as ice boxes for class PC1 to PC5 inclusive where. The calculated volume for each of the ice boxes shall be at least $1m^3$ for every 750 kW of the total installed power. For PC6 and PC7 there shall be at least one ice box located preferably near centre line.

2.3 Ice boxes are to be designed for an effective separation of ice and venting of air.

2.4 Sea inlet valves are to be secured directly to the ice boxes. The valve shall be a full bore type.

2.5 Ice boxes and sea bays are to have vent pipes and are to have shut off valves connected direct to the shell.

2.6 Means are to be provided to prevent freezing of sea bays, ice boxes, ship side valves and fittings above the load waterline.

2.7 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.

2.8 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.

2.9 Openings in ship sides for ice boxes are to be fitted with gratings, or holes or slots in shell plates. The net area through these openings is to be not less than 5 times the area of the inlet pipe. The diameter of holes and width of slot in shell plating is to be not less than 20 mm. Gratings of the ice boxes are to be provided with a means of clearing. Clearing pipes are to be provided with screw-down type non return valves.

3. Ballast and other tanks

3.1 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.

3.2 Fresh water, ballast, fuel & lube oil tanks shall be carefully located and fitted with heating facilities.

3.3 Heating facilities may be needed also for further tanks (e.g. tanks for sludge, leakage, bilge water, sewage, etc.), pending on location and media.

4. Ventilation System

4.1 The air intakes for machinery and accommodation ventilation are to be located on both sides of the ship.

The air intakes are to be sufficient for safe operation of the ship in heavy weather respectively in ice storm conditions.

4.2 Accommodation and ventilation air intakes shall be provided with means of heating.

4.3 The temperature of inlet air provided to machinery from the air intakes shall be suitable for the safe operation of the machinery. Direct ducting to the engines with own heating facilities shall be considered.

5. Steering systems

5.1 Rudder stops are to be provided and integrated into the hull. The design ice force on rudder shall be transmitted to the rudder stops without damage to the steering system.

Ice horn shall in general be fitted to protect the rudder in centre position. Design shall be performed according to Section 2, O.

5.2 The effective holding torque of the rudder actuator, at safety valve set pressure, is obtained by multiplying the open water requirement at design speed (maximum18 knots) by the factors defined in Table 3.11, but not less than the working torque according to Chapter 4 – Machinery, Section 9, A.4.1.2.

Table 3.11 – Factor for holding torque of rudder actuator

Ice class	PC1	PC2	PC3	PC4	PC5	PC6	PC7
Factor	5	5	3	3	3	2	1.5

The design pressure for calculating the scantlings of piping and other steering gear components subjected to internal hydraulic pressure shall be at least 1.25 times the set pressure of the safety valves, but not less than the design pressure according to Chapter 4 – Machinery, Section 9, A.4.1.2.

5.3 It is considered for a Polar Class ship to be able to move her rudder somewhat faster than a seagoing ship operating in open water. So the requirements according to Chapter 4 – Machinery, Section 9, A.3.2 shall be extended to a turning speed according to Table 3.12.

The minimum discharge capacity of the relief valve(s) as mentioned under I.5.2 shall be determined by the turning speed of the rudder actuator according to Table 3.12.

Table 3.12 – Turning speeds for rudder actuator

Ice class	PC1-2	PC3-5	PC6-7
Turning speeds [deg/s]	8	6	4

5.4 The minimum discharge capacity of the additional relief valve(s) as mentioned under I.5.3 shall be determined by the turning speed for the rudder actuator according to Table 3.13.

Table 3.13 - Turning speeds of the rudder actuator for rudders pushed rapidly hard over

Ice class	PC1-2	PC3-5	PC6-7
Turning speeds [deg/s]	40	20	10

The fast acting relieve system shall not allow to cause more than 50 % increase of torque above the set pressure of relief valves according to I.5.2 due to a too slowly acting torque release system. In some cases, a fast acting relief valve with typically 10 milliseconds response time, or a bursting disc, will be needed.

Furthermore, if the specified angular velocity results in an increase in torque of greater than 50 % due to constriction of hydraulic flow, means shall be provided to allow for an improved flow. In some cases a dump tank for the hydraulic fluid may be required.

Following any event, the system capability shall be regained quickly.

J. Foundation of Equipment

1. General

Essential equipment and main propulsion machinery supports shall be suitable for the accelerations as indicated in as follows. Accelerations are to be considered acting independently.

2. Longitudinal Impact Accelerations, a_1

Maximum longitudinal impact acceleration at any point along the hull girder:

$$a_1 = \left(F_{IB}/\Delta\right)\left\{\left[1.1\ \tan\left(\gamma + \Phi\right)\right] + \left[7\frac{H}{L}\right]\right\} \quad [\mathrm{m/s^2}] \qquad (57)$$

F_{IB} = Vertical impact force, defined in Section 2, M.2

Δ = Displacement of the ship [kt]

γ = Bow stem angle at waterline [deg.]

Φ = Maximum friction angle between steel and ice, normally taken as 10° [deg.]

H = Distance in meters from the waterline to the point being considered [m]

L = Length between perpendiculars [m]

3. Vertical Impact Acceleration, a_v

Combined vertical impact acceleration at any point along the hull girder:

$$a_v = 2.5\ (F_{IB}/\Delta)\ F_x \quad [\mathrm{m/s^2}] \qquad (58)$$

F_x = 1.3 at FP
= 0.2 at midships
= 0.4 at AP
= 1.3 at AP for vessels conducting ice breaking astern

Intermediate values to be interpolated linearly.

4. Transverse Impact Acceleration, a_t

Combined transverse impact acceleration at any point along hull girder:

$$a_t = 3\,F_i\,\frac{F_x}{\Delta} \quad [m/s^2] \qquad (59)$$

F_i = Total force normal to shell plating in the bow area due to oblique ice impact, defined in Section 2, C.2.1.3

F_x = 1.5 at FP

= 0.25 at midships

= 0.5 at AP

= 1.5 at AP for vessels conducting ice breaking astern

Intermediate values to be interpolated linearly.

K. Alternative Design

As an alternative, a comprehensive design study may be submitted and may be requested to be validated by an agreed test programme.

APPENDIX

Table 1 - Load cases for open propeller

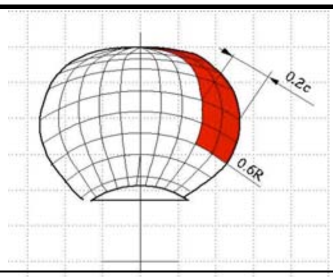
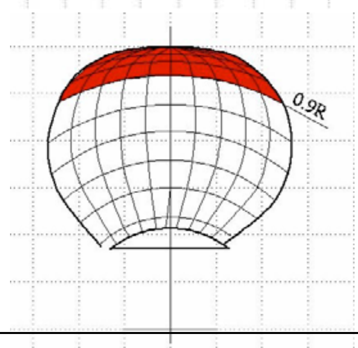
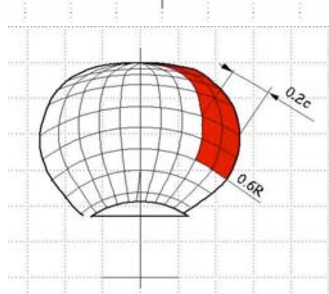
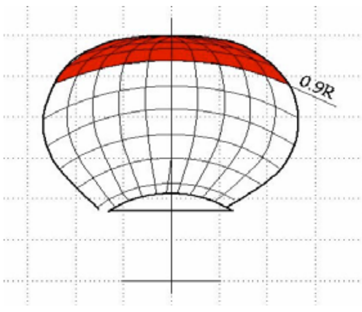
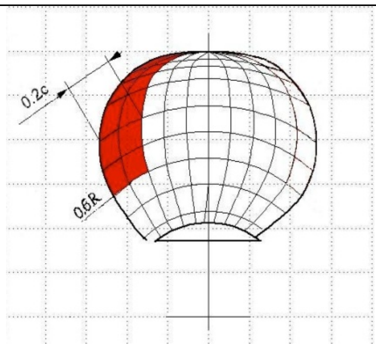
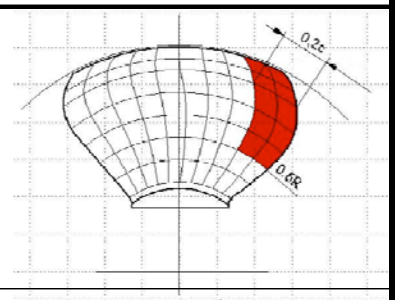
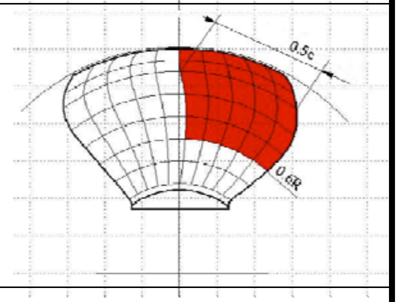
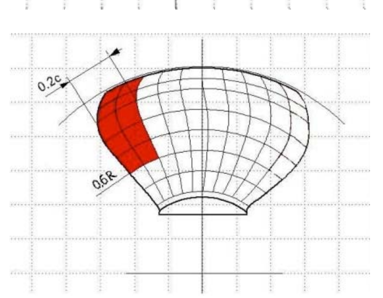
	Force	Loaded area	Right handed propeller blade seen from back
Load case 1	F_b	Uniform pressure applied on the back of the blade (suction side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length.	
Load case 2	50 % of F_b	Uniform pressure applied on the back of the blade (suction side) on the propeller tip area outside of 0.9R radius.	
Load case 3	F_f	Uniform pressure applied on the blade face (pressure side) to an area from 0.6R to the tip and from the leading edge to 0.2 times the chord length.	
Load case 4	50 % of F_f	Uniform pressure applied on propeller face (pressure side) on the propeller tip area outside of 0.9R radius.	
Load case 5	60 % of F_f or F_b which one 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.	

Table 2 - Load cases for ducted propeller

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

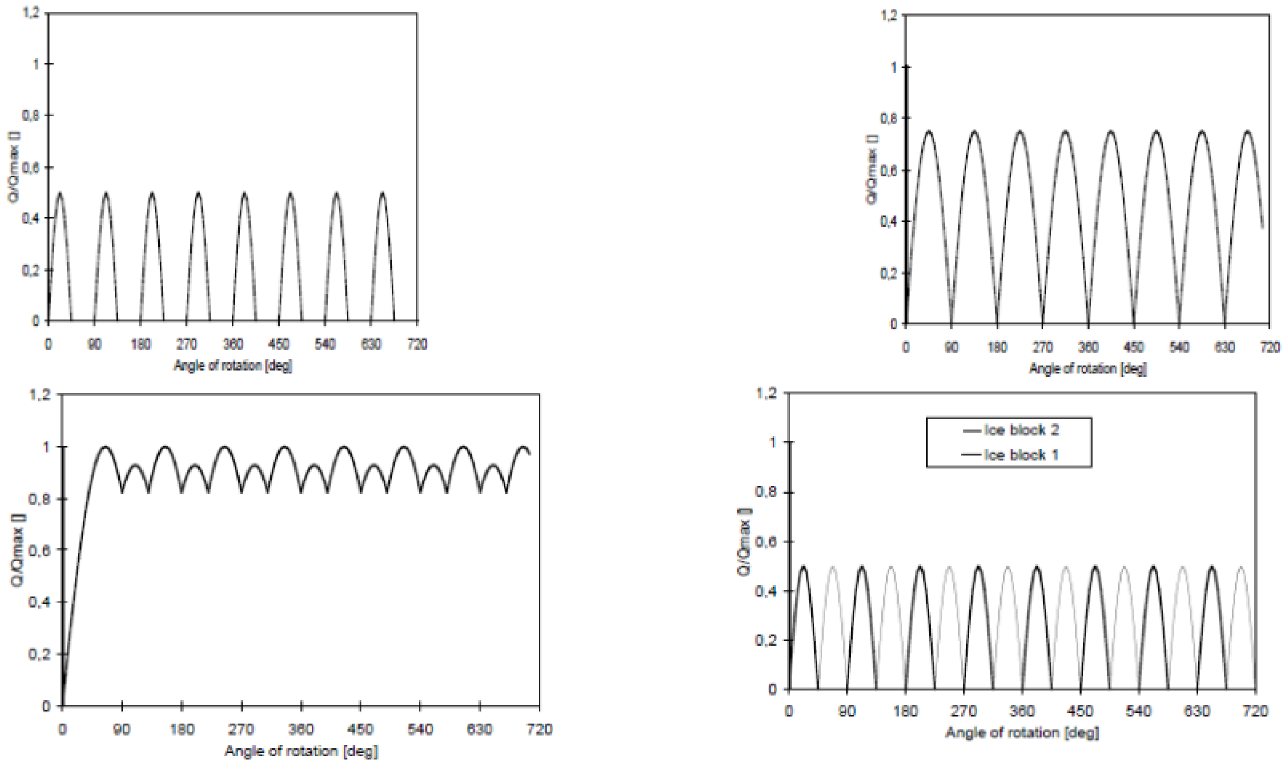


Figure 1 - The shape of the propeller ice torque excitation for 45, 90, 135 degrees single blade impact sequences and 45 degrees double blade impact sequence (two ice pieces) on a four bladed propeller