Requirements concerning

POLAR CLASS

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I1 Polar Class Descriptions and Application

(August 2006) (Rev.1 Jan 2007) (Corr.1 Oct 2007) (Rev.2 Apr 2016)

I1.1 Application

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

I1.1.2 Ships that comply with the UR I2 and UR I3 can be considered for a Polar Class notation as listed in Table 1. The requirements of UR I2 and UR I3 are in addition to the open water requirements of the Classification Society. If the hull and machinery are constructed such as to comply with the requirements of different Polar Classes, then both the hull and machinery are to be assigned the lower of these classes in the Certificate of Classification. Compliance of the hull or machinery with the requirements of a higher Polar Class is also to be indicated in the Certificate of Classification or equivalent.

I1.1.3 Ships which are assigned a Polar Class notation and complying with the relevant requirements of UR I2 and UR I3 may be given the additional notation “Icebreaker”. “Icebreaker" refers to any ship having an operational profile that includes escort or ice management functions, having powering and dimensions that allow it to undertake aggressive operations in ice-covered waters.

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

I1.1.5 For ships which are assigned a Polar Class notation PC 1 through PC 5, bows with vertical sides, and bulbous bows are generally to be avoided. Bow angles should in general be within the range specified in I2.3.1 (v).

Note:

1. UR I1 applies to ships contracted for construction on or after 1 July 2007.

2. Rev.1 of this UR is to be uniformly applied by IACS Societies on ships contracted for construction on and after 1 March 2008.

3. Rev.2 of this UR is to be uniformly implemented by IACS Societies on ships contracted for construction on and after 1 July 2017.

4. The “contracted for construction” date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. For further details regarding the date of “contract for construction”, refer to IACS Procedural Requirement (PR) No. 29.

I1 (cont)

I1.1.6 For ships which are assigned a Polar Class notation PC 6 and PC 7, and are designed with a bow with vertical sides or bulbous bows, operational limitations (restricted from intentional ramming) in design conditions are to be stated in the Certificate of Classification or equivalent.

I1.2 Polar Classes

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

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

Table 1 - Polar Class descriptions

Polar Class	Ice descriptions (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

I1.3 Upper and Lower Ice Waterlines

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

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

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I2 Structural Requirements for Polar Class Ships

(August 2006) (Rev.1 Jan 2007) (Corr.1 Oct 2007) (Rev.2 Nov 2010) (Rev.3 Apr 2016) (Rev.4 Dec 2019)

I2.1 General

I2.1.1 Application*

This UR applies to Polar Class ships according to UR I1.

I2.1.2 Definitions

I2.1.2.1 The length L_{UI} is the distance, in m, measured horizontally from the fore side of the stem at the intersection with the upper ice waterline (UIWL) to the after side of the rudder post, or the centre of the rudder stock if there is no rudder post. L_{UI} is not to be less than 96%, and need not be greater than 97%, of the extreme length of the upper ice waterline (UIWL) measured horizontally from the fore side of the stem. In ships with unusual stern and bow arrangement the length L_{UI} will be specially considered.

I2.1.2.2 The ship displacement D_{UI} is the displacement, in kt, of the ship corresponding to the upper ice waterline (UIWL). Where multiple waterlines are used for determining the UIWL, the displacement is to be determined from the waterline corresponding to the greatest displacement.

I2.2 Hull areas

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

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* Note:

1. UR I2 applies to ships contracted for construction on or after 1 July 2007.

2. Rev.1 of this UR is to be uniformly applied by IACS Societies on ships contracted for construction on and after 1 March 2008.

3. Rev.2 of this UR is to be uniformly implemented by the IACS Societies on ships contracted for construction on or after 1 January 2012.

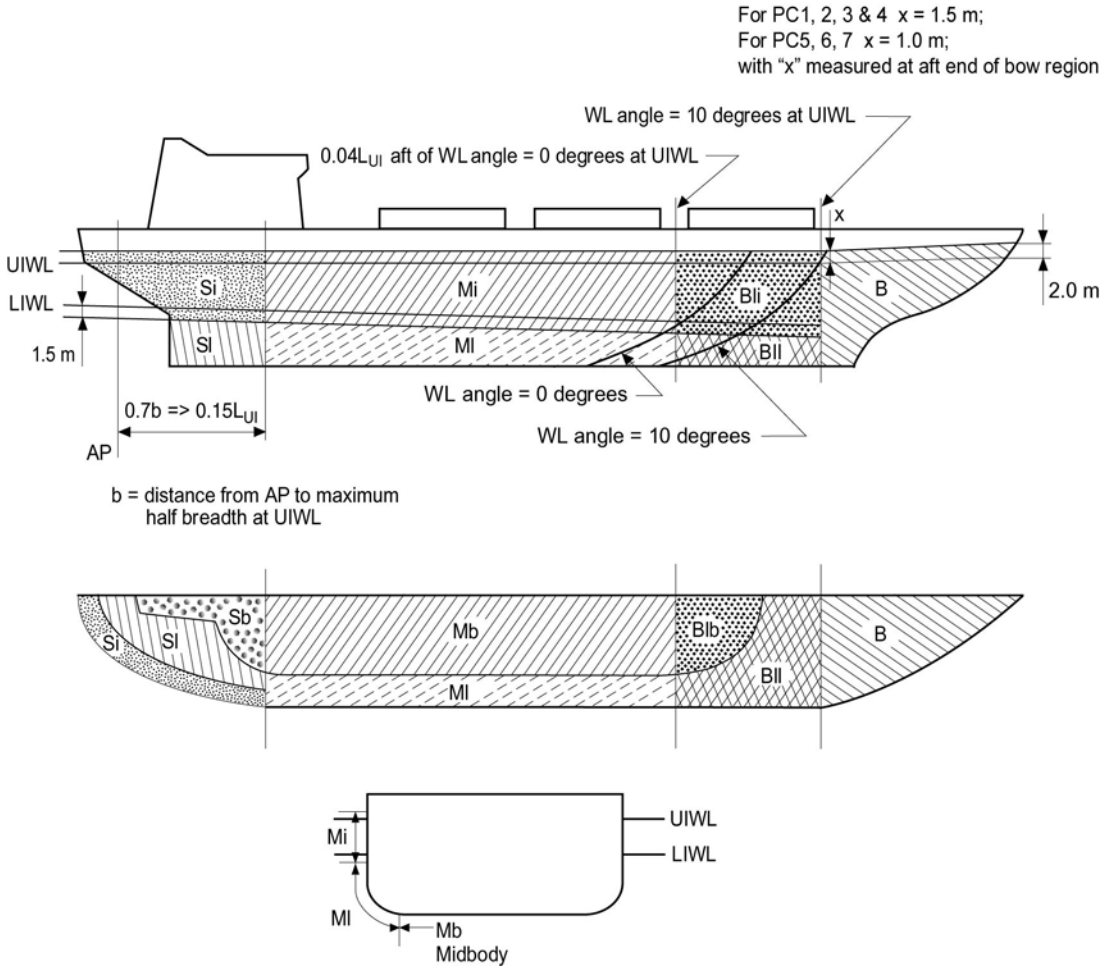
4. Rev.3 of this UR is to be uniformly implemented by the IACS Societies on ships contracted for construction on or after 1 July 2017.

5. The “contracted for construction” date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. For further details regarding the date of “contract for construction”, refer to IACS Procedural Requirement (PR) No. 29.

6. Rev.4 of this UR is to be uniformly implemented by the IACS Societies on ships contracted for construction on or after 1 Jan 2021.

I2
(cont)

Figure 1 - Hull area extents



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

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

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

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

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

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

I2 (cont)

I2.3 Design ice loads

I2.3.1 General

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

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

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

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

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

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

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

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

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

I2.3.2 Glancing impact load characteristics

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

I2
(cont)

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

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

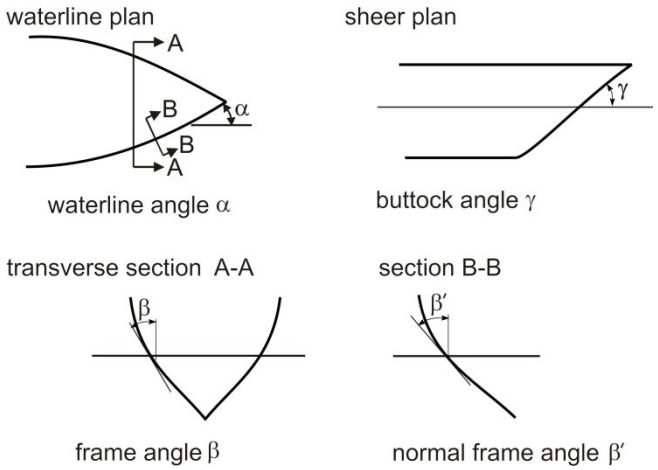
Table 2 - Class factors to be used in I2.3.2.1 (iv)

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

I2.3.2.1 Bow area

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

Figure 2 - Definition of hull angles



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

I2 (cont)

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

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

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

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

where $fa_{i,1} = (0.097 - 0.68 \cdot (x/ L_{UI} - 0.15)^2) \cdot \alpha_i / (\beta'_i)^{0.5}$
$fa_{i,2} = 1.2 \cdot CF_F / (\sin (\beta'_i) \cdot CF_C \cdot D_{UI}^{0.64})$
$fa_{i,3} = 0.60$

(b) Force, F_i:

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

(c) Load patch aspect ratio, AR_i:

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

(d) Line load, Q_i:

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

(e) Pressure, P_i:

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

where i = sub-region considered
L_{UI} = length as defined in I2.1.2.1 [m]
x = distance from the fore side of the stem at the intersection with the upper ice waterline (UIWL) to station under consideration [m]
α = waterline angle [deg], see Figure 2
β' = normal frame angle [deg], see Figure 2
D_{UI} = displacement as defined in I2.1.2.2, not to be taken less than 5 [kt]
CF_C = Crushing failure Class Factor from Table 1
CF_F = Flexural failure Class Factor from Table 1
CF_D = Load patch dimensions Class Factor from Table 1

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

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

$$fa_i = \alpha_i / 30$$

(b) Force, F_i:

$$F_i = fa_i \cdot CF_{CV} \cdot D_{UI}^{0.47} \text{ [MN]}$$

I2 (cont)

(c) Line load, Q_i:

$$Q_i = F_i^{0.22} \cdot CF_{QV} \text{ [MN/m]}$$

(d) Pressure, P_i:

$$P_i = F^{0.56} \cdot CF_{PV} \text{ [MPa]}$$

where i = sub-region considered
α = waterline angle [deg], see Figure 2
D_{UI} = displacement as defined in I2.1.2.2, not to be taken less than 5 [kt]
CF_{CV} = Crushing failure Class Factor from Table 2
CF_{QV} = Line load Class Factor from Table 2
CF_{PV} = Pressure Class Factor from Table 2

I2.3.2.2 Hull areas other than the bow

(i) In the hull areas other than the bow, the force (F_{NonBow}) and line load (Q_{NonBow}) used in the determination of the load patch dimensions (b_{NonBow}, w_{NonBow}) and design pressure (P_{avg}) are determined as follows:

(a) Force, F_{NonBow}:

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

(b) Line Load, Q_{NonBow}:

$$Q_{NonBow} = 0.639 \cdot F_{NonBow}^{0.61} \cdot CF_D \text{ [MN/m]}$$

where CF_C = Crushing failure Class Factor from Table 1
DF = ship displacement factor
$= D_{UI}^{0.64}$ if $D_{UI} \le CF_{DIS}$
$= CF_{DIS}^{0.64} + 0.10 \cdot (D_{UI} - CF_{DIS})$ if $D_{UI} > CF_{DIS}$
D_{UI} = displacement as defined in I2.1.2.2, not to be taken less than 10 [kt]
CF_{DIS} = Displacement Class Factor from Table 1
CF_D = Load patch dimensions Class Factor from Table 1

I2.3.3 Design load patch

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

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

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

where F_{Bow} = maximum force F_i in the Bow area [MN]
Q_{Bow} = maximum line load Q_i in the Bow area [MN/m]
P_{Bow} = maximum pressure P_i in the Bow area [MPa]

I2 (cont)

(ii) In hull areas other than those covered by I2.3.3 (i), the design load patch has dimensions of width, w_{NonBow}, and height, b_{NonBow}, defined as follows:

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

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

where F_{NonBow} = force as defined in I2.3.2.2 (i) (a) [MN]
Q_{NonBow} = line load as defined in I2.3.2.2 (i) (b) [MN/m]

I2.3.4 Pressure within the design load patch

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

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

where $F = F_{Bow}$ or F_{NonBow} as appropriate for the hull area under consideration [MN]
$b = b_{Bow}$ or b_{NonBow} as appropriate for the hull area under consideration [m]
$w = w_{Bow}$ or w_{NonBow} as appropriate for the hull area under consideration [m]

(ii) Areas of higher, concentrated pressure exist within the load patch. In general, smaller areas have higher local pressures. Accordingly, the peak pressure factors listed in Table 3 are used to account for the pressure concentration on localized structural members.

Table 3 - Peak Pressure Factors

Structural member		**Peak Pressure Factor (PPF_i)**
Plating	Transversely-framed	$PPF_p = (1.8 - s) \geq 1.2$
	Longitudinally-framed	$PPF_p = (2.2 - 1.2 \cdot s) \geq 1.5$
Frames in transverse framing systems	With load distributing stringers	$PPF_t = (1.6 - s) \geq 1.0$
	With no load distributing stringers	$PPF_t = (1.8 - s) \geq 1.2$
Frames in bottom structures		$PPF_s = 1.0$
Load carrying stringers Side longitudinals Web frames		$PPF_s = 1.0$, if $S_w \geq 0.5 \cdot w$ $PPF_s = 2.0 - 2.0 \cdot S_w / w$, if $S_w < (0.5 \cdot w)$
where: s = frame or longitudinal spacing [m] S_w = web frame spacing [m] w = ice load patch width [m]		

I2.3.5 Hull area factors

(i) Associated with each hull area is an Area Factor that reflects the relative magnitude of the load expected in that area. The Area Factor (AF) for each hull area is listed in Table 4.

(ii) In the event that a structural member spans across the boundary of a hull area, the largest hull area factor is to be used in the scantling determination of the member.

(iii) Due to their increased manoeuvrability, ships having propulsion arrangements with azimuth thruster(s) or “podded” propellers are to have specially considered Stern Icebelt (S_i) and Stern Lower (S_l) hull area factors.

I2 (cont)

(iv) For ships assigned the additional notation “Icebreaker”, the Area Factor (AF) for each hull area is listed in Table 5.

Table 4 - Hull Area Factors (AF)

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

Note to Table 4: * See I2.3.1 (iii);
** Indicates that strengthening for ice loads is not necessary.

Table 5 - Hull Area Factors (AF) for ships with additional notation “Icebreaker”

Hull area		Area	Polar Class						
			PC1	PC2	PC3	PC4	PC5	PC6	PC7
Bow (B)	All	B	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Bow Intermediate (BI)	Icebelt	BI_i	0.90	0.85	0.85	0.85	0.85	1.00	1.00
	Lower	BI_l	0.70	0.65	0.65	0.65	0.65	0.65	0.65
	Bottom	BI_b	0.55	0.50	0.45	0.45	0.45	0.45	0.45
Midbody (M)	Icebelt	M_i	0.70	0.65	0.55	0.55	0.55	0.55	0.55
	Lower	M_l	0.50	0.45	0.40	0.40	0.40	0.40	0.40
	Bottom	M_b	0.30	0.30	0.25	0.25	0.25	0.25	0.25
Stern (S)	Icebelt	S_i	0.95	0.90	0.80	0.80	0.80	0.80	0.80
	Lower	S_l	0.55	0.50	0.45	0.45	0.45	0.45	0.45
	Bottom	S_b	0.35	0.30	0.30	0.30	0.30	0.30	0.30

I2.4 Shell plate requirements

I2.4.1 The required minimum shell plate thickness, t, is given by:

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

where t_{net} = plate thickness required to resist ice loads according to I2.4.2 [mm]
t_s = corrosion and abrasion allowance according to I2.11 [mm]

I2.4.2 The thickness of shell plating required to resist the design ice load, t_{net}, depends on the orientation of the framing.

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

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

I2 (cont)

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

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

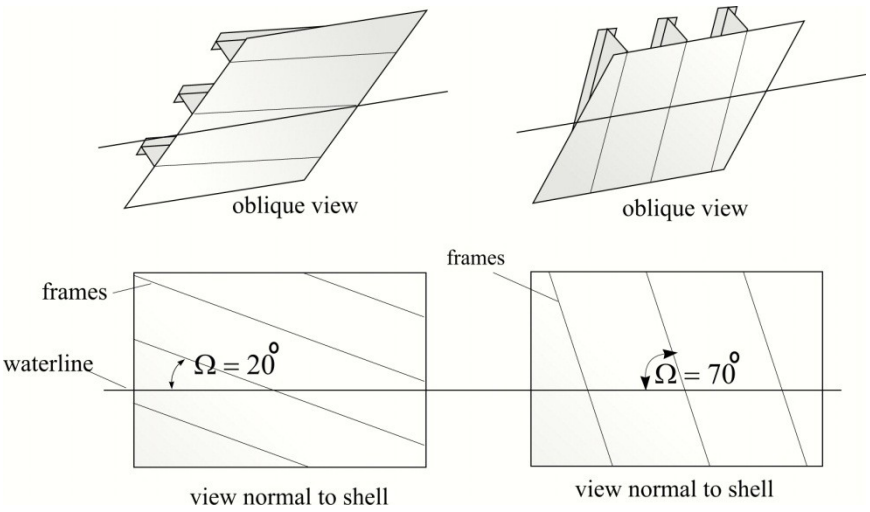
In the case of longitudinally-framed plating ($\Omega \leq 20$ deg), when b < s, the net thickness is given by:

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

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

where Ω = smallest angle between the chord of the waterline and the line of the first level framing as illustrated in Figure 3 [deg].
s = transverse frame spacing in transversely-framed ships or longitudinal frame spacing in longitudinally-framed ships [m]
AF = Hull Area Factor from Table 4 or Table 5
PPF_p = Peak Pressure Factor from Table 3
P_{avg} = average patch pressure as defined in I2.3.4 [MPa]
σ_y = minimum upper yield stress of the material [N/mm^2]
b = height of design load patch [m], where b is to be taken not greater than (*l* – s/4) in the case of determination of the net thickness for transversely framed plating
l = distance between frame supports, i.e. equal to the frame span as given in I2.5.5, but not reduced for any fitted end brackets [m]. When a load-distributing stringer is fitted, the length *l* need not be taken larger than the distance from the stringer to the most distant frame support.

Figure 3 - Shell framing angle Ω



I2
(cont)

I2.5 Framing - General

I2.5.1 Framing members of Polar Class ships are to be designed to withstand the ice loads defined in I2.3.

I2.5.2 The term “framing member” refers to transverse and longitudinal local frames, load-carrying stringers and web frames in the areas of the hull exposed to ice pressure, see Figure 1. Where load-distributing stringers have been fitted, the arrangement and scantlings of these are to be in accordance with the requirements of the Classification Society.

I2.5.3 The strength of a framing member is dependent upon the fixity that is provided at its supports. Fixity can be assumed where framing members are either continuous through the support or attached to a supporting section with a connection bracket. In other cases, simple support is to be assumed unless the connection can be demonstrated to provide significant rotational restraint. Fixity is to be ensured at the support of any framing which terminates within an ice-strengthened area.

I2.5.4 The details of framing member intersection with other framing members, including plated structures, as well as the details for securing the ends of framing members at supporting sections, are to be in accordance with the requirements of the Classification Society.

I2.5.5 The effective span of a framing member is to be determined on the basis of its moulded length. If brackets are fitted, the effective span may be reduced in accordance with the usual practice of the Classification Society. Brackets are to be configured to ensure stability in the elastic and post-yield response regions.

I2.5.6 When calculating the section modulus and shear area of a framing member, net thicknesses of the web, flange (if fitted) and attached shell plating are to be used. The shear area of a framing member may include that material contained over the full depth of the member, i.e. web area including portion of flange, if fitted, but excluding attached shell plating.

I2.5.7 The actual net effective shear area, A_w, of a transverse or longitudinal local frame is given by:

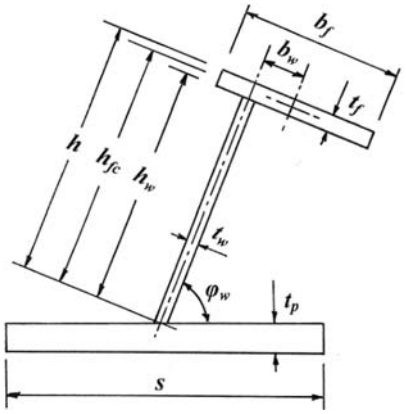
$$A_w = h \cdot t_{wn} \cdot \sin \varphi_w / 100 \ \text{[cm}^2\text{]}$$

where

h = height of stiffener [mm], see Figure 4
t_{wn} = net web thickness [mm]
$= t_w - t_c$
t_w = as built web thickness [mm], see Figure 4
t_c = corrosion deduction [mm] to be subtracted from the web and flange thickness (as specified by the Classification Society, but not less than t_s as required by I2.11.3).
φ_w = smallest angle between shell plate and stiffener web, measured at the midspan of the stiffener, see Figure 4. The angle φ_w may be taken as 90 deg provided the smallest angle is not less than 75 deg.

I2
(cont)

Figure 4 - Stiffener geometry



I2.5.8 When the cross-sectional area of the attached plate flange exceeds the cross-sectional area of the local frame, the actual net effective plastic section modulus, Z_p, of a transverse or longitudinal frame is given by:

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

where

h, t_{wn}, t_c, and φ_w as given in I2.5.7 and s as given in I2.4.2.
A_{pn} = net cross-sectional area of the local frame [cm^2]
t_{pn} = fitted net shell plate thickness [mm] (complying with t_{net} as required by I2.4.2)
h_w = height of local frame web [mm], see Figure 4
A_{fn} = net cross-sectional area of local frame flange [cm^2]
h_{fc} = height of local frame measured to centre of the flange area [mm], see Figure 4
b_w = distance from mid thickness plane of local frame web to the centre of the flange area [mm], see Figure 4

When the cross-sectional area of the local frame exceeds the cross-sectional area of the attached plate flange, the plastic neutral axis is located a distance z_{na} above the attached shell plate, given by:

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

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

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

I2.5.9 In the case of oblique framing arrangement (70 deg > Ω > 20 deg, where Ω is defined as given in I2.4.2), linear interpolation is to be used.

I2 (cont)

I2.6 Framing - Local frames in bottom structures and transverse local frames in side structures

I2.6.1 The local frames in bottom structures (i.e. hull areas B_{lb}, M_b and S_b) and transverse local frames in side structures are to be dimensioned such that the combined effects of shear and bending do not exceed the plastic strength of the member. The plastic strength is defined by the magnitude of midspan load that causes the development of a plastic collapse mechanism.
For bottom structure the patch load shall be applied with the dimension (b) parallel with the frame direction.

I2.6.2 The actual net effective shear area of the frame, A_w, as defined in I2.5.7, is to comply with the following condition: $A_w \geq A_t$, where:

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

where LL = length of loaded portion of span
= lesser of a and b [m]
a = local frame span as defined in I2.5.5 [m]
b = height of design ice load patch as defined in I2.3.3 (i) or I2.3.3 (ii) [m]
s = spacing of local frame [m]
AF = Hull Area Factor from Table 4 or Table 5
PPF = Peak Pressure Factor, PPF_t or PPF_s as appropriate from Table 3
P_{avg} = average pressure within load patch as defined in I2.3.4 [MPa]
σ_y = minimum upper yield stress of the material [N/mm^2]

I2.6.3 The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in I2.5.8, is to comply with the following condition: $Z_p \geq Z_{pt}$, where Z_{pt} is to be the greater calculated on the basis of two load conditions: a) ice load acting at the midspan of the local frame, and b) the ice load acting near a support. The A_1 parameter defined below reflects these two conditions:

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

where AF, PPF, P_{avg}, LL, b, s, a and σ_y are as given in I2.6.2
$Y = 1 - 0.5 \cdot (LL / a)$
A_1 = maximum of
$A_{1A} = 1 / (1 + j / 2 + k_w \cdot j / 2 \cdot [(1 - a_1^2)^{0.5} - 1])$
$A_{1B} = (1 - 1 / (2 \cdot a_1 \cdot Y)) / (0.275 + 1.44 \cdot k_z^{0.7})$
j = 1 for a local frame with one simple support outside the ice-strengthened areas
= 2 for a local frame without any simple supports
$a_1 = A_t / A_w$
A_t = minimum shear area of the local frame as given in I2.6.2 [cm^2]
A_w = effective net shear area of the local frame (calculated according to I2.5.7) [cm^2]
$k_w = 1 / (1 + 2 \cdot A_{fn} / A_w)$ with A_{fn} as given in I2.5.8
$k_z = z_p / Z_p$ in general
= 0.0 when the frame is arranged with end bracket
z_p = sum of individual plastic section moduli of flange and shell plate as fitted [cm^3]
$= (b_f \cdot t_{fn}^2 / 4 + b_{eff} \cdot t_{pn}^2 / 4) / 1000$
b_f = flange breadth [mm], see Figure 4
t_{fn} = net flange thickness [mm]
$= t_f - t_c$ (t_c as given in I2.5.7)
t_f = as-built flange thickness [mm], see Figure 4
t_{pn} = the fitted net shell plate thickness [mm] (not to be less than t_{net} as given in I2.4)

I2 (cont)

b_{eff} = effective width of shell plate flange [mm]
= 500 · s
Z_p = net effective plastic section modulus of the local frame (calculated according to I2.5.8) [cm^3]

I2.6.4 The scantlings of the local frame are to meet the structural stability requirements of I2.9.

I2.7 Framing - Longitudinal local frames in side structures

I2.7.1 Longitudinal local frames in side structures are to be dimensioned such that the combined effects of shear and bending do not exceed the plastic strength of the member. The plastic strength is defined by the magnitude of midspan load that causes the development of a plastic collapse mechanism.

I2.7.2 The actual net effective shear area of the frame, A_w, as defined in I2.5.7, is to comply with the following condition: $A_w \geq A_L$, where:

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

where AF = Hull Area Factor from Table 4 or Table 5
PPF_s = Peak Pressure Factor from Table 3
P_{avg} = average pressure within load patch as defined in I2.3.4 [MPa]
$b_1 = k_o \cdot b_2$ [m]
$k_o = 1 - 0.3 / b'$
$b' = b / s$
b = height of design ice load patch as defined in I2.3.3 (i) or I2.3.3 (ii) [m]
s = spacing of longitudinal frames [m]
$b_2 = b \cdot (1 - 0.25 \cdot b')$ [m], if $b' < 2$
= s [m], if $b' \geq 2$
a = effective span of longitudinal local frame as given in I2.5.5 [m]
σ_y = minimum upper yield stress of the material [N/mm^2]

I2.7.3 The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in I2.5.8, is to comply with the following condition: $Z_p \geq Z_{pL}$, where:

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

where AF, PPF_s, P_{avg}, b_1, a and σ_y are as given in I2.7.2
$A_4 = 1 / (2 + k_{wl} \cdot [(1 - a_4^2)^{0.5} - 1])$
$a_4 = A_L / A_w$
A_L = minimum shear area for longitudinal as given in I2.7.2 [cm^2]
A_w = net effective shear area of longitudinal (calculated according to I2.5.7) [cm^2]
$k_{wl} = 1 / (1 + 2 \cdot A_{fn} / A_w)$ with A_{fn} as given in I2.5.8

I2.7.4 The scantlings of the longitudinals are to meet the structural stability requirements of I2.9.

I2.8 Framing - Web frames and load carrying stringers

I2.8.1 Web frames and load-carrying stringers are to be designed to withstand the ice load patch as defined in I2.3. The load patch is to be applied at locations where the capacity of these members under the combined effects of bending and shear is minimised.

I2 (cont)

I2.8.2 Web frames and load-carrying stringers are to be dimensioned such that the combined effects of shear and bending do not exceed the limit state(s) defined by the Classification Society. Where the structural configuration is such that members do not form part of a grillage system, the appropriate peak pressure factor (PPF) from Table 3 is to be used. Special attention is to be paid to the shear capacity in way of lightening holes and cut-outs in way of intersecting members.

I2.8.3 For determination of scantlings of load carrying stringers, web frames supporting local frames, or web frames supporting load carrying stringers forming part of a structural grillage system, appropriate methods as outlined in I2.17 are normally to be used.

I2.8.4 The scantlings of web frames and load-carrying stringers are to meet the structural stability requirements of I2.9.

I2.9 Framing - Structural stability

I2.9.1 To prevent local buckling in the web, the ratio of web height (h_w) to net web thickness (t_{wn}) of any framing member is not to exceed:

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

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

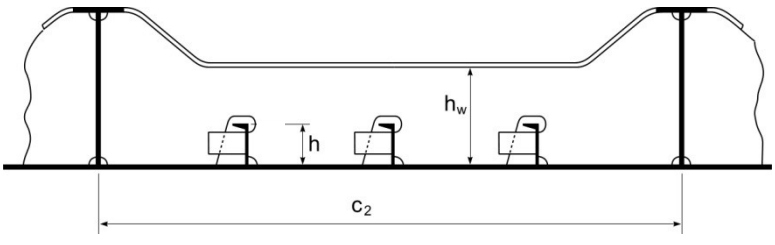
where h_w = web height
t_{wn} = net web thickness
σ_y = minimum upper yield stress of the material [N/mm^2]

I2.9.2 Framing members for which it is not practicable to meet the requirements of I2.9.1 (e.g. load carrying stringers or deep web frames) are required to have their webs effectively stiffened. The scantlings of the web stiffeners are to ensure the structural stability of the framing member. The minimum net web thickness for these framing members is given by:

$$t_{wn} = 2.63 \cdot 10^{-3} \cdot c_1 \cdot (\sigma_y / (5.34 + 4 \cdot (c_1 / c_2)^2))^{0.5} \text{ [mm]}$$

where $c_1 = h_w - 0.8 \cdot h$ [mm]
h_w = web height of stringer / web frame [mm] (see Figure 5)
h = height of framing member penetrating the member under consideration (0 if no such framing member) [mm] (see Figure 5)
c_2 = spacing between supporting structure oriented perpendicular to the member under consideration [mm] (see Figure 5)
σ_y = minimum upper yield stress of the material [N/mm^2]

Figure 5 - Parameter definition of web stiffening



I2 (cont)

I2.9.3 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]

I2.9.4 To prevent local flange buckling of welded profiles, the following are to be satisfied:

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

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

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

where t_{fn} = net thickness of flange [mm]
σ_y = minimum upper yield stress of the material [N/mm^2]

I2.10 Plated structures

I2.10.1 Plated structures are those stiffened plate elements in contact with the hull and subject to ice loads. These requirements are applicable to an inboard extent which is the lesser of:

(i) web height of adjacent parallel web frame or stringer; or
(ii) 2.5 times the depth of framing that intersects the plated structure

I2.10.2 The thickness of the plating and the scantlings of attached stiffeners are to be such that the degree of end fixity necessary for the shell framing is ensured.

I2.10.3 The stability of the plated structure is to adequately withstand the ice loads defined in I2.3.

I2.11 Corrosion/abrasion additions and steel renewal

I2.11.1 Effective protection against corrosion and ice-induced abrasion is recommended for all external surfaces of the shell plating for Polar Class ships.

I2.11.2 The values of corrosion/abrasion additions, t_s, to be used in determining the shell plate thickness are listed in Table 6.

I2.11.3 Polar Class ships are to have a minimum corrosion/abrasion addition of t_s = 1.0 mm applied to all internal structures within the ice-strengthened hull areas, including plated members adjacent to the shell, as well as stiffener webs and flanges.

I2
(cont)

Table 6 - 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

I2.11.4 Steel renewal for ice strengthened structures is required when the gauged thickness is less than t_{net} + 0.5 mm.

I2.12 Materials

I2.12.1 Steel grades of plating for hull structures are to be not less than those given in Table 8 based on the as-built thickness, the Polar Class and the material class of structural members according to I2.12.2.

Table 7 - Material classes for structural members

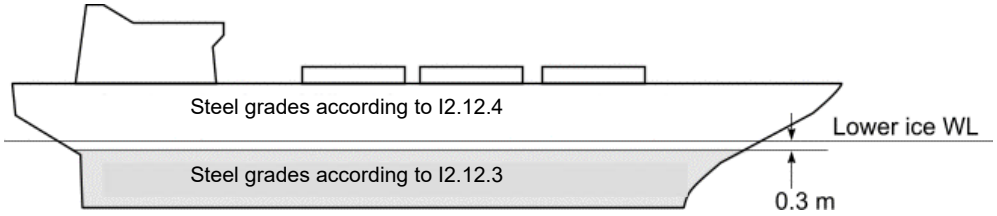
Structural members	Material class
Shell plating within the bow and bow intermediate icebelt hull areas (B, B_{Ii})	II
All weather and sea exposed SECONDARY and PRIMARY, as defined in Table 1 of UR S6.1, structural members outside 0.4 L_{UI} amidships	I
Plating materials for stem and stern frames, rudder horn, rudder, propeller nozzle, shaft brackets, ice skeg, ice knife and other appendages subject to ice impact loads	II
All inboard framing members attached to the weather and sea-exposed plating, including any contiguous inboard member within 600 mm of the plating	I
Weather-exposed plating and attached framing in cargo holds of ships which by nature of their trade have their cargo hold hatches open during cold weather operations	I
All weather and sea exposed SPECIAL, as defined in Table 1 of UR S6.1, structural members within 0.2 L_{UI} from FP	II

I2.12.2 Material classes specified in Table 1 of UR S6.1 are applicable to Polar Class ships regardless of the ship's length. In addition, material classes for weather and sea exposed structural members and for members attached to the weather and sea exposed plating are given in Table 7. Where the material classes in Table 7 and those in Table 1 of UR S6.1 differ, the higher material class is to be applied.

I2.12.3 Steel grades for all plating and attached framing of hull structures and appendages situated below the level of 0.3 m below the lower waterline, as shown in Figure 6, are to be obtained from Table 6 and Table 7 of UR S6 based on the material class for structural members in Table 7 above, regardless of Polar Class.

I2
(cont)

Figure 6 - Steel grade requirements for submerged and weather exposed shell plating



I2.12.4 Steel grades for all weather exposed plating of hull structures and appendages situated above the level of 0.3 m below the lower ice waterline, as shown in Figure 6, are to be not less than given in Table 8.

Table 8 - Steel grades for weather exposed plating[1)]

Thickness, t [mm]	Material class I				Material class II				Material class III					
	PC1-5		PC6&7		PC1-5		PC6&7		PC1-3		PC4&5		PC6&7	
	MS	HT	MS	HT	MS	HT	MS	HT	MS	HT	MS	HT	MS	HT
$t \leq 10$	B	AH	B	AH	B	AH	B	AH	E	EH	E	EH	B	AH
$10 < t \leq 15$	B	AH	B	AH	D	DH	B	AH	E	EH	E	EH	D	DH
$15 < t \leq 20$	D	DH	B	AH	D	DH	B	AH	E	EH	E	EH	D	DH
$20 < t \leq 25$	D	DH	B	AH	D	DH	B	AH	E	EH	E	EH	D	DH
$25 < t \leq 30$	D	DH	B	AH	E	EH[2)]	D	DH	E	EH	E	EH	E	EH
$30 < t \leq 35$	D	DH	B	AH	E	EH	D	DH	E	EH	E	EH	E	EH
$35 < t \leq 40$	D	DH	D	DH	E	EH	D	DH	∅	FH	E	EH	E	EH
$40 < t \leq 45$	E	EH	D	DH	E	EH	D	DH	∅	FH	E	EH	E	EH
$45 < t \leq 50$	E	EH	D	DH	E	EH	D	DH	∅	FH	∅	FH	E	EH
∅ Not applicable														

Notes to Table 8:
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.

I2.12.5 Castings are to have specified properties consistent with the expected service temperature for the cast component.

I2.13 Longitudinal strength

I2.13.1 Application

I2.13.1.1 A ramming impact on the bow is the design scenario for the evaluation of the longitudinal strength of the hull.

I2 (cont)

I2.13.1.2 Intentional ramming is not considered as a design scenario for ships which are designed with vertical or bulbous bows, see I1.1.6. Hence the longitudinal strength requirements given in I2.13 is not to be considered for ships with stem angle γ_{stem} equal to or larger than 80 deg.

I2.13.1.3 Ice loads are only to be combined with still water loads. The combined stresses are to be compared against permissible bending and shear stresses at different locations along the ship's length. In addition, sufficient local buckling strength is also to be verified.

I2.13.2 Design vertical ice force at the bow

I2.13.2.1 The design vertical ice force at the bow, F_{IB}, is to be taken as

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

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

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

K_I = indentation parameter = K_f / K_h

a) for the case of a blunt bow form

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

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

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

$K_h = 0.01 \cdot A_{wp}$ [MN/m]

CF_L = Longitudinal Strength Class Factor from Table 1

e_b = bow shape exponent which best describes the waterplane (see Figures 7 and 8)

= 1.0 for a simple wedge bow form

= 0.4 to 0.6 for a spoon bow form

= 0 for a landing craft bow form

An approximate e_b determined by a simple fit is acceptable

γ_{stem} = stem angle to be measured between the horizontal axis and the stem tangent at the upper ice waterline [deg] (buttock angle as per Figure 2 measured on the centreline)

α_{stem} = waterline angle measured in way of the stem at the upper ice waterline (UIWL) [deg] (see Figure 7)

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

B_{UI} = moulded breadth corresponding to the upper ice waterline (UIWL) [m]

L_B = bow length used in the equation $y = B_{UI} / 2 \cdot (x/L_B)^{eb}$ [m] (see Figures 7 and 8)

D_{UI} = displacement as defined in I2.1.2.2, not to be taken less than 10 [kt]

A_{wp} = waterplane area corresponding to the upper ice waterline (UIWL) [m^2]

CF_F = Flexural Failure Class Factor from Table 1

I2
(cont)

Figure 7 - Bow shape definition

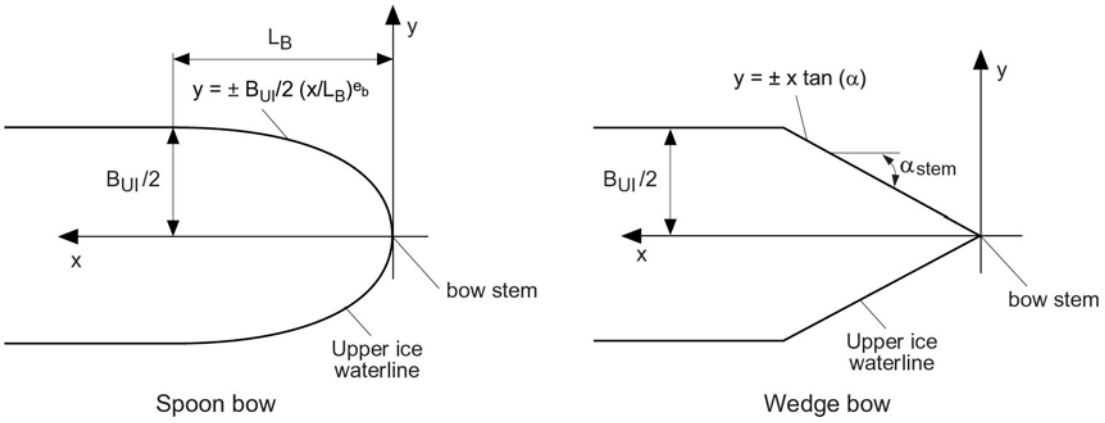
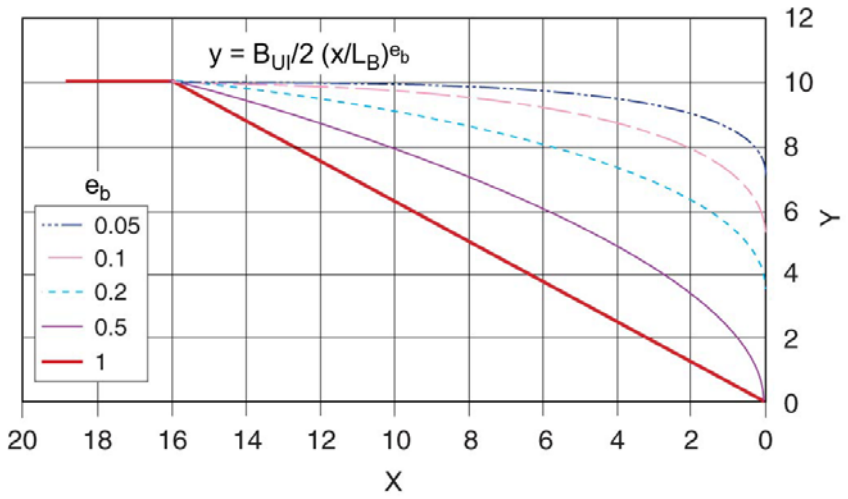


Figure 8 - Illustration of e_b effect on the bow shape for B_{UI} = 20 and L_B =16



I2.13.3 Design vertical shear force

I2.13.3.1 The design vertical ice shear force, F_I, along the hull girder is to be taken as:

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

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

(a) Positive shear force
C_f = 0.0 between the aft end of L_{UI} and 0.6L_{UI} from aft
C_f = 1.0 between 0.9 L_{UI} from aft and the forward end of L_{UI}

(b) Negative shear force
C_f = 0.0 at the aft end of L_{UI}
C_f = -0.5 between 0.2 L_{UI} and 0.6L_{UI} from aft
C_f = 0.0 between 0.8 L_{UI} from aft and the forward end of L_{UI}

Intermediate values are to be determined by linear interpolation

I2 (cont)

I2.13.3.2 The applied vertical shear stress, τ_a, is to be determined along the hull girder in a similar manner as in UR S11.5.4.2 by substituting the design vertical ice shear force for the design vertical wave shear force.

I2.13.4 Design vertical ice bending moment

I2.13.4.1 The design vertical ice bending moment, M_I, along the hull girder is to be taken as:

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

where L_{UI} = length as defined in I2.1.2.1 [m]
γ_{stem} is as given in I2.13.2.1
F_{IB} = design vertical ice force at the bow [MN]
C_m = longitudinal distribution factor for design vertical ice bending moment to be taken as follows:
C_m = 0.0 at the aft end of L_{UI}
C_m = 1.0 between $0.5L_{UI}$ and $0.7L_{UI}$ from aft
C_m = 0.3 at $0.95L_{UI}$ from aft
C_m = 0.0 at the forward end of L_{UI}
Intermediate values are to be determined by linear interpolation

I2.13.4.2 The applied vertical bending stress, σ_a, is to be determined along the hull girder in a similar manner as in UR S11.5.4.1, by substituting the design vertical ice bending moment for the design vertical wave bending moment. The ship still water bending moment is to be taken as the permissible still water bending moment in sagging condition.

I2.13.5 Longitudinal strength criteria

I2.13.5.1 The strength criteria provided in Table 9 are to be satisfied. The design stress is not to exceed the permissible stress.

Table 9 - Longitudinal strength criteria

Failure mode	**Applied stress**	**Permissible stress when $\sigma_y / \sigma_u \leq 0.7$**	**Permissible stress when $\sigma_y / \sigma_u > 0.7$**
Tension	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 UR S11.5 [N/mm^2]
τ_c = critical buckling stress in shear, according to UR S11.5 [N/mm^2]
η = 0.8
η = 0.6 for ships which are assigned the additional notation “Icebreaker”

I2
(cont)

I2.14 Stem and stern frames

I2.14.1 The stem and stern frame are to be designed according to the requirements of the Classification Society. For PC6/PC7 ships requiring IAS/IA equivalency, the stem and stern requirements of the Finnish-Swedish Ice Class Rules may need to be additionally considered.

I2.15 Appendages

I2.15.1 All appendages are to be designed to withstand forces appropriate for the location of their attachment to the hull structure or their position within a hull area.

I2.15.2 Load definition and response criteria are to be determined by the Classification Society.

I2.16 Local details

I2.16.1 For the purpose of transferring ice-induced loads to supporting structure (bending moments and shear forces), local design details are to comply with the requirements of the Classification Society.

I2.16.2 The loads carried by a member in way of cut-outs are not to cause instability. Where necessary, the structure is to be stiffened.

I2.17 Direct calculations

I2.17.1 Direct calculations are not to be utilised as an alternative to the analytical procedures prescribed for the shell plating and local frame requirements given in I2.4, I2.6, and I2.7.

I2.17.2 Direct calculations are to be used for load carrying stringers and web frames forming part of a grillage system.

I2.17.3 Where direct calculation is used to check the strength of structural systems, the load patch specified in I2.3 is to be applied, without being combined with any other loads. The load patch is to be applied at locations where the capacity of these members under the combined effects of bending and shear is minimised. Special attention is to be paid to the shear capacity in way of lightening holes and cut-outs in way of intersecting members.

I2.17.4 The strength evaluation of web frames and stringers may be performed based on linear or non-linear analysis. Recognized structural idealisation and calculation methods are to be applied, but the detailed requirements are to be specified by the Classification Society. In the strength evaluation, the guidance given in I2.17.5 and I2.17.6 may generally be considered.

I2.17.5 If the structure is evaluated based on linear calculation methods, the following are to be considered:

(1) Web plates and flange elements in compression and shear to fulfil relevant buckling criteria as specified by the Classification Society

(2) Nominal shear stresses in member web plates to be less than $\sigma_y / \sqrt{3}$

(3) Nominal von Mises stresses in member flanges to be less than $1.15\ \sigma_y$

I2 (cont)

I2.17.6 If the structure is evaluated based on non-linear calculation methods, the following are to be considered:

(1) The analysis is to reliably capture buckling and plastic deformation of the structure

(2) The acceptance criteria are to ensure a suitable margin against fracture and major buckling and yielding causing significant loss of stiffness

(3) Permanent lateral and out-of plane deformation of considered member are to be minor relative to the relevant structural dimensions

(4) Detailed acceptance criteria to be decided by the Classification Society

I2.18 Welding

I2.18.1 All welding within ice-strengthened areas is to be of the double continuous type.

I2.18.2 Continuity of strength is to be ensured at all structural connections.

End of Document

I3 Machinery Requirements for Polar Class Ships

(Aug 2006) (Rev.1 Jan2007) (Corr.1 Oct 2007) (Rev.2 Jan 2023 Complete Revision) (Corr.1 Dec 2024)

1 Application

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

The vessel operating conditions are defined in UR I1.

The requirements herein are additional to those applicable for the basic open water class of the vessel.

2 General

2.1 The following drawings and particulars are to be submitted.

2.1.1 Details of the intended environmental operational conditions and the required ice strengthening for the machinery, if different from ship's ice class

2.1.2 Detailed drawings and descriptions of the main propulsion, steering, emergency and auxiliary machinery and information on the essential main propulsion load control functions. The descriptions are to include operational limitations.

2.1.3 Description detailing where main, emergency and auxiliary systems are located and how they are protected to prevent problems from freezing, ice and snow accumulation and evidence of their capability to operate in the intended environmental conditions

2.1.4 Calculations and documentation indicating compliance with the requirements of this chapter

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Note:

1. This UR is to be uniformly applied by IACS Societies on ships contracted for construction on and after 1 March 2008.

2. The "contracted for construction" date means the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. For further details regarding the date of "contract for construction", refer to IACS Procedural Requirement (PR) No. 29.

3. Rev.2 of this UR is to be uniformly applied by IACS Societies on ships contracted for construction on or after 1 July 2024.

I3
(cont)

2.2 System Design

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

2.2.2 Vessels classed PC1 to PC5 inclusive shall have means provided to ensure sufficient vessel operation in the case of propeller damage including the Controllable Pitch (CP) mechanism. Sufficient vessel operation means that the vessel should be able to reach safe haven (safe location) where repairs can be undertaken. This may be achieved either by a temporary repair at sea, or by towing, assuming assistance is available. This would lead however to a condition of approval.

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

2.2.4 The propeller shall be fully submerged at the ships LIWL.

3 Materials

Materials shall be of an approved ductile material. Ferritic nodular cast iron may be used for parts other than bolts. For nodular cast iron an averaged impact energy value of 10 J at testing temperature is regarded as equivalent to the Charpy V test requirements defined below.

3.1 Materials exposed to sea water

Materials exposed to sea water, such as propeller blades, propeller hubs and cast thruster bodies shall have an elongation not less than 15% on a test specimen according to UR W2.

Charpy V-notch impact testing is to be carried out for materials other than bronze and austenitic steel. The tests shall be carried out on three specimens at minus 10 ºC, and the average energy value is to be not less than 20 J. However, Charpy V impact test requirements of UR W7 or UR W27 as applicable for ships with ice class notation, shall also be applied to ships covered by this UR.

3.2 Materials exposed to sea water temperature

Charpy V-notch impact testing is to be carried out for materials other than bronze and austenitic steel. The tests shall be carried out on three specimens at minus 10 ºC, and the average energy value is to be not less than 20 J. However, the Charpy V impact test requirements of UR W7 as applicable for ships with ice class notation, shall also be applied to ships covered by this UR.

This requirement applies to components such as but not limited to blade bolts, CP-mechanisms, shaft bolts, propeller shaft, strut-pod connecting bolts, etc. This requirement does not apply to surface hardened components, such as bearings and gear teeth or sea water cooling lines (heat exchangers, pipes, valves, fittings etc.). For a definition of structural boundaries exposed to sea water temperature see UR I2 Figure 6.

3.3 Material exposed to low air temperature

Materials of exposed machinery and foundations shall be manufactured from steel or other approved ductile material. An average impact energy value of 20 J taken from three Charpy V tests is to be obtained at 10 ºC below the lowest design temperature. Charpy V impact tests are not required for bronze and austenitic steel.

I3 (cont)

This requirement does not apply to surface hardened components, such as bearings and gear teeth. For a definition of structural boundaries exposed to air temperature see UR I2 Figure 6.

4 Definitions

4.1 Definition of Symbols

Table 1: Definition of symbols

Symbol	Unit	Definition
c	m	chord length of blade section
$c_{0.7}$	m	chord length of blade section at $0.7R$ propeller radius
CP	-	controllable pitch
D	m	propeller diameter
d	m	external diameter of propeller hub (at propeller plane)
d_{pin}	mm	diameter of shear pin
D_{limit}	m	limit value for propeller diameter
EAR		expanded blade area ratio
F_b	kN	maximum backward blade force for the ship's service life (negative sign)
F_{ex}	kN	ultimate blade load resulting from blade failure through plastic bending
F_f	kN	maximum forward blade force for the ship's service life (positive sign)
F_{ice}	kN	ice load
$(F_{ice})_{max}$	kN	maximum ice load for the ship's service life
FP	-	fixed pitch
h_0	m	depth of the propeller centreline from lower ice waterline ($LIWL$)
(H_{ice})	m	Ice block dimension for propeller load definition
I	kgm^2	equivalent mass moment of inertia of all parts on engine side of component under consideration
I_t	kgm^2	equivalent mass moment of inertia of the whole propulsion system
k	-	shape parameter for Weibull distribution
$LIWL$	m	lower ice waterline
m	-	slope for S-N curve in log/log scale
M_{BL}	kNm	blade bending moment
MCR	-	maximum continuous rating
N	-	number of ice load cycles
n	rev/s	propeller rotational speed
n_n	rev/s	nominal propeller rotational speed at MCR in free running condition
N_{class}	-	reference number of ice impacts per propeller revolution per ice class
N_{ice}	-	total number of ice load cycles on propeller blade for the ship's service life
N_R	-	reference number of ice load cycles for equivalent fatigue stress (10^8 cycles)
N_Q	-	number of propeller revolutions during a milling sequence
$P_{0.7}$	m	propeller pitch at $0.7R$ radius
$P_{0.7n}$	m	propeller pitch at $0.7R$ radius at MCR in free running condition
$P_{0.7b}$	m	propeller pitch at $0.7R$ radius at MCR in bollard condition
PCD	m	pitch circle diameter

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$Q(\varphi)$	kNm	Torque
Q_{Amax}	kNm	maximum response torque amplitude as a simulation result
Q_{emax}	kNm	maximum engine torque
$Q_F(\varphi)$	kNm	Ice torque excitation for frequency domain calculations
Q_{fr}	kNm	friction torque in pitching mechanism; reduction of spindle torque
Q_{max}	kNm	maximum torque on the propeller resulting from propeller/ice interaction
Q_{motor}	kNm	electric motor peak torque
Q_n	kNm	nominal torque at MCR in free running condition
$Q_r(t)$	kNm	response torque along the propeller shaft line
Q_{peak}	kNm	maximum of the response torque $Q_r(t)$
Q_{smax}	kNm	maximum spindle torque of the blade for the ship's service life
Q_{sex}	kNm	extreme spindle torque corresponding to the blade failure load F_{ex}
Q_{vib}	kNm	Vibratory torque at considered component, taken from frequency domain open water TVC
R	m	propeller radius
S	-	Safety factor
S_{fat}	-	Safety factor for fatigue
S_{ice}	-	Ice strength index for blade ice force
r	m	blade section radius
T	kN	Hydrodynamic propeller thrust in bollard condition
T_b	kN	maximum backward propeller ice thrust for the ship's service life
T_f	kN	maximum forward propeller ice thrust for the ship's service life
T_n	kN	propeller thrust at MCR in free running condition
T_r	kN	maximum response thrust along the shaft line
T_{kmax}	kNm	maximum torque capacity of flexible coupling
T_{kmax2}	kNm	T_{kmax} at $N = 1$ load cycle
T_{max1}	kNm	T_{kmax} at $N = 5 \times 10^4$ load cycles
T_{kv}	kNm	vibratory torque amplitude at $N = 10^6$ load cycles
ΔT_{kmax}	kNm	maximum range of T_{kmax} at $N = 5 \times 10^4$ load cycles
t	m	maximum blade section thickness
Z	-	number of propeller blades
z_{pin}	-	number of shear pins
α_i	deg	duration of propeller blade/ice interaction expressed in rotation angle
γ_ε	-	the reduction factor for fatigue; scatter and test specimen size effect
γ_v	-	the reduction factor for fatigue; variable amplitude loading effect
γ_m	-	the reduction factor for fatigue; mean stress effect
ρ	-	a reduction factor for fatigue correlating the maximum stress amplitude to the equivalent fatigue stress for 10^8 stress cycles
$\sigma_{0.2}$	MPa	proof yield strength (at 0.2% plastic strain) of material
σ_{exp}	MPa	mean fatigue strength of blade material at 10^8 cycles to failure in sea water
σ_{fat}	MPa	equivalent fatigue ice load stress amplitude for 10^8 stress cycles
σ_{fl}	MPa	characteristic fatigue strength for blade material
σ_{ref1}	MPa	reference stress $\sigma_{ref1} = 0.6\,\sigma_{0.2} + 0.4\,\sigma_u$
σ_{ref2}	MPa	reference stress $\sigma_{ref2} = 0.7\,\sigma_u$ or $\sigma_{ref2} = 0.6\,\sigma_{0.2} + 0.4\,\sigma_u$ whichever is less

I3 (cont)

σ_{st}	MPa	maximum stress resulting from F_b or F_f
σ_u	MPa	ultimate tensile strength of blade material
$(\sigma_{ice})_{bmax}$	MPa	principal stress caused by the maximum backward propeller ice load
$(\sigma_{ice})_{fmax}$	MPa	principal stress caused by the maximum forward propeller ice load
$(\sigma_{ice})_{Amax}$	MPa	maximum ice load stress amplitude at the considered location on the blade
σ_{mean}	MPa	mean stress
$(\sigma_{ice})_A(N)$	MPa	blade stress amplitude distribution

4.2 Definition of Loads

Table 2: Definitions of loads

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

I3 (cont)

Q_{max}	The maximum ice-induced torque resulting from propeller/ice interaction on one propeller blade, including hydrodynamic loads on that blade.	Is used for estimation of the response torque Q_r along the propulsion shaft line and as excitation for torsional vibration calculations.
F_{ex}	Ultimate blade load resulting from blade loss through plastic bending. The force that is needed to cause total failure of the blade so that plastic hinge is caused to the root area. The force is acting on $0.8R$.	Blade failure load is used to dimension the blade bolts, pitch control mechanism, propeller shaft, propeller shaft bearing and trust bearing. The objective is to guarantee that total propeller blade failure should not cause damage to other components.
Q_{sex}	Maximum spindle torque resulting from blade failure load	Is used to ensure pyramid strength principle for the pitching mechanism
Q_r	Maximum response torque along the propeller shaft line, taking into account the dynamic behaviour of the shaft line for ice excitation (torsional vibration) and hydrodynamic mean torque on propeller.	Design torque for propeller shaft line components.
T_r	Maximum response thrust along shaft line, taking into account the dynamic behaviour of the shaft line for ice excitation (axial vibration) and hydrodynamic mean thrust on propeller.	Design thrust for propeller shaft line components.

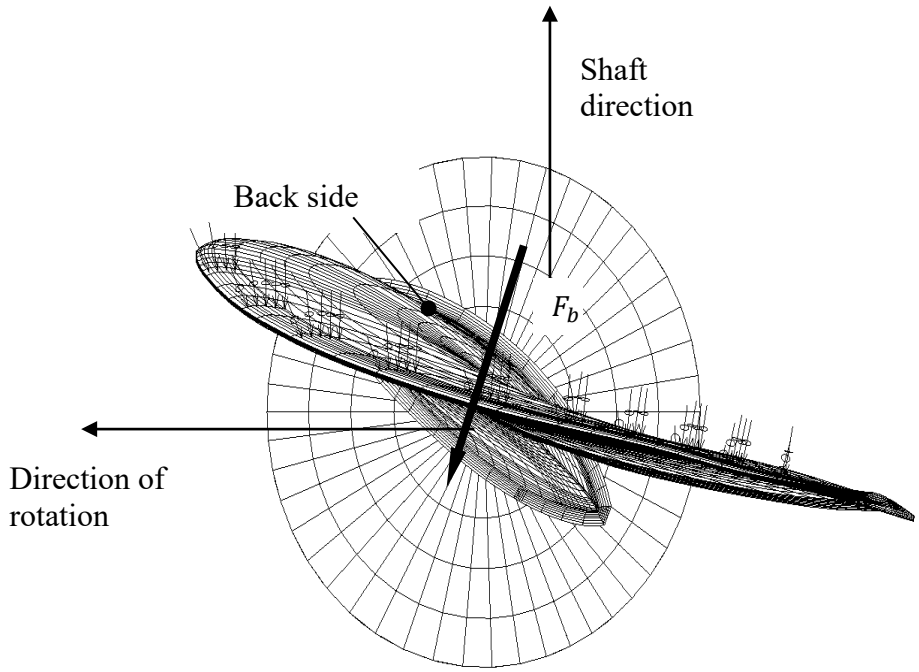


Figure 1 : Direction of the backward blade force resultant taken perpendicular to the chord line at radius $0.7R$. Ice contact pressure at leading edge is shown with small arrows.

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(cont)

5 Design Ice Loads

5.1 General

These Rules cover open and ducted type propellers situated at the stern of a vessel having controllable pitch or fixed pitch blades. Ice loads on bow-mounted propellers shall receive special consideration at the discretion of each classification society. The given loads are expected, single occurrence, maximum values for the whole ship's service life for normal operational conditions, including loads resulting from directional change of rotation where applicable. These loads do not cover off-design operational conditions, for example when a stopped propeller is dragged through ice. These Rules also cover loads due to propeller ice interaction for azimuthing and fixed thrusters with geared transmission or an integrated electric motor ("geared and podded propulsors"). However, the load models of the regulations do not include propeller/ice interaction loads when ice enters the propeller of a turned azimuthing thruster from the side (radially) or loads when ice blocks hit on the propeller hub of a pulling propeller. Ice loads resulting from ice impacts on the body of thrusters shall be estimated on a case by case basis, however are not included within the following section.

The loads given in section 5.3 are total loads including ice-induced loads and hydrodynamic loads (unless otherwise stated) during ice interaction and are to be applied separately (unless otherwise stated) and are intended for component strength calculations only.

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

5.2 Ice Class Factors

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

Table 3: Design Ice Class Factors

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

I3 (cont)

5.3 Propeller Ice Interaction Loads

5.3.1 Maximum backward blade force F_b for open propellers

when $D < D_{limit}$:

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

when $D \geq D_{limit}$:

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

where:

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

Here n is the nominal rotational speed at MCR in the free running open water condition for CP-propellers and 85% of the nominal rotational speed (at MCR free running condition) for a FP-propeller (regardless driving engine type) [rps].

For vessels with the additional notation Icebreaker, the above stated backward blade force F_b shall be multiplied by a factor of 1.1.

5.3.2 Maximum forward blade force F_f for open propellers

when $D < D_{limit}$:

$$F_f = 250 \cdot \left(\frac{EAR}{Z}\right) \cdot D^2 \quad \text{[kN]} \quad \text{[Equation 4]}$$

when $D \geq D_{limit}$:

$$F_f = 500 \cdot \left(\frac{1}{1-\frac{d}{D}}\right) \cdot H_{ice} \cdot \left(\frac{EAR}{Z}\right) \cdot D \quad \text{[kN]} \quad \text{[Equation 5]}$$

where:

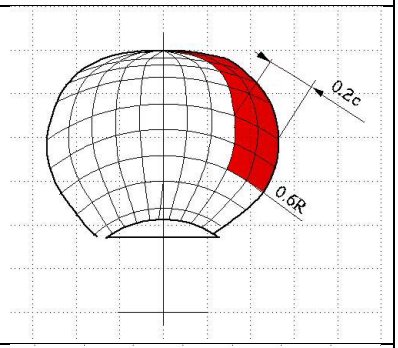
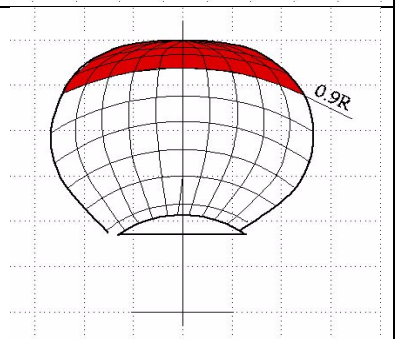
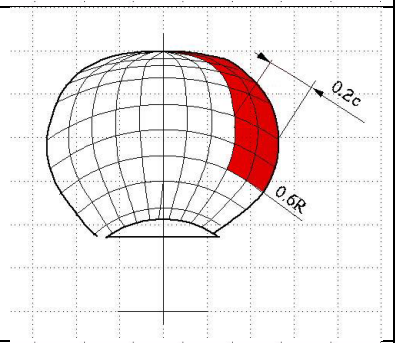
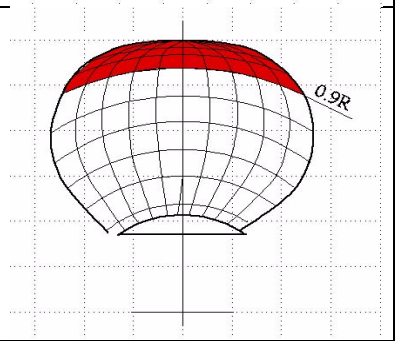
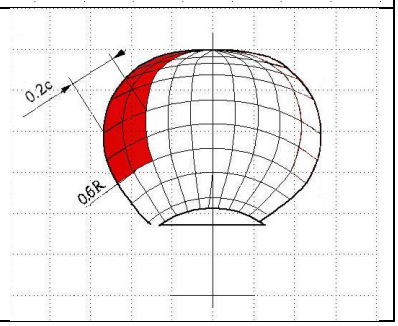
$$D_{limit} = \left(\frac{2}{1-\frac{d}{D}}\right) \cdot H_{ice} \quad \text{[m]} \quad \text{[Equation 6]}$$

5.3.3 Loaded area on the blade for open propellers

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

I3
(cont)

Table 4: Loaded areas and load case definition for open propellers

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

I3 (cont)

5.3.4 Maximum backward blade ice force F_b for ducted propellers

when $D < D_{limit}$:

$$F_b = 9.5 \cdot S_{ice} \cdot (n \cdot D)^{0.7} \cdot \left(\frac{EAR}{Z}\right)^{0.3} \cdot D^2 \quad \text{[kN]} \quad \text{[Equation 7]}$$

when $D \geq D_{limit}$:

$$F_b = 66 \cdot S_{ice} \cdot (n \cdot D)^{0.7} \cdot \left(\frac{EAR}{Z}\right)^{0.3} \cdot (H_{ice})^{1.4} \cdot D^{0.6} \quad \text{[kN]} \quad \text{[Equation 8]}$$

where:

$$D_{limit} = 4 \cdot H_{ice} \quad \text{[m]} \quad \text{[Equation 9]}$$

n shall be taken as in 5.3.1

For vessels with the additional notation Icebreaker, the above stated backward blade force F_b shall be multiplied by a factor 1.1.

5.3.5 Maximum forward blade ice force F_f for ducted propellers

when $D \leq D_{limit}$:

$$F_f = 250 \cdot \left(\frac{EAR}{Z}\right) \cdot D^2 \quad \text{[kN]} \quad \text{[Equation 10]}$$

when $D > D_{limit}$:

$$F_f = 500 \cdot \left(\frac{EAR}{Z}\right) \cdot D \cdot \frac{1}{\left(1-\frac{d}{D}\right)} \cdot H_{ice} \quad \text{[kN]} \quad \text{[Equation 11]}$$

where:

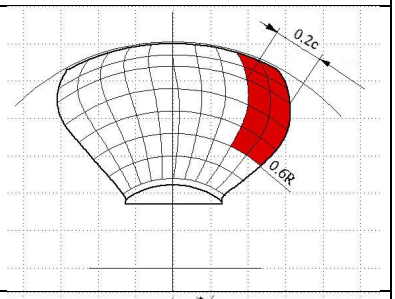
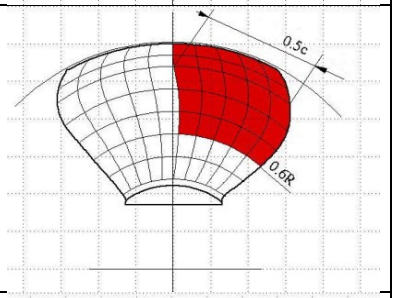
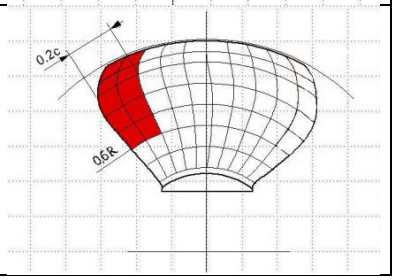
$$D_{limit} = \frac{2}{\left(1-\frac{d}{D}\right)} \cdot H_{ice} \quad \text{[m]} \quad \text{[Equation 12]}$$

5.3.6 Loaded area on the blade for ducted propellers

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

I3 (cont)

Table 5: Loaded areas and load case definition for ducted propellers

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

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

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

Default value $Q_{smax} = 0.25 \cdot F \cdot c_{0.7}$ [kNm] [Equation 13]

where:

F is taken as either F_b or F_f, whichever has the greater absolute value.

The blade failure spindle torque Q_{sex} is defined under 5.4.

I3
(cont)

5.3.8 Load distributions (spectra) for blade loads

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

$$P\left(\frac{F_{ice}}{(F_{ice})_{max}} \geq \frac{F}{(F_{ice})_{max}}\right) = e^{\left(-\left(\frac{F}{(F_{ice})_{max}}\right)^{k} \cdot ln(N_{ice})\right)} \quad \text{[Equation 14]}$$

where:

k = shape parameter of the spectrum

N_{ice} = number of load cycles in the spectrum, see 5.3.9

F_{ice} = random variable for ice loads on the blade, $0 \leq F_{ice} \leq (F_{ice})_{max}$.

This results in a blade stress amplitude distribution

$$(\sigma_{ice})_A(N) = (\sigma_{ice})_{Amax} \cdot \left(1 - \frac{\log(N)}{\log(N_{ice})}\right)^{\frac{1}{k}} \quad \text{[Equation 15]}$$

where:

$$(\sigma_{ice})_{Amax} = \frac{(\sigma_{ice})_{fmax} - (\sigma_{ice})_{bmax}}{2} \quad \text{[Equation 16]}$$

The shape parameter k = 0.75 shall be used for the ice force distribution of an open propeller and the shape parameter k = 1.0 for that of a ducted propeller blade.

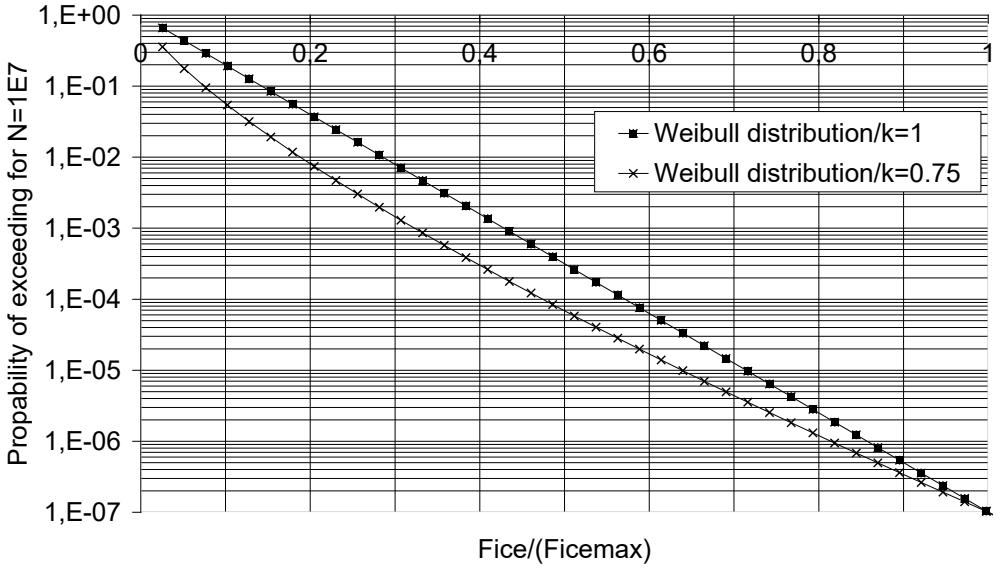


Figure 2: The Weibull-type distribution (probability that F_{ice} exceeds $(F_{ice})_{max}$) that is used for fatigue design.

I3 (cont)

5.3.9 Number of ice loads

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

$$N_{ice} = k_1 \cdot k_2 \cdot k_3 \cdot N_{class} \cdot n \qquad \text{[Equation 17]}$$

where:

N_{class} = reference number of impacts per propeller revolution for each ice class (Table 6)

Table 6: Reference number of impacts

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

k_1 = 1 for centre propeller

= 2 for wing propeller

= 3 for pulling propeller (wing and centre)

k_2 = 0.8 - f when f < 0

= 0.8 - 0.4·f when 0 ≤ f ≤ 1

= 0.6 - 0.2·f when 1< f ≤ 2.5

= 0.1 when f > 2.5

where the immersion function f is:

$$f = \frac{h_0 - H_{ice}}{D/2} - 1 \qquad \text{[Equation 18]}$$

If h_0 is not known, $h_o = D/2$.

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

For vessels with the additional notation Icebreaker, the above stated number of load cycles N_{ice} shall be multiplied by a factor of 3.

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

5.4 Blade Failure Load for both Open and Ducted Propellers

5.4.1 Bending Force, F_{ex}

The minimum load required resulting in blade failure through plastic bending. This shall be calculated iteratively along the radius of the blade from blade root to 0.5R using below Equation 19 with the ultimate load assumed to be acting at 0.8R in the weakest direction.

I3 (cont)

The blade failure load is:

$$F_{ex} = \frac{0.3 \cdot c \cdot t^2 \cdot \sigma_{ref1}}{0.8 \cdot D - 2 \cdot r} \cdot 10^3 \quad \text{[kN]} \quad \text{[Equation 19]}$$

where:

$$\sigma_{ref1} = 0.6 \cdot \sigma_{0.2} + 0.4 \cdot \sigma_u \quad \text{[MPa]}$$

σ_u (minimum ultimate tensile strength to be specified on the drawing) and $\sigma_{0.2}$ (minimum yield or 0.2% proof strength to be specified on the drawing) are representative values for the blade material

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

The classification society may approve alternative means of failure load calculation by means of an appropriate stress analysis reflecting the non-linear plastic material behaviour of the actual blade. A blade is regarded as having failed, if the tip is bent by more than 10% of the propeller diameter.

5.4.2 Spindle Torque, Q_{sex}

The maximum spindle torque due to a blade failure load acting at 0.8R shall be determined. The force that causes blade failure typically reduces when moving from the propeller centre towards the leading and trailing edges. At a certain distance from the blade centre of rotation the maximum spindle torque will occur. This maximum spindle torque shall be defined by an appropriate stress analysis or using equation 20 below.

$$Q_{sex} = max(c_{LE0.8}; 0.8 \cdot c_{TE0.8}) \cdot C_{spex} \cdot F_{ex} \quad \text{[kNm]} \quad \text{[Equation 20]}$$

where :

$$C_{spex} = C_{sp} \cdot C_{fex} = 0.7 \cdot \left(1 - \left(4 \cdot \frac{EAR}{Z}\right)^3\right) \quad \text{[-]} \quad \text{[Equation 21]}$$

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

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

If C_{spex} is below 0.3, a value of 0.3 shall be used for C_{spex}.

$c_{LE0.8}$c is the leading edge portion of the chord length at 0.8R

$c_{TE0.8}$ is the trailing edge portion of the chord length at 0.8R

The figure below illustrates the spindle torque values due to blade failure loads across the whole chord length.

I3
(cont)

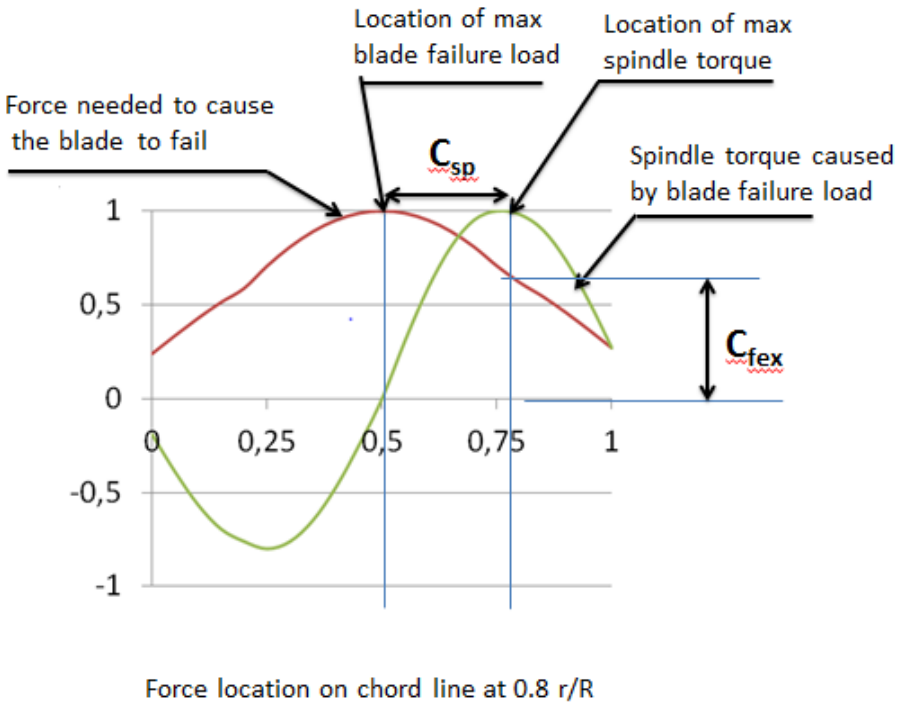


Figure 3: Schematic figure showing blade failure load and related spindle torque when the force acts at different location on the chord line at radius 0.8R.

5.5 Axial design loads acting on open and ducted propellers

5.5.1 Maximum ice thrust on propeller T_f and T_b acting on open and ducted propellers

The maximum forward and backward ice thrusts are given by the following formula:

$T_f = 1.1 \cdot F_f$ [kN] [Equation 22]

$T_b = 1.1 \cdot F_b$ [kN] [Equation 23]

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

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

The design thrust along the propeller shaft line is to be calculated with the formulae below. The greater value of the forward and backward directional load shall be taken as the design load for both directions. The factors 2.2 and 1.5 take into account the dynamic magnification resulting from axial vibration.

In a forward direction

$T_r = T + 2.2 \cdot T_f$ [kN] [Equation 24]

In a backward direction

$T_r = 1.5 \cdot T_b$ [kN] [Equation 25]

I3 (cont)

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

Table 7: Guidance for bollard thrust values

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

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

For pulling type propellers ice interaction loads on propeller hub must be considered in addition to the above. These will be specially considered by the Classification Society.

5.6 Torsional design loads acting on open and ducted propellers

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

Q_{max} is the maximum torque on a propeller resulting from ice/propeller interaction.

when $D < D_{limit}$:

$$Q_{max} = k_{open} \cdot \left(1 - \frac{d}{D}\right) \cdot \left(\frac{P_{0.7}}{D}\right)^{0.16} \cdot (n \cdot D)^{0.17} \cdot D^3 \quad \text{[kNm]} \quad \text{[Equation 26]}$$

where:

k_{open} = 14.7 for PC1 – PC5; and

k_{open} = 10.9 for PC6 – PC7

when $D \geq D_{limit}$:

$$Q_{max} = 1.9 \cdot k_{open} \cdot \left(1 - \frac{d}{D}\right) \cdot (H_{ice})^{1.1} \cdot \left(\frac{P_{0.7}}{D}\right)^{0.16} \cdot (n \cdot D)^{0.17} \cdot D^{1.9} \quad \text{[kNm]} \quad \text{[Equation 27]}$$

where:

$$D_{limit} = 1.8 \cdot H_{ice} \quad \text{[m]} \quad \text{[Equation 28]}$$

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

Table 8: Guidance for rotational speeds to calculate torsional loads

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

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

I3 (cont)

5.6.2 Design ice torque on propeller Q_{max} for ducted propellers

when $D < D_{limit}$:

$$Q_{max} = k_{ducted} \cdot \left(1 - \frac{d}{D}\right) \cdot \left(\frac{P_{0.7}}{D}\right)^{0.16} \cdot (n \cdot D)^{0.17} \cdot D^3 \quad \text{[kNm]} \quad \text{[Equation 29]}$$

where:

k_{ducted} = 10.4 for PC1 – PC5; and

k_{ducted} = 7.7 for PC6 – PC7

when $D \geq D_{limit}$:

$$Q_{max} = 1.9 \cdot k_{ducted} \cdot \left(1 - \frac{d}{D}\right) \cdot (H_{ice})^{1.1} \cdot \left(\frac{P_{0.7}}{D}\right)^{0.16} \cdot (nD)^{0.17} \cdot D^{1.9} \text{ [kNm]} \quad \text{[Equation 30]}$$

where:

$$D_{limit} = 1.8 \cdot H_{ice} \quad \text{[m]} \quad \text{[Equation 31]}$$

n shall be taken as in 5.6.1.

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

5.6.3 Ice torque excitation for open and ducted propellers

The given excitations are used to estimate the maximum torque likely to be experienced once during the service life of the ship. The following load cases are intended to reflect the operational loads on the propulsion system when the propeller interacts with ice and the corresponding reaction of the complete system. The ice impact and system response cause loads in the individual shaft line components. The ice torque Q_{max} may be taken as a constant value in the complete speed range. When considerations at specific shaft speeds are performed a relevant Q_{max} may be calculated using the relevant speed.

Diesel engine plants without an elastic coupling shall be calculated at the least favourable phase angle for ice versus engine excitation, when calculated in time domain. The engine firing pulses shall be included in the calculations and their standard steady state harmonics can be used. A phase angle between ice and gas force excitation does not need to be regarded in frequency domain analysis. Misfiring does not need to be considered.

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

See also Guidelines for calculations given in 5.7

I3 (cont)

5.6.3.1 Excitation for the time domain calculation

The propeller ice torque excitation for shaft line transient dynamic analysis (time domain) is defined as a sequence of blade impacts which are of half sine shape and occur at the blade. The torque due to a single blade ice impact as a function of the propeller rotation angle is then defined as:

$$Q(\varphi) = C_q \cdot Q_{max} \cdot \sin\left(\varphi(180/\alpha_i)\right) \quad \text{[Equation 32]}$$

when φ rotates from 0 to α_i plus integer revolutions.

$$Q(\varphi) = 0$$

when φ rotates from α_i to 360 plus integer revolutions.

Where

φ = rotation angle starting when the first impact occurs

C_q and α_i parameters are given in the Table 9 below. α_i is the duration of propeller blade/ice interaction expressed in propeller rotation angle.

Table 9: Ice impact magnification and duration factors for different blade numbers

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

The total ice torque is obtained by summing the torque of single blades, taking into account the phase shift 360 deg./Z.

At the beginning and at the end of the milling sequence (within calculated duration) linear ramp functions shall be used to increase C_q to its maximum within one propeller revolution and vice versa to decrease it to zero (see examples for different Z numbers in the appendix).

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

$$N_Q = 2 \cdot H_{ice} \quad \text{[Equation 33]}$$

The number of impacts is $Z \cdot N_Q$ for blade order excitation.

An illustration of all excitation cases for different blade numbers is given in the Appendix.

The dynamic simulation shall be performed for all excitation cases starting at MCR nominal, MCR bollard condition and just above all resonance speeds (1st engine and 1st blade harmonic), so that the resonant vibration responses can be obtained. For a fixed pitch propeller plant the dynamic simulation shall also cover bollard pull condition with a corresponding speed assuming maximum possible output of the engine.

I3 (cont)

If a speed drop occurs down to stand still of the main engine, it indicates that the engine may not be sufficiently powered for the intended service task. For the consideration of loads, the maximum occurring torque during the speed drop process shall be applied. On these cases, the excitation shall follow the shaft speed.

5.6.3.2 Frequency domain excitation

For frequency domain calculations the following torque excitation may be used. The excitation has been derived so that the time domain half sine impact sequences have been assumed to be continuous and the Fourier series components for blade order and twice the blade order components have been derived. The frequency domain analysis is generally considered as conservative compared to the time domain simulation provided there is a first blade order resonance in the considered speed range.

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

[Equation 34]

where :

C_{q0} = mean torque component

C_{q1} = first blade order excitation amplitude

C_{q2} = second blade order excitation amplitude

E_0 = number of ice blocks in contact

φ = angle of rotation

$\alpha_{1,2}$ = phase angle of excitation component

Z = number of blades

Table 10: Coefficients for simplified excitation torque estimation

Torque excitation	Z=3					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0,375	0.375	-90	0	0	1
Excitation case 2	0.7	0.33	-90	0.05	-45	1
Excitation case 3	0.25	0.25	-90	0	0	2
Excitation case 4	0.2	0.25	0	0.05	-90	1
Torque excitation	Z=4					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.45	0.36	-90	0.06	-90	1
Excitation case 2	0.9375	0	-90	0.0625	-90	1
Excitation case 3	0.25	0.251	-90	0	0	2
Excitation case 4	0.2	0.25	0	0.05	-90	1
Torque excitation	Z=5					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.45	0.36	-90	0.06	-90	1
Excitation case 2	1.19	0.17	-90	0.02	-90	1
Excitation case 3	0.3	0.25	-90	0.048	-90	2
Excitation case 4	0.2	0.25	0	0.05	-90	1

I3
(cont)

Torque excitation	Z=6					
	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
Excitation case 1	0.45	0.375	-90	0.05	-90	1
Excitation case 2	1.435	0.1	-90	0	0	1
Excitation case 3	0.3	0.25	-90	0.048	-90	2
Excitation case 4	0.2	0.25	0	0.05	-90	1

Torsional vibration responses shall be calculated for all excitation cases.

The results of the relevant excitation cases at the most critical rotational speeds are to be used in the following way:

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

The highest torque amplitude during a sequence of impacts is to be determined as half of the range from max to min torque and is referred to as Q_{Amax}.

An illustration of Q_{Amax} is given in Figure 4. It can be determined by

$$Q_{Amax} = \left(\frac{max(Q_r(time)) - min(Q_r(time))}{2}\right) \quad \text{[kNm]} \qquad \text{[Equation 35]}$$

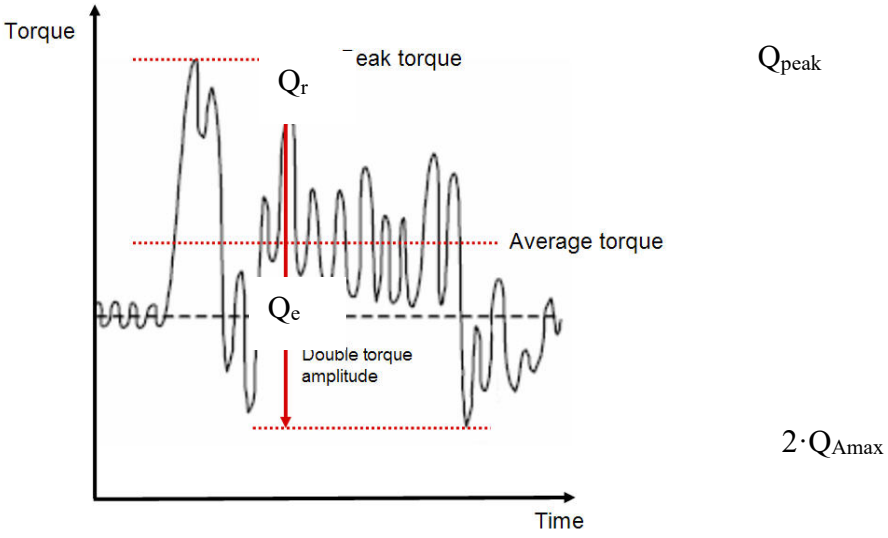


Figure 4: Interpretation of different torques in a measured curve, as example

I3 (cont)

5.6.4 Design torque along shaft line

a) If there is no relevant first order propeller torsional resonance in the range 20% (of n_n) above and 20% below the maximum operating speed in bollard condition (see Table 8), the following estimation ([Equation 36] and [Equation 37] respectively) of the maximum response torque can be used to calculate the design torque along the propeller shaft line.

$$Q_r = Q_{emax} + Q_{vib} + Q_{max} \cdot \frac{I}{I_t} \quad \text{[kNm]} \quad \text{[Equation 36]}$$

Equation 36 is to be applied for directly coupled two stroke Diesel engines without flexible coupling.

For all other plants:

$$Q_r = Q_{emax} + Q_{max} \cdot \frac{I}{I_t} \quad \text{[kNm]} \quad \text{[Equation 37]}$$

where:

- I = equivalent mass moment of inertia of all parts on engine side of component under consideration and
- I_t = equivalent mass moment of inertia of the whole propulsion system.

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

If the maximum torque, Q_{emax}, is not known, it shall be taken as follows:

Table 11: Guideline for the determination of maximum motor torque

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

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

b) If there is a first blade order torsional resonance in the range 20% (of n_n) above and 20% below the maximum operating speed (bollard condition), the design torque (Q_r) of the shaft component shall be determined by means of a dynamic torsional vibration analysis of the entire propulsion line in the time domain or alternatively in the frequency domain. It is then assumed that the plant is sufficiently designed to avoid harmful operation in barred speed range.

5.7 Guideline for torsional vibration calculation

The aim of torsional vibration calculations is to estimate the torsional loads for individual shaft line components over the life time in order to determine scantlings for safe operation. The model can be taken from the normal lumped mass elastic torsional vibration model (frequency domain) including the damping. Standard harmonics may be used to consider the gas forces. The engine torque - speed curve of the actual plant shall be applied.

I3 (cont)

For time domain analysis the model should include the ice excitation at propeller, the mean torques provided by the prime mover and the hydrodynamic mean torque produced by the propeller as well as any other relevant excitations. The calculations should cover the variation of phase between the ice excitation and prime mover excitation. This is extremely relevant for propulsion lines with direct driven combustion engines.

For frequency domain calculations the load should be estimated as a Fourier component analysis of the continuous sequence of half sine load peaks. The first and second order blade components should be used for excitation. The calculation should cover the whole relevant shaft speed range. The analysis of the responses at the relevant torsional vibration resonances may be performed for open water (without ice excitation) and ice excitation separately. The resulting maximum torque can be obtained for directly coupled plants by the following superposition:

$Q_{peak} = Q_{emax} + Q_{opw} + Q_{ice}$ [kNm] [Equation 38]

where:

Q_{emax} is the maximum engine torque at considered rotational speed

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

Q_{ice} is the calculated torque using frequency domain analysis for the relevant shaft speeds, ice excitation cases 1-4, resulting in the maximum response torque due to ice excitation

6 Design

6.1 Design Principle

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

The propulsion line components shall withstand maximum and fatigue operational loads with the relevant safety margin. The loads do not need to be considered for shaft alignment or other calculations of normal operational conditions.

6.2 Fatigue design in general

The design loads shall be based on the ice excitation and where necessary (shafting) dynamic analysis, described as a sequence of blade impacts (5.6.3.1). The shaft response torque shall be determined according 5.6.4.

The propulsion line components are to be designed so as to prevent accumulated fatigue failure when considering the relevant loads using the linear elastic Miner's rule as defined below.

$D = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \cdots + \frac{n_k}{N_k} \leq 1$ [Equation 39]

or

$D = \sum_{J=1}^{j=k} \frac{n_j}{N_j} \leq 1$ [Equation 40]

I3 (cont)

Where:

k is the number of stress levels

$N_{1...k}$ is the number of load cycles to failure of the individual stress level class

$n_{1...k}$ is the accumulated number of load cycles of the case under consideration, per class

D Miners damage sum

Guidance:

The stress distribution should be divided into a frequency load spectrum having minimum 10 stress blocks (every 10 % of the load). Calculation with 5 stress blocks has been found to be too conservative. The maximum allowable load is limited by $\sigma_{ref\ 2}$ for propeller blades and yield strength for all other components. The load distribution (spectrum) should be in accordance with the Weibull distribution.

6.3 Propeller blades

6.3.1 Calculation of blade stresses due to static loads

The blade stresses (equivalent and principal stresses) shall be calculated for the design loads given in section 5.3. Finite element analysis (FEA) shall be used for stress analysis as part of the final approval for all propeller blades. The von Mises stresses, taken as σ_{st} , shall comply with Equation 42.

Alternatively, the following simplified [Equation 41] can be used in estimating the blade stresses for all propellers in the root area (r/R < 0.5) for final approval

$$\sigma_{st} = C_1 \frac{M_{BL}}{100 \cdot ct^2} \quad \text{[MPa]} \quad \text{[Equation 41]}$$

where:
constant C_1 is the $\frac{actual\ stress}{stress\ obtained\ with\ beam\ equation}$.

If the actual value is not available, C_1 should be taken as 1.6.

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

6.3.2 Acceptability criterion for static loads

The following criterion for calculated blade stresses shall be fulfilled:

$$\frac{\sigma_{ref\ 2}}{\sigma_{st}} \geq 1.3 \quad \text{[-]} \quad \text{[Equation 42]}$$

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

I3 (cont)

6.3.3 Fatigue design of propeller blade

6.3.3.1 General

For materials with a two slope S-N curve (Figure 5) the fatigue calculations defined in this chapter are not required if the following criterion is fulfilled.

$$\sigma_{exp} \geq B_1 \cdot {\sigma_{ref\,2}}^{B_2} \cdot log(N_{ice})^{B_3}$$

[Equation 43]

where:

$B1$, $B2$ and $B3$ are coefficients for open and ducted propellers, given in the Table 12 below.

Table 12: Coefficients to check a dispense from fatigue calculation

	Open propeller	Ducted propeller
$B1$	0.00328	0.00223
$B2$	1.0076	1.0071
$B2$	2.101	2.471

Where the above criterion is not fulfilled the fatigue requirements defined below shall be applied:

The fatigue design of the propeller blade is based on an estimated load distribution for the service life of the ship and the S-N curve for the blade material. An equivalent stress σ_{fat} that produces the same fatigue damage as the expected load distribution shall be calculated according to Miner's rule and the acceptability criterion for fatigue should be fulfilled as given in this section. The equivalent stress is normalised for 100 million cycles.

The blade stresses at various selected load levels for fatigue analysis are to be taken proportional to the stresses calculated for maximum loads given in section 5.3.
The peak principal stresses σ_f and σ_b are determined from F_f and F_b using FEA. The peak stress range $\Delta\sigma_{max}$ and the maximum stress amplitude σ_{Amax} are determined on the basis of load cases 1 and 3, 2 and 4.

$$\Delta\sigma_{max} = 2 \cdot \sigma_{Amax} = \left|(\sigma_{ice})_{f\,max}\right| + \left|(\sigma_{ice})_{b\,max}\right|$$

[Equation 44]

The load spectrum for backward loads is normally expected to have a lower number of cycles than the load spectrum for forward loads. Taking this into account in a fatigue analysis introduces complications that are not justified considering all uncertainties involved.
For the calculation of equivalent stress two types of S-N curves are available.

Two slope S-N curve (slopes 4.5 and 10), see Figure 5.

One slope S-N curve (the slope can be chosen), see Figure 6.

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

I3
(cont)

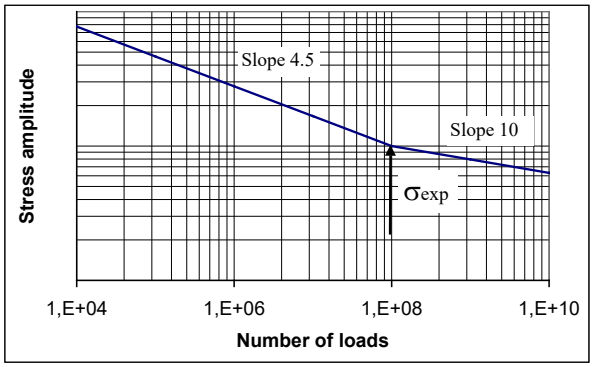


Figure 5: Two-slope S-N curve

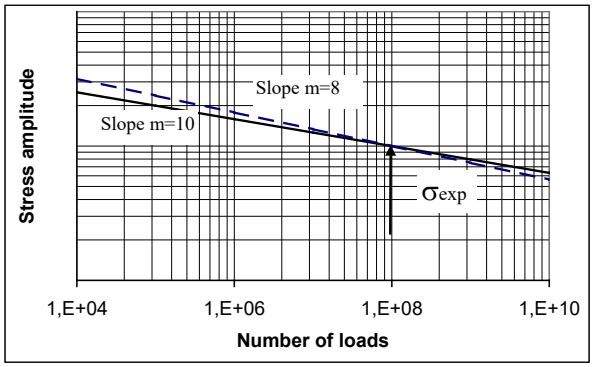


Figure 6: Constant-slope S-N curve

6.3.3.2 Equivalent fatigue stress

Note: A more general method of determining the equivalent fatigue stress of propeller blades is described in 6.5, where the principal stresses are considered according to 5.3 using the Miner's rule. For a total number of load blocks n_{bl} > 100, both methods deliver the same result. Therefore, they are regarded as equivalent.
The equivalent fatigue stress for 10^8 cycles which produces the same fatigue damage as the load distribution is:

$$\sigma_{fat} = \rho \cdot (\sigma_{ice})_{max} \qquad \text{[Equation 45]}$$

where:

$$(\sigma_{ice})_{max} = 0.5 \cdot \left((\sigma_{ice})_{f\,max} - (\sigma_{ice})_{b\,max}\right) \qquad \text{[Equation 46]}$$

$(\sigma_{ice})_{max}$ = mean value of the principal stress amplitudes resulting from design forward and backward blade forces at the location being studied.
$(\sigma_{ice})_{f\,max}$ = principal stress resulting from forward load
$(\sigma_{ice})_{b\,max}$ = principal stress resulting from backward load

In the calculation of $(\sigma_{ice})_{max}$, case 1 and case 3 or case 2 and case 4 are considered as pairs for $(\sigma_{ice})_{f\,max}$, and $(\sigma_{ice})_{b\,max}$ calculations. Case 5 is excluded from the fatigue analysis.
Calculation of parameter ρ for two-slope S-N curve

The error of the following method to determine the parameter ρ is sufficiently small, if the number of load cycles N_{ice} is in the range

$$5 \cdot 10^6 \leq N_{ice} \leq 10^8$$

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

$$\rho = C_1 \cdot (\sigma_{ice})_{max}{}^{C_2} \cdot \sigma_{fl}{}^{C_3} \cdot log(N_{ice})^{C_4} \qquad \text{[Equation 47]}$$

where:

$\sigma_{fl} = \gamma_{\varepsilon 1} \cdot \gamma_{\varepsilon 2} \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp}$ is the blade material fatigue strength at 10^8 load cycles, see 6.3.3.3.

The coefficients C_1, C_2 , C_3 , and C_4 are given in Table 13

I3
(cont)

Table 13 Coefficients to evaluate material fatigue strength

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

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

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

$$\rho = \left(G\frac{N_{ice}}{N_R}\right)^{\frac{1}{m}}\left(ln(N_{ice})\right)^{-\frac{1}{k}} \quad \text{[Equation 48]}$$

where:

k = shape parameter of the Weibull distribution

k = 1.0 for ducted propellers and

k = 0.75 for open propellers

N_R = reference number of load cycles (=10^8)

Values for the parameter G are given in Table 14 below. Linear interpolation may be used to calculate the value of G for m/k ratios other than those given in the Table 14.

Table 14: Value for the parameter G for different m/k ratios

m/k	G	m/k	G	m/k	G
3	6	6.5	1871	10	3.629×10^6
3.5	11.6	7	5040	10.5	11.899×10^6
4	24	7.5	14034	11	39.917×10^6
4.5	52.3	8	40320	11.5	136.843×10^6
5	120	8.5	119292	12	479.002×10^6
5.5	287.9	9	362880		
6	720	9.5	1.133×10^6		

6.3.3.3 Acceptability criterion for fatigue

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

$$\frac{\sigma_{fl}}{\sigma_{fat}} \geq 1.5 \quad \text{[Equation 49]}$$

where:

$$\sigma_{fl} = \gamma_{\varepsilon 1} \cdot \gamma_{\varepsilon 2} \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp} \quad \text{[Equation 50]}$$

$\gamma_{\varepsilon 1}$ = reduction factor due to scatter (equal to one standard deviation)
$\gamma_{\varepsilon 2}$ = reduction factor for test specimen size effect
γ_v = reduction factor for variable amplitude loading.
γ_m = reduction factor for mean stress.

I3 (cont)

σ_{exp} = mean fatigue strength of the blade material at 10^8 cycles to failure in seawater
σ_{exp} in Table 15 has been defined from the results of constant amplitude loading fatigue tests at 10^7 load cycles and 50% survival probability and has been extended to 10^8 load cycles.

Fatigue strength values and correction factors other than those given in Table 15 may be used, provided the values are determined under conditions approved by the classification society.

The S-N curve characteristics are based on two slopes, the first slope 4.5 is from 1000 to 10^8 load cycles; the second slope 10 is above 10^8 load cycles.

The maximum allowable stress for one or low number of cycles is limited to σ_{ref2}/S, with S=1.3 for static loads.

The fatigue strength σ_{fat} is the fatigue limit at 100 million load cycles.
The geometrical size factor ($\gamma_{\varepsilon 2}$) is:

$$\gamma_{\varepsilon 2} = 1 - a \cdot ln\left(\frac{t}{0.025}\right) \quad \text{[Equation 51]}$$

where:

"a" is as given in Table 15 below and "t" is the maximum blade thickness at the considered point

The mean stress effect (γ_m) is

$$\gamma_m = 1.0 - \left(\frac{1.4 \cdot \sigma_{mean}}{\sigma_u}\right)^{0.75} \quad \text{[Equation 52]}$$

The following values should be used for the reduction factors if actual values are not available: $\gamma_{\varepsilon 1}$ = 0.85, γ_v = 0.75, and γ_m= 0.75.

Table 15: Mean fatigue strength σ_{exp} for different material types at 10^8 load cycles and stress ratio R = -1 with a survival probability of 50%.

Mean fatigue strength σ_{exp} for different material types at 10^8 load cycles			
Bronze and brass (a=0.10)		Stainless steel (a=0.05)	
Mn-Bronze, CU1 (high tensile brass)	84 MPa	Ferritic (12Cr 1Ni)	144*) MPa
Mn-Ni-Bronze, CU2 (high tensile brass)	84 MPa	Martensitic (13Cr 4Ni/13Cr 6Ni)	156 MPa
Ni-Al-Bronze, CU3	120 MPa	Martensitic (16Cr 5Ni)	168 MPa
Mn-Al-Bronze, CU4	113 MPa	Austenitic (19Cr 10Ni)	132 MPa

*) This value may be used, provided a perfect galvanic protection is active. Otherwise a reduction of about 30 MPa shall be applied.

I3
(cont)

6.4 Blade bolts, propeller hub and CP mechanism

6.4.1 General

The blade bolts, CP mechanism, propeller boss and the fitting of the propeller to the propeller shaft shall be designed to withstand the maximum static and fatigue design loads (as applicable), as defined in 5.3 and 6.3. The safety factor S against yielding due to static loads and against fatigue shall be greater than 1.5, if not stated otherwise. The safety factor S for loads, resulting from propeller blade failure as defined in 5.4 shall be greater than 1.0 against yielding.

Provided that calculated stresses duly considering local stress concentrations are less than yield strength, or maximum of 70% of σ_u of respective materials, detailed fatigue analysis is not required. In all other cases components shall be analysed for cumulative fatigue. An approach similar to that used for shafting assessment may be applied (6.5).

6.4.2 Blade bolts

Blade bolts shall withstand the following bending moment considered around a tangent on bolt pitch circle, or any other relevant axis for non-circular joints, parallel to considered root section:

$$M_{bolt} = S \cdot F_{ex\left(0.8\frac{D}{2} - r_{bolt}\right)} \quad \text{[kNm]} \quad \text{[Equation 53]}$$

where:

r_{bolt} = radius to the bolt plane [m]

S = 1.0 safety factor

Blade bolt pre-tension shall be sufficient to avoid separation between mating surfaces when the maximum forward and backward ice loads defined in 5.3 (open and ducted propellers respectively) are applied. For conventional arrangements, the following formula may be applied:

$$d_{bb} = 41 \cdot \sqrt[2]{\frac{F_{ex} \cdot (0.8 \cdot D - d) \cdot S \cdot \alpha}{\sigma_{0.2} \cdot Z_{bb} \cdot PCD}} \quad \text{[mm]} \quad \text{[Equation 54]}$$

where:

α = 1.6 torque guided tightening
= 1.3 elongation guided
= 1.2 angle guided
= 1.1 elongated by other additional means
other factors may be used, if evidence is demonstrated

d_{bb} effective diameter of blade bolt in way of thread [mm]

Z_{bb} number of blade bolts

S = 1.0 safety factor

I3
(cont)

6.4.3 CP mechanism

Separate means, e.g. dowel pins, shall be provided in order to withstand the spindle torque resulting from blade failure Q_{sex} (5.4.2) or ice interaction Q_{smax} (5.3.7), whichever is greater. Other components of the CP mechanism shall not be damaged by the maximum spindle torques (Q_{smax}, Q_{sex}). One third of the spindle torque is assumed to be consumed by friction, if not otherwise documented trough further analysis.

The diameter of fitted pins d_{fp} between the blade and blade carrier can be calculated using the formula:

$$d_{fp} = 66 \cdot \sqrt{\frac{(Q_s - Q_{fr})}{PCD \cdot z_{pin} \cdot \sigma_{0.2}}} \quad \text{[mm]} \quad \text{[Equation 55]}$$

where:

Q_s = $max(S \cdot Q_{smax}; S \cdot Q_{sex})$ [kNm] [Equation 56]

S = 1.3 for Q_{smax} and
= 1.0 for Q_{sex}

Q_{fr} = friction between connected surfaces = 0.33·Q_s

The classification society may approve alternative Q_{fr} calculation according to reaction forces due to F_{ex}, or F_f, F_b whichever is relevant, utilising a friction coefficient = 0.15.
The stress in the actuating pin can be estimated by

$$\sigma_{vMises} = \sqrt{\left(\frac{\left(F \cdot \frac{h_{pin}}{2}\right)}{\frac{\pi \cdot {d_{pin}}^3}{32}}\right)^2 + 3 \cdot \left(\frac{F}{\frac{\pi}{4} {d_{pin}}^2}\right)^2} \quad \text{[MPa]} \quad \text{[Equation 57]}$$

where:

$$F = \frac{Q_S - Q_{fr}}{l_m} \quad \text{[kN]} \quad \text{[Equation 58]}$$

l_m distance pitching centre of blade – axis of pin [m]

h_{pin} height of actuating pin [mm]

d_{pin} diameter of actuating pin [mm]

Q_{fr} friction torque in blade bearings acting on the blade palm and caused by the reaction forces due to F_{ex}, or F_f, F_b whichever is relevant; taken to one third of spindle torque Q_s

The blade failure spindle torque Q_{sex} shall not lead to any consequential damage.
Fatigue strength is to be considered for parts transmitting the spindle torque from the blade to a servo system considering the ice spindle torque acting on one blade. The maximum amplitude $Q_{sa\max}$ is defined as:

$$Q_{sa\max} = \frac{Q_{sb} + Q_{sf}}{2} \quad \text{[kNm]} \quad \text{[Equation 59]}$$

I3 (cont)

where:

Q_{sb} spindle torque due to $|F_b|$ [kNm]

Q_{sf} spindle torque due to $|F_f|$ [kNm]

6.4.4 Servo pressure

The design pressure for the servo system shall be taken as the pressure caused by Q_{smax} or, Q_{sex} when not protected by relief valves on the hydraulic actuator side, reduced by relevant friction losses in bearings caused by the respective ice loads. The design pressure shall in any case not be less than relief valve set pressure.

6.5 Propulsion line components

The ultimate load resulting from total blade failure F_{ex} as defined in 5.4 shall consist of combined axial and bending load components, wherever this is significant. The minimum safety factor against yielding is to be 1.0 for all shaft line components.

The shafts and shafting components, such as bearings, couplings and flanges shall be designed to withstand the operational propeller/ice interaction loads as given in 5.

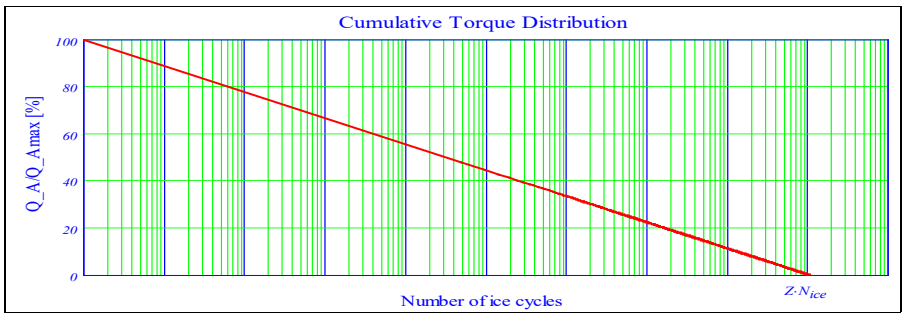
The given loads are not intended to be used for shaft alignment calculation.
Cumulative fatigue calculations shall be conducted according to the Miner's rule. A fatigue calculation is not necessary, if the maximum stress is below fatigue strength at 10^8 load cycles.

The torque and thrust amplitude distribution (spectrum) in the propulsion line is to be taken as (because Weibull exponent k = 1):

$$Q_A(N) = Q_{Amax} \cdot \left(1 - \frac{log(N)}{log(Z \cdot N_{ice})}\right) \qquad \text{[Equation 60]}$$

This is illustrated by the example in the Figure 7.

Figure7: Cumulative torque distribution



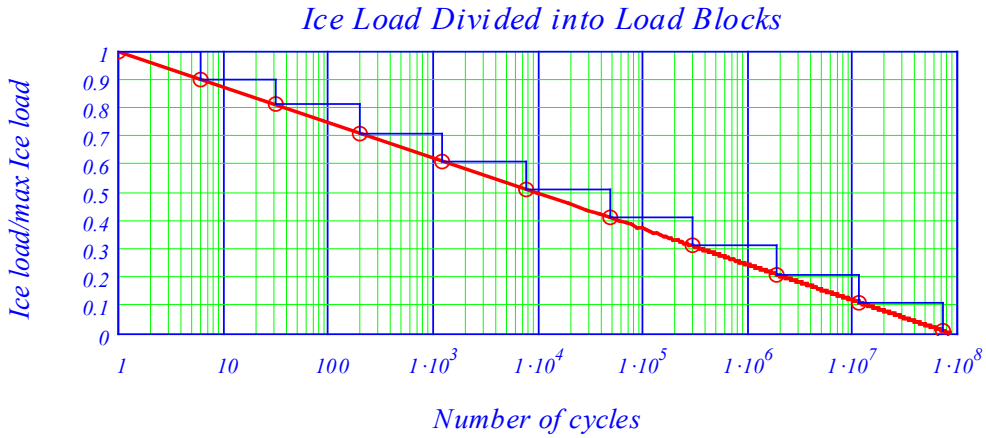
The number of load cycles in the load spectrum is defined as $Z \cdot N_{ice}$.

The Weibull exponent should be considered as k = 1.0 for both open and ducted propeller torque and bending forces. The load distribution is an accumulated load spectrum, and the load spectrum should be divided into a minimum of ten load blocks when using the Miner summation method.

I3 (cont)

The load spectrum used counts the number of cycles for 100% load to be the number of cycles above the next step, e.g. 90 % load. This ensures that the calculation is on the conservative side. Consequently, the fewer stress blocks used the more conservative the calculated safety margin.

Figure 8: Example of ice load distribution (spectrum) for the shafting (k = 1)



The load spectrum is divided into n_{bl}-number of load blocks for the Miner summation method.

The following formula can be used for calculation of the number of cycles for each load block.

$$n_i = N_{ice}^{\,1-\left(1-\frac{i}{n_{bl}}\right)^k} - \sum_{i=1}^{i} n_{i-1} \qquad \text{[Equation 61]}$$

where:

i = single load block i and n_{bl} is the number of load blocks

6.5.1 Propeller fitting to the shaft

6.5.1.1 Keyless cone mounting

The friction capacity (at 0° C) shall be at least S = 2.0 times the highest peak torque Q_{peak} as determined in 5.6 without exceeding the permissible hub stresses.

The necessary surface pressure $P_{0°C}$ can be determined as:

$$P_{0°C} = \frac{2 \cdot S \cdot Q_{peak}}{\pi \cdot \mu \cdot {D_S}^2 \cdot L \cdot 10^3} \quad \text{[MPa]} \qquad \text{[Equation 62]}$$

where:

μ = 0.15 for steel-steel,
= 0.13 for steel-bronze

D_S = is the shrinkage diameter at the mid-length of the taper [m]

L = is the effective length of taper [m]

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

I3
(cont)

6.5.1.2 Key mounting

Key mounting is not permitted.

6.5.1.3 Flange mounting

The flange thickness is to be at least 25% of the required aft end shaft diameter (IACS UR M34).

Any additional stress raisers such as recesses for bolt heads shall not interfere with the flange fillet unless the flange thickness is increased correspondingly.

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

The diameter of shear pins shall be calculated according to the following equation:

$$d_{pin} = 66 \cdot \sqrt[2]{\frac{Q_{peak} \cdot S}{PCD \cdot z_{pin} \cdot \sigma_{0.2}}} \quad \text{[mm]} \qquad \text{[Equation 63]}$$

where

z_{pin} = number of shear pins

S = 1.3 safety factor

The bolts are to be designed so that the blade failure load F_{ex} (5.4) in backward direction does not cause yielding of the bolts. The following equation should be applied:

$$d_b = 41 \cdot \sqrt{\frac{F_{ex} \cdot \left(0.8 \cdot \frac{D}{PCD} + 1\right) \cdot \alpha}{\sigma_{0.2} \cdot z_b}} \quad \text{[mm]} \qquad \text{[Equation 64]}$$

where:

α = 1.6 torque guided tightening
= 1.3 elongation guided
= 1.2 angle guided
= 1.1 elongated by other additional means
other factors may be used, if evidence is demonstrated

d_b diameter flange bolt [mm]

z_b number of flange bolts

I3
(cont)

6.5.2 Propeller shaft

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

6.5.2.1 The blade failure load F_{ex} (5.4) applied parallel to the shaft (forward or backwards) shall not cause yielding. The bending moment need not to be combined with any other loads. The diameter d_p in way of the aft stern tube bearing shall not be less than:

$$d_p = 160 \cdot \sqrt[3]{\frac{F_{ex} \cdot D}{\sigma_{0.2} \cdot \left(1 - \frac{{d_i}^4}{{d_p}^4}\right)}} \quad \text{[mm]} \qquad \text{[Equation 65]}$$

where:

d_p = propeller shaft diameter [mm]

d_i = propeller shaft inner diameter [mm]

Forward from the aft stern tube bearing the shaft diameter may be reduced based on direct calculation of the actual bending moment, or by the assumption that the bending moment caused by F_{ex} is linearly reduced to 25% at the next bearing and in front of this linearly to zero at third bearing.

Bending due to maximum blade forces F_b and F_f have been disregarded since the resulting stress levels are much lower than the stresses caused by the blade failure load.

6.5.2.2 The stresses due to the peak torque Q_{peak} shall have a minimum safety factor of S=1.5 against yielding in plain sections and S=1.0 in way of stress concentrations in order to avoid bent shafts.

Minimum diameter of:

plain shaft:

$$d_p = 210 \cdot \sqrt[3]{\frac{Q_{peak} \cdot S}{\sigma_{0.2} \cdot \left(1 - \frac{{d_i}^4}{d^4}\right)}} \quad \text{[mm]} \qquad \text{[Equation 66]}$$

notched shaft:

$$d_p = 210 \cdot \sqrt[3]{\frac{Q_{peak} \cdot S \cdot \alpha_t}{\sigma_{0.2} \cdot \left(1 - \frac{{d_i}^4}{d^4}\right)}} \quad \text{[mm]} \qquad \text{[Equation 67]}$$

where:

α_t = local stress concentration factor in torsion.

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

6.5.2.3 The torque amplitudes (5.6.4) with the corresponding number of load cycles shall be used in an accumulated fatigue evaluation where the safety factor is S_{fat}=1.5. If the plant has high engine excited torsional vibrations (e.g. direct coupled 2-stroke engines), this shall also be considered.

I3 (cont)

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

$$\sigma_F = 0.436 \cdot \sigma_{0.2} + 77 = \tau_F \cdot \sqrt{3} \quad \text{[MPa]} \quad \text{[Equation 68]}$$

This is valid for small polished specimens (no notch) and reversed stresses, see "VDEH 1983 Bericht Nr. ABF11 Berechnung von Wöhlerlinien für Bauteile aus Stahl".

The high cycle fatigue (HCF) is to be assessed based on the above fatigue strengths, notch factors (i.e. geometrical stress concentration factors and notch sensitivity), size factors, mean stress influence and the required safety factor of 1.6 at 3 million cycles increasing to 1.8 at 10^9 cycles.

The low cycle fatigue (LCF) representing 10^4 cycles is to be based on the smaller value of yield or 0.7 of tensile strength/$\sqrt{3}$. The criterion utilises a safety factor of 1.25.

The LCF and HCF as given above represent the upper and lower knees in a stress-cycle diagram. Since the required safety factors are included in these values, a Miner sum of unity is acceptable.

6.5.3 Intermediate shafts

The intermediate shafts are to be designed to fulfil 6.5.2.2 to 6.5.2.4.

6.5.4 Shaft connections

6.5.4.1 Shrink fit couplings (keyless)

See 6.5.1.1. A safety factor of S = 1.8 shall be applied.

6.5.4.2 Key mounting

Key mounting is not permitted.

6.5.4.3 Flange mounting

The flange thickness is to be at least 20% of the required shaft diameter (IACS UR M34).

Any additional stress raisers such as recesses for bolt heads shall not interfere with the flange fillet unless the flange thickness is increased correspondingly.

The flange fillet radius is to be at least 8% of the shaft diameter (IACS UR M34).

The diameter of ream fitted (light press fit) bolts shall be chosen so that the peak torque is transmitted with a safety factor of 1.9. This accounts for a prestress. Pins shall transmit the peak torque with a safety factor of 1.5 against yielding ([Equation 63]).

The bolts are to be designed so that the blade failure load (5.4) in backward direction does not cause yielding.

I3
(cont)

6.5.4.4 Splined shaft connections

Splined shaft connections can be applied where no axial or bending loads occur. A safety factor of S = 1.5 against allowable contact and shear stress resulting from Q_{peak} shall be applied.

6.5.5 Gear transmissions

6.5.5.1 Shafts

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

6.5.5.2 Gearing

The gearing shall fulfil following three acceptance criteria:

- Tooth root stresses

- Pitting of flanks

- Scuffing

In addition to above 3 criteria subsurface fatigue may need to be considered.

Common for all criteria is the influence of load distribution over the face width. All relevant parameters are to be considered, such as elastic deflections (of mesh, shafts and gear bodies), accuracy tolerances, helix modifications, and working positions in bearings (especially for multiple input single output gears).

The load spectrum (see 6.5) may be applied in such a way that the numbers of load cycles for the output wheel are multiplied by a factor of (number of pinions on the wheel / number of propeller blades Z). For pinions and wheels operating at higher speeds the numbers of load cycles are found by multiplication with the gear ratios. The peak torque (Q_{peak}) is also to be considered during calculations.

Cylindrical gears can be assessed on the basis of the international standard ISO 6336 series (i.e. ISO 6336-1:2019, ISO 6336-2:2019, ISO 6336-3:2019, ISO 6336-4:2019, ISO 6336-5:2016 and ISO 6336-6:2019), provided that “method B” is used. Standards within the classification societies can also be applied provided that they are considered equivalent to the above mentioned ISO 6336.

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

Tooth root safety shall be assessed against the peak torque, torque amplitudes (with the pertinent average torque) as well as the ordinary loads (open water free running) by means of accumulated fatigue analyses. The resulting factor of safety is to be at least 1.5. (Ref ISO 6336 Pt 1, 3 and 6 and IACS UR M56)

The safety against pitting shall be assessed in the same way as tooth root stresses, but with a minimum resulting safety factor of 1.2. (Ref ISO 6336-1:2019, ISO 6336-2:2019 and ISO 6336-6:2019 as well as IACS UR M56).

I3 (cont)

The scuffing safety (flash temperature method – ref. ISO/TR 13989-1:2000 and ISO/TR 13989-2:2000) based on the peak torque shall be at least 1.2 when the FZG class of the oil is assumed one stage below specification.

The safety against subsurface fatigue of flanks for surface hardened gears (oblique fracture from active flank to opposite root) is to be assessed at the discretion of each Classification Society. (It should be noted that high overloads can initiate subsurface fatigue cracks that may lead to a premature failure. In lieu of analyses UT inspection intervals may be used.)

6.5.5.3 Bearings

See section 6.5.9.

6.5.5.4 Gear wheel shaft connections

The torque capacity shall be at least 1.8 times the highest peak torque Q_{peak} (at considered rotational speed) as determined in 6.5 without exceeding the permissible hub stresses of 80% yield.

6.5.6 Clutches

Clutches shall have a static friction torque of at least 1.3 times the peak torque Q_{peak} and dynamic friction torque 2/3 of the static.
Emergency operation of clutch after failure of e.g. operating pressure shall be made possible within reasonably short time. If this is arranged by bolts, it shall be on the engine side of the clutch in order to ensure access to all bolts by turning the engine.

6.5.7 Elastic couplings

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

$Q_{peak} < 0.8 \cdot T_{kmax}\ (N = 1)$ [kNm] [Equation 69]

There shall be a separation margin of at least 20% between the maximum response torque Q_{peak} (see Figure 4) and the torque where any mechanical twist limitation and/or the permissible maximum torque of the elastic coupling, valid for at least a single load cycle (N=1), is reached.

A sufficient fatigue strength shall be demonstrated at design torque level $Q_r(N = x)$ and $Q_A(N = x)$. This may be demonstrated by interpolation in a Weibull torque distribution (similar to Figure 7):

$\frac{Q_r(N=x)}{Q_r(N=1)} = 1 - \frac{\log(x)}{\log(Z \cdot N_{ice})}$ [-] [Equation 70]

respectively

$\frac{Q_A(N=x)}{Q_A(N=1)} = 1 - \frac{\log(x)}{\log(Z \cdot N_{ice})}$ [-] [Equation 71]

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

Q_r (N=5E4)$\cdot S < T_{Kmax}$ (N=5E4) [kNm] [Equation 72]
Q_r (N=1E6)$\cdot S < T_{KV}$ [kNm] [Equation 73]

I3 (cont)

Q_A (N=5E4)·S < ΔT_{max} (N=5E4) [kNm] [Equation 74]

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

See illustration in below Figure 9, Figure 10 and Figure 11.

The torque amplitude (or range Δ) shall not lead to fatigue cracking, i.e. exceeding the permissible vibratory torque. The permissible torque may be determined by interpolation in a Weibull torque distribution where T_{Kmax1} respectively ΔT_{Kmax} refer to 50000 cycles and T_{KV} refer to 10^6 cycles. See illustration in below Figure 9, Figure 10 and Figure 11.

$T_{Kmax1} \geq Q_r \; at \; 5 \cdot 10^4$ load cycles [kNm] [Equation 75]

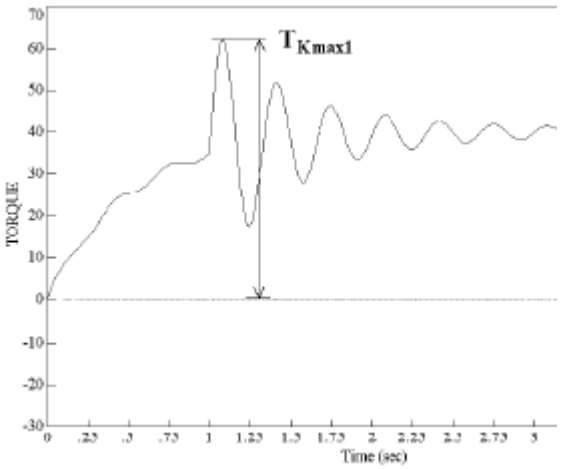


Figure 9

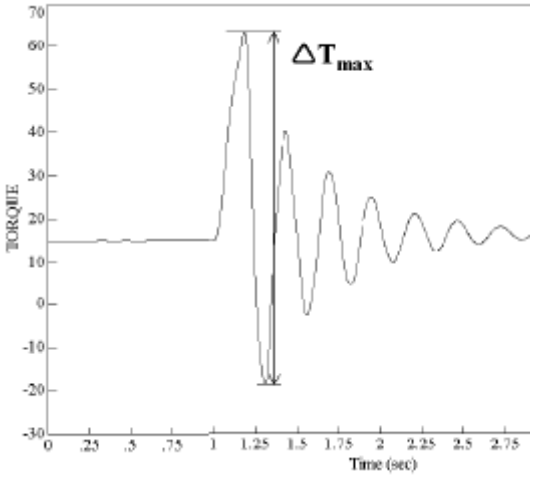


Figure 10

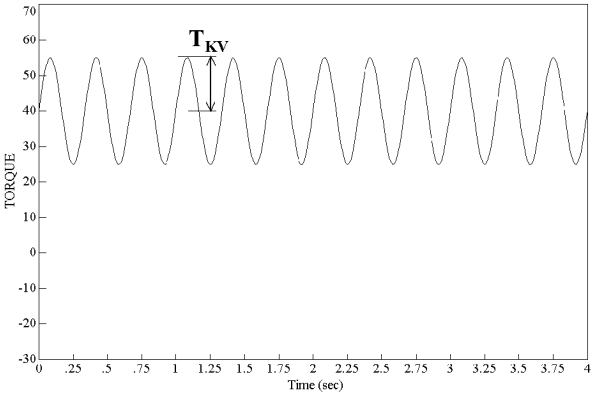


Figure 11

6.5.8 Crankshafts

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

6.5.9 Bearings

The aft stern tube bearing as well as the next shaft line bearing are to withstand F_{ex} as given in 5.4, in such a way that the ship can maintain operational capability. Rolling bearings are to have an L_{10a} lifetime of at least 40 000 hours according to ISO 281:2007. Thrust bearings

I3 (cont)

and their housings are to be designed to withstand with a safety factor S = 1.0 the maximum response thrust 5.5 and the axial force resulting from the blade failure load F_{ex} in 5.4. For the purpose of calculation, except for F_{ex}, the shafts are assumed to rotate at rated speed. For pulling propellers special consideration is to be given to loads from ice interaction on the propeller hub.

6.5.10 Seals

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

Seals installed are to be suitable for the intended application. The manufacturer is to provide service experience in similar applications and/or testing results for consideration.

6.6 Azimuthing main propulsors

In addition to the above requirements, special consideration shall be given to those loading cases which are extraordinary for propulsion units when compared with conventional propellers. The estimation of load cases shall reflect the way the thrusters are intended to operate on the specific ship. In this respect, for example, the loads caused by the impacts of ice blocks on the propeller hub of a pulling propeller shall be considered. Furthermore, loads resulting from the thrusters operating at an oblique angle to the flow shall be considered. The steering mechanism, the fitting of the unit, and the body of the thruster shall be designed to withstand the loss of a blade without damage. The loss of a blade shall be considered for the propeller blade orientation which causes the maximum load on the component being studied. Typically, top-down blade orientation places the maximum bending loads on the thruster body.

Azimuth thrusters shall also be designed for estimated loads caused by thruster body/ice interaction. The thruster body shall withstand the loads obtained when the maximum ice blocks, which are given in section 5.2, strike the thruster body when the ship is at a typical ice operating speed. In addition, the design situation in which an ice sheet glides along the ship's hull and presses against the thruster body should be considered. The thickness of the sheet should be taken as the thickness of the maximum ice block entering the propeller, as defined in section 5.2.

7 Prime Movers

7.1 Propulsion engines

Engines are to be capable of being started and running the propeller in bollard condition.

Propulsion plants with CP propeller are to be capable being operated even when the CP system is at full pitch as limited by mechanical stoppers.

7.2 Starting arrangements

The capacity of the air receivers shall be sufficient to provide, without recharging, not less than 12 consecutive starts of the propulsion engine, if this has to be reversed for going astern or 6 consecutive starts if the propulsion engine does not have to be reversed for going astern.

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

I3 (cont)

The capacity of the air compressors shall be sufficient for charging the air receivers from atmospheric to full pressure in one (1) hour, except for a ship with the ice class PC6 to PC1, if its propulsion engine has to be reversed for going astern, in which case the compressor shall be able to charge the receivers in half an hour.

7.3 Emergency power units

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

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

8 Equipment fastening loading accelerations

8.1 General

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

8.2 Longitudinal Impact Accelerations, a_1

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

$$a_1 = \frac{F_{IB}}{\Delta} \cdot \left(1.1 \cdot \tan(\gamma + \varphi) + \left(7 \cdot \frac{H}{L}\right)\right) \qquad [m/s^2] \qquad [Equation 76]$$

8.3 Vertical acceleration, a_v

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

$$a_v = 2.5 \cdot \left(\frac{F_{IB}}{\Delta}\right) \cdot F_X \qquad [m/s^2] \qquad [Equation 77]$$

F_X = 1.3 at FP

= 0.2 at midships

= 0.4 at AP

= 1.3 at AP for vessels conducting ice breaking astern

Intermediate values to be interpolated linearly.

8.4 Transverse impact acceleration, a_t

Combined transverse impact acceleration at any point along hull girder,

$$a_t = 3F_i \frac{F_X}{\Delta} \qquad [m/s^2] \qquad [Equation 78]$$

F_X = 1.5 at FP

= 0.25 at midships

I3 (cont)

= 0.5 at AP

= 1.5 at AP for vessels conducting ice breaking astern

Intermediate values to be interpolated linearly.

where:

φ = maximum friction angle between steel and ice, normally taken as 10 [degrees]

γ = bow stem angle at waterline [degrees]

Δ = displacement

L = length between perpendiculars [m]

H = distance in meters from the water line to the point being considered [m]

F_{IB} = vertical impact force, defined in UR I2.13.2.1

F_i = total force normal to shell plating in the bow area due to oblique ice impact, defined in UR I2.3.2.1

9 Auxiliary Systems

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

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

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

10 Sea Inlets and Cooling Water Systems

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

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

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

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

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

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

I3 (cont)

10.7 Efficient means are to be provided to re-circulate cooling seawater to the ice box. Total sectional area of the circulating pipes is not to be less than the area of the cooling water discharge pipe.

10.8 Detachable gratings or manholes are to be provided for ice boxes. Manholes are to be located above the deepest load line. Access is to be provided to the ice box from above.

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

11 Ballast Tanks

11.1 Efficient means are to be provided to prevent freezing in fore and after peak tanks and wing tanks located above the water line and where otherwise found necessary.

12 Ventilation Systems

12.1 The air intakes for machinery and accommodation ventilation are to be located on both sides of the ship at locations where manual de-icing is possible. Anti-icing protection of the air inlets may be accepted as an equivalent solution to location on both sides of the ship and manual de-icing at the Society's discretion. Notwithstanding the above, multiple air intakes are to be provided for the emergency generating set and are to be as far apart as possible.

12.2 The temperature of the inlet air shall be suitable for:

- the safe operation of the machinery; and
- the thermal comfort in the accommodation.

Accommodation and ventilation air intakes shall be provided with means of heating, if needed.

13 Steering Systems

13.1 Rudder stops are to be provided. The design ice force on rudder shall be transmitted to the rudder stops without damage to the steering system.

An ice knife shall in general be fitted to protect the rudder in centre position. The ice knife shall extend below BWL. Design forces shall be determined according to the I2.15.

13.2 The rudder actuator is to comply with the following requirements 13.2.1 and 13.2.2:

13.2.1 The rudder actuator is to be designed for a holding torque obtained by multiplying the open water torque resulting from the application of SOLAS Reg. II-1 /29.3.2 (considering however a maximum speed of 18 knots, by following factors:

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

I3 (cont)

13.2.2 The design pressure for calculations to determine the scantlings of the rudder actuator is to be at least 1.25 times the maximum working pressure corresponding to the holding torque defined in 13.2.1 (Derived from SOLAS Reg. II-1 / 29.2.2).

13.3 The rudder actuator is to be protected by torque relief arrangements, assuming the following turning speeds [deg/s] without an undue pressure rise (ref UR M42 for undue pressure rise):

Table 17: Steering gear turning speeds

Ice Class	PC1 and PC2	PC3 to PC5	PC6 and PC7
Turning speeds [deg/s]	10	7.5	6

If the rudder and actuator design can withstand such rapid loads, this special relief arrangement is not necessary and a conventional one may be used instead (UR M42).

13.4 Additionally for icebreakers, fast-acting torque relief arrangements are to be fitted in order to provide effective protection of the rudder actuator in case of the rudder being pushed rapidly hard over against the stops.

For hydraulically operated steering gear, the fast-acting torque relief arrangement is to be so designed that the pressure cannot exceed 115% of the set pressure of the safety valves when the rudder is being forced to move at the speed indicated in Table 18, also when taking into account the oil viscosity at the lowest expected ambient temperature in the steering gear compartment.

For alternative steering systems the fast-acting torque relief arrangement is to demonstrate an equivalent degree of protection to that required for hydraulically operated arrangements.

The turning speeds to be assumed for each ice class are shown in Table 18 below.

Table 18: Steering gear turning speeds for icebreakers

Ice Class	PC1 and PC2	PC3 to PC5	PC6 and PC7
Turning speeds [deg/s]	40	20	15

The arrangement is to be designed such that steering capacity can be speedily regained.

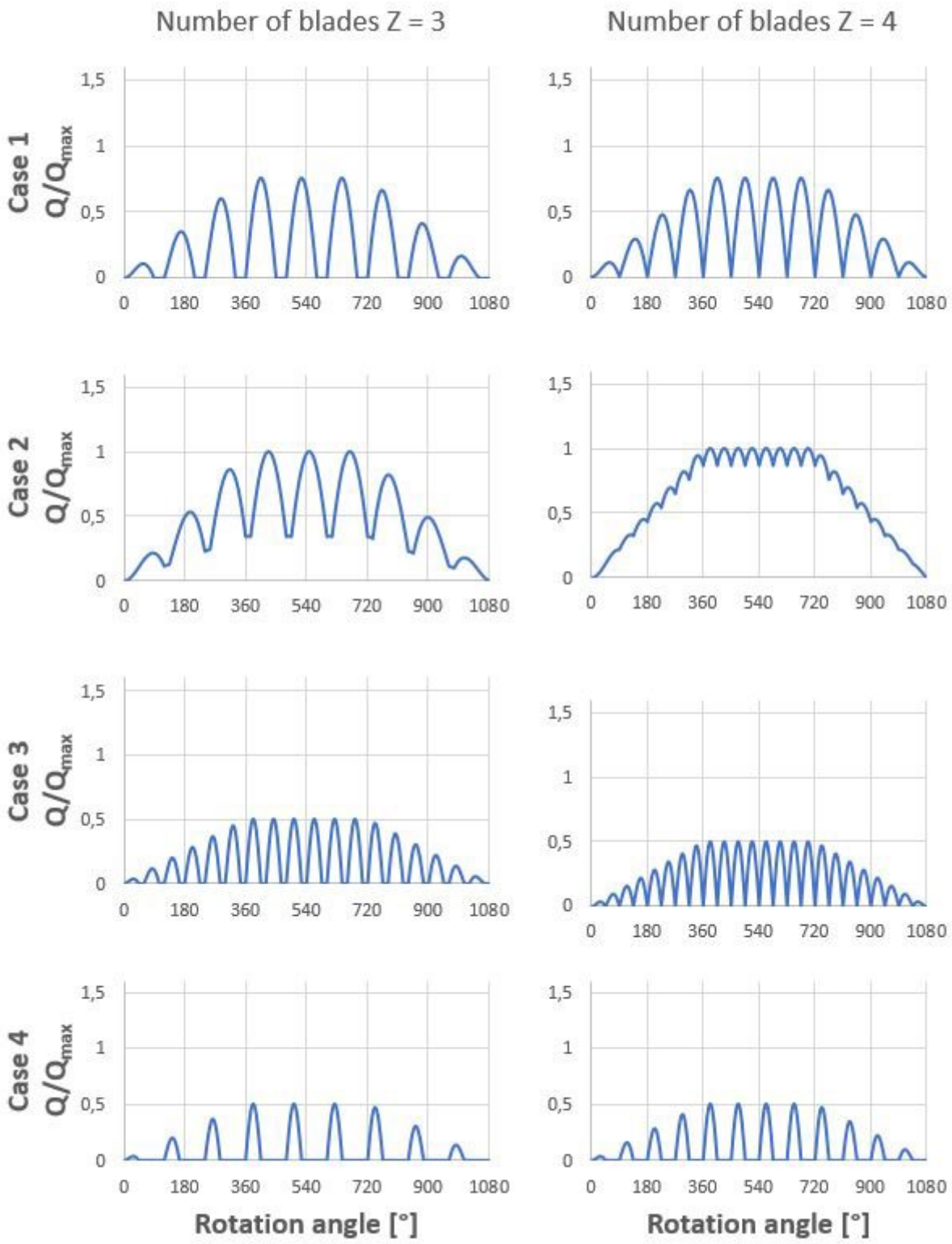
14 Alternative Design

14.1 As an alternative to UR I3 – a comprehensive design study may be submitted and may be requested to be validated by an agreed test programme.

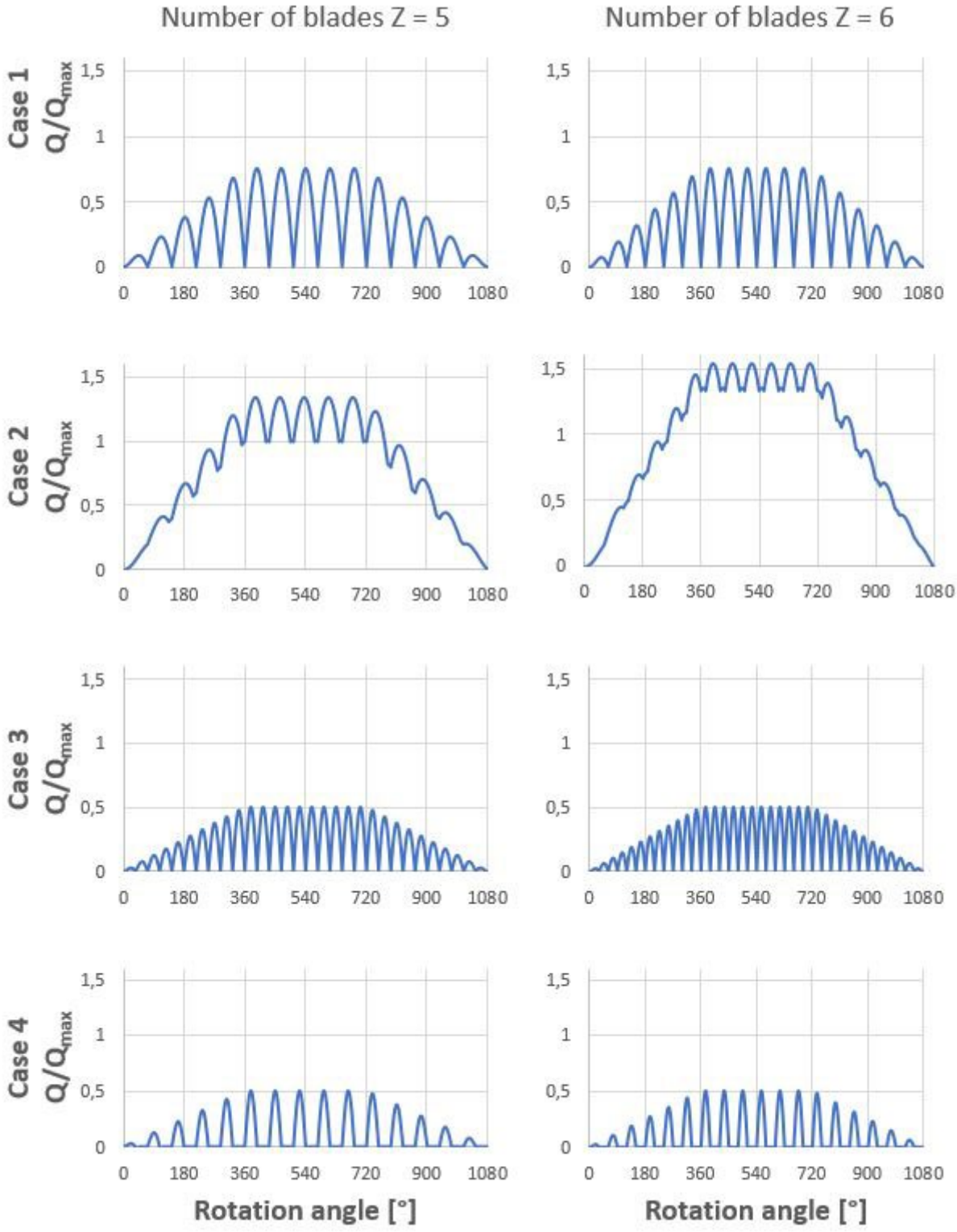
APPENDIX

I3
(cont)

The following illustrations show the excitation torque for all torsional load cases given in this UR for different blade numbers (Z). The plots have been made using data for PC7 (H_{ice} = 1.5)



I3
(cont)



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