RULES FOR BUILDING AND CLASSING

STEEL VESSELS
2019

PART 6
SPECIALIZED ITEMS AND SYSTEMS

(Updated July 2019)

American Bureau of Shipping
Incorporated by Act of Legislature of
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PART

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Specialized Items and Systems *(1 July 2017)*

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CHAPTER 1 Strengthening for Navigation in Ice

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CHAPTER 1 Strengthening for Navigation in Ice

Foreword *(2012)*

This Chapter provides requirements for optional ice strengthening classes. Sections 6-1-1 through 6-1-3 contain Polar Class ice strengthening requirements, based on IACS UR I1, I2, and I3, for vessels intended for operation in ice-covered Polar waters. Section 6-1-4 contains requirements for Enhanced Polar Class notation. Section 6-1-5 contains general ice strengthening requirements for vessels intended for navigation in first-year ice. Section 6-1-6 represents *1985 Finnish Swedish Ice Class Rules*, as amended.

The requirements in this Section are applicable to vessels of any length and are in addition to those in other Sections of these Rules or *Rules for Building and Classing Steel Vessels Under 90 Meters (295 Feet) in Length*, as appropriate.

Vessels intended for navigation in the Canadian Arctic are to comply with the requirements of the *Canadian Arctic Shipping Pollution Prevention Regulations*. ABS can issue an Arctic Pollution Prevention Certificate when authorized by the Canadian flag administration.

It is the responsibility of the owner to determine which ice class is most suitable for the intended service.

With publication of this edition of the Rules, the IACS Unified Requirements for Polar Class Ships (April 2016) are incorporated into Sections 6-1-1 through 6-1-3, replacing the requirements of the previous ABS general ice classes: **A5** through **A1**. Where the IACS requirements do not have a requirement comparable to an existing ABS requirement, the ABS requirement has been retained in Section 6-1-4 for an optional **Enhanced** notation.

With these changes, the ice strengthening requirements for general ice classes **A0**, **B0**, **C0**, **D0**, and **E0** and for non-self-propelled vessels have been retained in Section 6-1-5.

The attention of designers, owners and operators is directed to the optional ABS *Guide for Vessels Operating in Low Temperature Environments* for considerations not covered in this Chapter. Further, IMO statutory instruments having requirements specific to cold regions operations also is to be considered.

CHAPTER 1 Strengthening for Navigation in Ice

SECTION 1 Introduction *(2012)*

1 General

The ice classes are as follows in 6-1-1/Table 1.

TABLE 1
Ice Class Notations[2] *(1 July 2017)*

Polar Class (6-1-1, 6-1-2, 6-1-3)	*Polar Class (6-1-1, 6-1-2, 6-1-3)*	*Polar Class, Enhanced (6-1-4)*	*First-year Ice Class (6-1-5)*	*Baltic Class (6-1-6)*
PC1	**Ice Breaker, PC1**	**PC1, Enhanced**		
PC2	**Ice Breaker, PC2**	**PC2, Enhanced**		
PC3	**Ice Breaker, PC3**	**PC3, Enhanced**		
PC4	**Ice Breaker, PC4**	**PC4, Enhanced**		
PC5	**Ice Breaker, PC5**	**PC5, Enhanced**		
PC6	**Ice Breaker, PC6**	**PC6, Enhanced**		**I AA**
PC7	**Ice Breaker, PC7**	**PC7, Enhanced**	**A0**	**I A**
			B0	**I B**
			C0	**I C**
			D0	
			E0	

Notes:

1 The shaded ice classes are eligible for **Ice Breaker** class notation.

2 *(1 July 2017)* This table shows the approximate correspondence between different ABS ice class notations. It is not to be interpreted to imply direct equivalencies between ice classes.

1.1 Application – Polar Class *(1 July 2017)*

i) The requirements for Polar Class Vessels apply to vessels constructed of steel and intended for independent navigation in ice-infested polar waters.

ii) Vessels that comply with the requirements of this Section, 6-1-1 and 6-1-2 can be considered for a Polar Class notation as listed in 6-1-1/Table 2. The requirements of these Sections are in addition to the open water requirements of the Rules. 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.

iii) Vessels requiring ice breaker assistance are to comply with the additional requirements in 6-1-4/17.1.

iv) Provided all Polar Class Vessel requirements as specified in this Chapter are met, the vessels will be distinguished in the Record by **Ice Class** followed by ice class **PC 7** through **PC 1**, as applicable.

v) Vessels which are assigned a Polar Class notation and complying with the relevant requirements of Sections 6-1-1, 6-1-2 and 6-1-3 may be given the additional notation **Ice Breaker**. **Ice Breaker** refers to any vessel 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. These vessels are to be distinguished in the *Record* by the notation **Ice Breaker** followed by an appropriate Ice Class notation in 6-1-1/3.1 (e.g., ✠ **A1, Ice Breaker, Ice Class PC3**)

vi) For vessels 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 6-1-1/Table 2 for the corresponding Polar Class. For vessels and vessel-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.

vii) For vessels which are assigned a Polar Class notation **PC1** through **PC5**, bows with vertical sides, and bulbous bows are generally to be avoided. Bow angles should in general be within the range specified in 6-1-2/5.1v).

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

3 Description of Polar Class

3.1 Selection of Polar Classes

The Polar Class (PC) notations and descriptions are given in 6-1-1/Table 2. It is the responsibility of the Owner to select an appropriate Polar Class. The descriptions in 6-1-1/Table 2 are intended to guide owners, designers and administrations in selecting an appropriate Polar Class to match the requirements for the vessel with its intended voyage or service.

The Polar Class notations are used throughout this Chapter to convey the differences between classes with respect to operational capability and strength.

TABLE 2
Polar Class Descriptions *(2012)*

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

** Note:* "Old ice" means 2nd year ice or multi-year ice.

5 Definitions

5.1 Ice Belt

The ice belt is that reinforced portion of the shell and hull appendages that overlaps the upper and lower ice waterlines and is subject to the design ice loads. The required ice belt overlap extends from 1.5 m below the lower ice waterline to 1.0 m or 1.5 m above the upper ice waterline, depending upon Polar Class. In the bow area, the overlap increases linearly to 2.0 m above the upper ice waterline at the stem. See 6-1-2/Figure 1.

5.3 Upper and Lower Ice Waterlines *(1 July 2017)*

The upper and lower ice waterlines upon which the design of the vessel has been based are to be indicated on the Certificate of Classification. The upper ice waterline (UIWL) is to be defined by the maximum drafts fore, amidships and aft. The lower ice waterline (LIWL) is to be defined by the minimum drafts fore, amidships and aft.

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

5.5 Displacement

The displacement, D, is the molded displacement in metric tons (long tons) at the upper ice waterline.

5.7 Length

The vessel's length, L, is as defined in 3-1-1/3.1, but measured on the upper ice waterline, in m (ft).

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CHAPTER **1 Strengthening for Navigation in Ice**

SECTION **2 Structural Requirements for Polar Class Vessels** *(2012)*

1 Application

These requirements apply to Polar Class vessels according to 6-1-1/1.1.

3 Hull Areas

3.1 General

i) The hull of all Polar Class vessels 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 6-1-2/Figure 1.

ii) The upper ice waterline (UIWL) and lower ice waterline (LIWL) are as defined in 6-1-1/5.3.

iii) 6-1-2/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 vessel baseline.

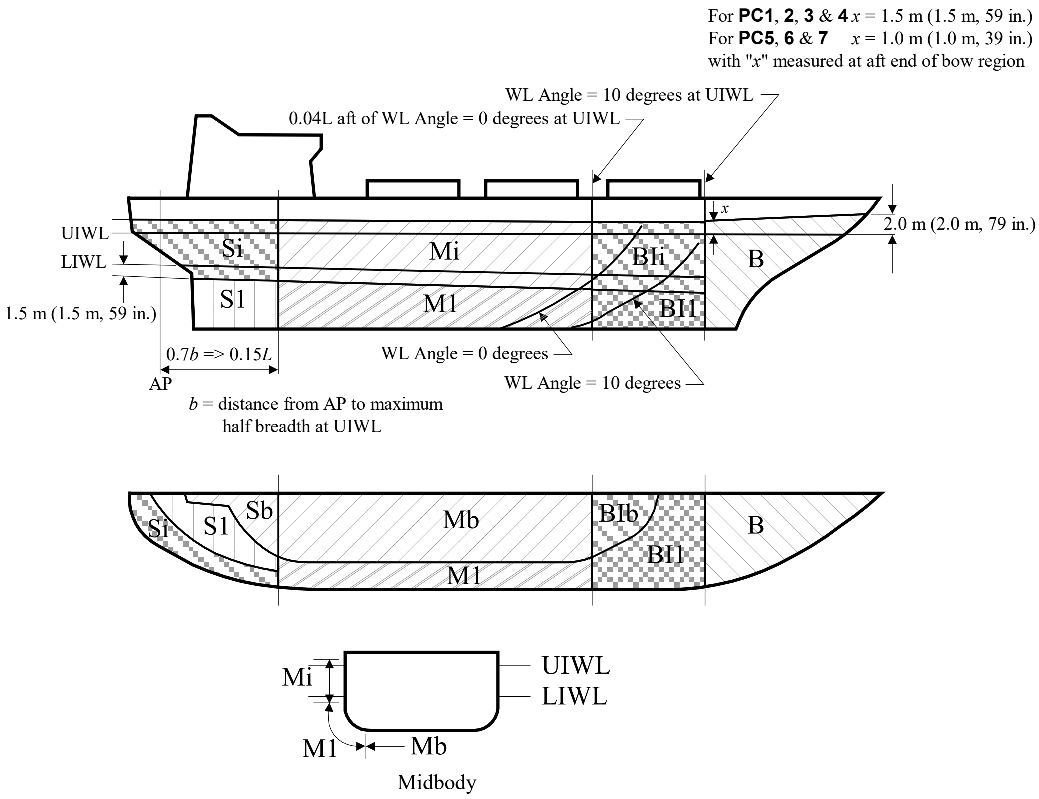
iv) 6-1-2/Figure 1 notwithstanding, the aft boundary of the Bow region need not be more than $0.45L$ aft of the forward perpendicular (FP).

v) The boundary between the bottom and lower regions is to be taken at the point where the shell tangent is inclined 7 degrees from horizontal.

vi) If a vessel is intended to operate astern in ice regions, the aft section of the vessel is to be designed using the Bow and Bow Intermediate hull area requirements as prescribed in 6-1-2/3.1vii).

vii) *(1 July 2017)* 6-1-2/Figure 1 notwithstanding, if the vessel is assigned the additional notation **Ice Breaker**, the forward boundary of the stern region is to be at least $0.04L$ forward of the section where the parallel ship side at the upper ice waterline (UIWL) ends.

FIGURE 1
Hull Area Extents *(2012)*



5 Design Ice Loads

5.1 General *(1 July 2017)*

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 vessels, 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 6-1-2/5.5iii) are applicable for bow forms where the buttock angle γ at the stem is positive and less than 80°, and the normal frame angle β' at the center of the foremost sub-region, as defined in 6-1-2/5.5i), is greater than 10°.

vi) Design ice forces calculated according to 6-1-2/5.5iv) 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 6-1-2/5.5i), are between 0° and 10°.

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 6-1-2/5.5iv). In addition, the design forces are not to be taken less than those given in 6-1-2/5.5iii), assuming fa = 0.6 and AR = 1.3.

viii) For ships with bow forms other than those defined in 6-1-2/5.1v) to 6-1-2/5.1vii), design forces are to be specially considered.

ix) Vessel 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 the maximum accelerations as given in 6-1-3/13, are to be considered in the design of these structures.

5.3 Glancing Impact Load Characteristics *(1 July 2017)*

The parameters defining the glancing impact load characteristics are reflected in the Class Factors listed in 6-1-2/Table 1 and 6-1-2/Table 2.

TABLE 1
Class Factors to be Used in 6-1-2/5.5iii) *(1 July 2017)*

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 (1804 ,1794)	68.60 (6995 ,6885)	2.01 (0.122 ,0.308)	250 (250 ,246)	7.46 (753,473)
PC2	9.89 (1009 ,1003)	46.80 (4772 ,4697)	1.75 (0.1062 ,0.268)	210 (210 ,207)	5.46 (551, 346)
PC3	6.06 (618 ,614)	21.17 (2159 ,2125)	1.53 (0.093 ,0.234)	180 (180 ,177)	4.17 (421, 264)
PC4	4.50 (459 ,456)	13.48 (1375 ,1353)	1.42 (0.086 ,0.218)	130 (130 ,128)	3.15 (318, 200)
PC5	3.10 (316 ,314)	9.00 (918 ,903)	1.31 (0.080 ,0.201)	70 (70 ,69)	2.50 (252, 158)
PC6	2.40 (245 ,243)	5.49 (560 ,551)	1.17 (0.071 ,0.179)	40 (40 ,39)	2.37 (239, 150)
PC7	1.80 (184 ,183)	4.06 (414 ,407)	1.11 (0.0673 ,0.170)	22 (22 ,22)	1.81 (183, 115)

Note on Units: There are 3 system of units employed in this document. The first is SI, as is used in the IACS Unified Requirement. The second is the MKS system, and the third is the US customary units. In the document units and constants will be shown as SI (MKS, US), as for example: MPa (kgf/mm^2, psi). In many cases the SI and MKS values are the same, but 3 values are always given for complete clarity.

TABLE 2
Class Factors to be Used in 6-1-2/5.5iv) *(1 July 2017)*

Polar Class	*Crushing Failure Class Factor (CF_{CV})*	*Line Load Class Factor (CF_{QV})*	*Pressure Class Factor (CF_{PV})*
PC6	3.43 (350, 347)	2.82 (1.039, 2.608)	0.65 (0.00497, 7.137)
PC7	2.60 (265, 263)	2.33 (0.859, 2.155)	0.65 (0.00497, 7.137)

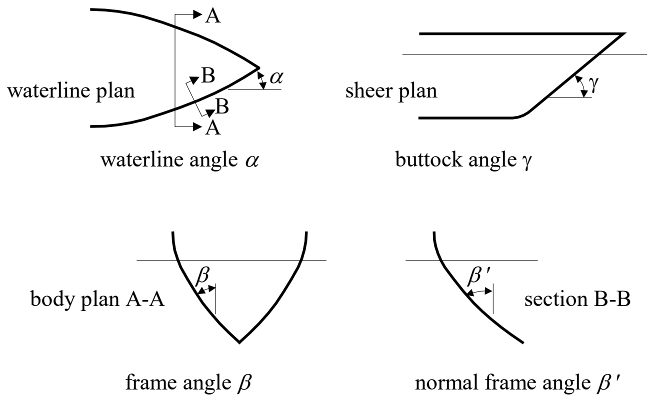
5.5 Bow Area

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 (f_a). The hull angles are defined in 6-1-2/Figure 2.

ii) The waterline length of the bow region is generally to be divided into four 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) *(1 July 2017)* The Bow area load characteristics for bow forms defined in 6-1-2/5.1v) are determined as follows:

FIGURE 2
Definition of Hull Angles *(2012)*



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

a) *Shape Coefficient.* Shape coefficient, fa_i, is to be taken as:

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

where

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

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

$fa_{i,3}$ = 0.60

i = sub-region considered

L = vessel length as defined in 6-1-1/5.7, in m (m, ft)

x = distance from the forward perpendicular (FP) to station under consideration, in m (m, ft)

α = waterline angle, in degrees, see 6-1-2/Figure. 2

β' = normal frame angle, in degrees, see 6-1-2/Figure 2

D = vessel displacement, in kt (kt, kLt), not to be taken less than 5 kt (5 kt, 4.9 kLt) See 6-1-1/5.5

CF_C = Crushing Failure Class Factor from 6-1-2/Table 1

CF_F = Flexural Failure Class Factor from 6-1-2/Table 1

b) *Force.* Force, F, in MN (tf, Ltf) is to be taken as:

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

where

i = sub-region considered

fa_i = shape coefficient of sub-region i

CF_C = Crushing Failure Class Factor from 6-1-2/Table 1

D = vessel displacement, in kt (kt, kLt), not to be taken less than 5 kt (5 kt, 4.9 kLt)

c) *Load Patch Aspect Ratio.* Load patch aspect ratio, AR, is to be taken as:

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

where

i = sub-region considered

β'_i = normal frame angle of sub-region i, in degrees

d) *Line Load.* Line load, Q, in MN/m (tf/cm, Ltf/in) is to be taken as:

$$Q_i = F_i^{0.61} \cdot CF_D/AR_i^{0.35}$$

where

i = sub-region considered

F_i = force of sub-region i, in MN (tf, Ltf)

CF_D = Load Patch Dimensions Class Factor from 6-1-2/Table 1

AR_i = load patch aspect ratio of sub-region i

e) *Pressure.* Pressure, P, in MPa (kgf/mm^2, psi) is to be taken as:

$$P_i = c_1 \cdot F_i^{0.22} \cdot CF_D^2 \cdot AR_i^{0.3}$$

where

i = sub-region considered

F_i = force of sub-region i, in MN (tf, Ltf)

CF_D = Load Patch Dimensions Class Factor from 6-1-2/Table 1

AR_i = load patch aspect ratio of sub-region i

c_1 = 1 (10, 2240)

iv) *(1 July 2017)* The Bow area load characteristics for bow forms defined in 6-1-2/5.1vi) are determined as follows:

a) *Shape Coefficient*

$$fa_i = \alpha_i/30$$

where

i = sub-region considered

α_i = waterline angle, in degrees, see 6-1-2/Figure 2

b) *Force.* Force, F, in MN (tf, Ltf) is to be taken as:

$$F_i = fa_i \cdot CF_{CV} \cdot D^{0.47}$$

where

i = sub-region considered

fa_i = shape coefficient of sub-region i

CF_{CV} = Crushing Failure Class Factor from 6-1-2/Table 2

D = vessel displacement, in kt (kt, kLt), not to be taken less than 5 kt (5 kt, 4.9 kLt) *c)* *Line Load*. Line load, Q, in MN/m (tf/cm, Ltf/in) is to be taken as:

$$Q_i = F_i^{0.22} \cdot CF_{QV}$$

where

i = sub-region considered

F_i = force of sub-region i, in MN (tf, Ltf)

CF_{QV} = Load Patch Dimensions Class Factor from 6-1-2/Table 2

d) *Pressure*. Pressure, P, in MPa (kgf/mm^2, psi) is to be taken as:

$$P_i = F_i^{0.56} \cdot CF_{PV}$$

where

i = sub-region considered

F^i = force of sub-region i, in MN (tf, Ltf)

CF_{PV} = Pressure Class Factor from 6-1-2/Table 2

5.7 Hull Areas Other Than the Bow

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:

5.7.1 Force

Force, F_{NonBow}, in MN (tf, Ltf) is to be taken as:

$$F_{NonBow} = 0.36 \cdot CF_C \cdot DF$$

where

CF_C = Crushing Force Class Factor from 6-1-2/Table 1

DF = vessel displacement factor

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

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

D = vessel displacement, in kt (kt, kLt), not to be taken less than 10 kt (10 kt, 9.8 kLt). See 6-1-1/5.5

CF_{DIS} = Displacement Class Factor from 6-1-2/Table 1

5.7.2 Line Load

Line Load, Q_{NonBow}, in MN/m (tf/cm, Ltf/in) is to be taken as:

$$Q_{NonBow} = 0.639 \cdot F_{NonBow}^{0.61} \cdot CF_D$$

where

F_{NonBow} = force from 6-1-2/5.7.1, in MN (tf, Ltf)

CF_D = Load Patch Dimensions Class Factor from 6-1-2/Table 1

5.9 Design Load Patch

5.9.1 Bow Area

In the Bow area, and the Bow Intermediate Ice belt area for vessels with class notation **PC6** and **PC7**, the design load patch has dimensions of width, w_{Bow}, and height, b_{Bow}, expressed in m (cm, in.) and defined as follows:

i) $w_{Bow} = F_{Bow}/Q_{Bow}$

ii) $b_{Bow} = c_1\, Q_{Bow}/P_{Bow}$

where

F_{Bow} = maximum F_i in the Bow area, in MN (tf, Ltf)

Q_{Bow} = maximum Q_i in the Bow area, in MN/m (tf/cm, Ltf/in)

P_{Bow} = maximum P_i in the Bow area, in MPa (kgf/mm^2, psi)

c_1 = 1 (10, 2240)

5.9.2 Other Hull Areas

In hull areas other than those covered by 6-1-2/5.9.1, the design load patch has dimensions of width, w_{NonBow}, and height, b_{NonBow}, expressed in m (cm, in.) and defined as follows:

i) $w_{NonBow} = F_{NonBow}/Q_{NonBow}$

ii) $b_{NonBow} = w_{NonBow}/3.6$

where

F_{NonBow} = force determined using 6-1-2/5.7.1, in MN (tf, Ltf)

Q_{NonBow} = line load determined using 6-1-2/5.7.2, in MN/m (tf/cm, Ltf/in)

5.11 Pressure within the Design Load Patch

5.11.1 Average Pressure

The average pressure, P_{avg}, in MPa (kgf/mm^2, psi) within a design load patch is determined as follows:

$$P_{avg} = c_1\, F/(b \cdot w)$$

where

F = F_{Bow} or F_{NonBow} as appropriate for the hull area under consideration, in MN (tf, Ltf)

b = b_{Bow} or b_{NonBow} as appropriate for the hull area under consideration, in m (cm, in.)

w = w_{Bow} or w_{NonBow} as appropriate for the hull area under consideration, in m (cm, in.)

c_1 = 1 (10, 2240)

5.11.2 Areas of Higher, Concentrated Pressure

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 6-1-2/Table 3 are used to account for the pressure concentration on localized structural members.

TABLE 3
Peak Pressure Factors *(1 July 2017)*

Structural Member		*Peak Pressure Factor (PPF_i)*
Plating	Transversely Framed	$PPF_p = (1.8 - s/c_2) \geq 1.2$
	Longitudinally Framed	$PPF_p = (2.2 - 1.2 \cdot s/c_2) \geq 1.5$
Frames in Transverse Framing Systems	With Load Distributing Stringers	$PPF_t = (1.6 - s/c_2) \geq 1.0$
	With No Load Distributing Stringers	$PPF_t = (1.8 - s/c_2) \geq 1.2$
Frames in bottom structures		$PPF_s = 1.0$
Load Carrying Stringers Side and Bottom 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, in m (m, in.)
- c_2 = 1 (1, 39.4)
- S_w = web frame spacing, in m (cm, in.)
- w = ice load patch width, in m (cm, in.)

5.13 Hull Area Factors *(1 July 2017)*

i) Associated with each hull area is an Area Factor that reflects the relative magnitude of the load expected in that area. The Area Factors (*AF*) for each hull area for Polar Class vessels are listed in 6-1-2/Table 4 and 6-1-2/Table 5. For ships assigned the additional notation, **Ice Breaker**, the Area Factors (*AF*) for each hull area are listed in 6-1-2/Table 6 and 6-1-2/Table 7.

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 maneuverability, vessels 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. The adjusted hull area factors are listed in 6-1-2/Table 5 and 6-1-2/Table 7 for Polar Class vessels and ships assigned with the additional notation **Ice Breaker**, respectively.

TABLE 4
Hull Area Factors (AF) for Vessels Intended to Operate Ahead Only *(1 July 2017)*

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

Notes:

* See 6-1-2/5.5iii).

** Indicates that strengthening for ice loads is not necessary.

TABLE 5
Hull Area Factors (AF) for Vessels Intended to Operate Ahead and Astern *(1 July 2017)*

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 Intermediate (SI)	Icebelt	SI_i	0.90	0.85	0.85	0.80	0.80	1.00*	1.00*
	Lower	SI_ℓ	0.70	0.65	0.65	0.60	0.55	0.55	0.50
	Bottom	SI_b	0.55	0.50	0.45	0.40	0.35	0.30	0.25
Stern (S)	All	S	1.00	1.00	1.00	1.00	1.00	1.00	1.00

Notes:

* See 6-1-2/5.5iii).

** Indicates that strengthening for ice loads is not necessary.

TABLE 6
Hull Area Factors (AF) for Vessels with Additional Notation Ice Breaker and Intended to Operate Ahead Only *(1 July 2017)*

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

TABLE 7
Hull Area Factors (AF) for Vessels with Additional Notation Ice Breaker and Intended to Operate Ahead and Astern *(1 July 2017)*

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 Intermediate (SI)	Icebelt	SI_i	0.90	0.85	0.85	0.85	0.85	1.00	1.00
	Lower	SI_ℓ	0.70	0.65	0.65	0.65	0.65	0.65	0.65
	Bottom	SI_b	0.55	0.50	0.45	0.45	0.45	0.45	0.45
Stern (S)	All	S	1.00	1.00	1.00	1.00	1.00	1.00	1.00

7 Shell Plate Requirements

7.1 Required Minimum Shell Plate Thickness

The required minimum shell plate thickness, t, expressed in mm (mm, in.), is given by:

$$t = t_{net} + t_s$$

where

t_{net} = plate thickness required to resist ice loads according to 6-1-2/7.3

t_s = corrosion and abrasion allowance according to 6-1-2/21

7.3 Shell Plate Thickness to Resist Ice Load *(1 July 2018)*

The thickness of shell plating required to resist the design ice load, t_{net}, expressed in mm (mm, in.), depends on the orientation of the framing.

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

$$t_{net} = n_0 \cdot s \cdot [(AF \cdot PPF_p \cdot P_{avg})/\sigma_y]^{0.5}/[1 + c_3\, s/(2 \cdot b)]$$

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

$$t_{net} = n_0 \cdot s \cdot [(AF \cdot PPF_p \cdot P_{avg})/\sigma_y]^{0.5}/[1 + s/(2 \cdot \ell)]$$

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

$$t_{net} = n_0 \cdot s \cdot [(AF \cdot PPF_p \cdot P_{avg})/\sigma_y]^{0.5} \cdot [2 \cdot b/(c_3\, s) - (b/(c_3\, s))^2]^{0.5}/[1 + s/(2 \cdot \ell)]$$

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

where

c_3 = 1 (100, 1)

n_0 = 500 (500, 0.5)

Ω = smallest angle between the chord of the waterline and the line of the first level framing as illustrated in 6-1-2/Figure 3, in degrees

s = transverse frame spacing in transversely-framed vessels or longitudinal frame spacing in longitudinally-framed vessels, in m (m, in.)

AF = Hull Area Factor from 6-1-2/Table 4, 6-1-2/Table 5, 6-1-2/Table 6, or 6-1-2/Table 7

PPF_p = Peak Pressure Factor from 6-1-2/Table 3

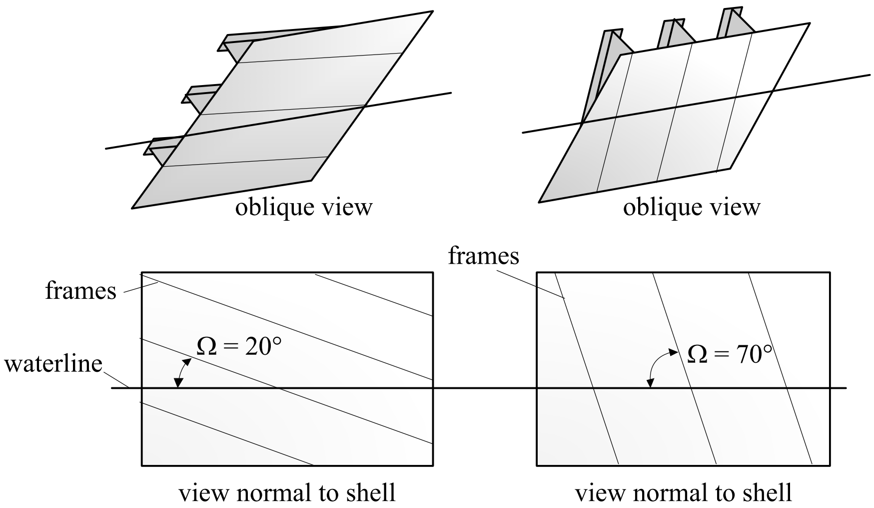
P_{avg} = average patch pressure determined by 6-1-2/5.11.1, in MPa (kgf/mm^2, psi)

σ_y = minimum upper yield stress of the material, in N/mm^2 (kgf/mm^2, psi), but not greater than 690 N/mm^2 (70 kgf/mm^2, 100 ksi)

b = height of design load patch, in m (cm, in.), where b is to be taken not greater than $(\ell - s/4)/c_3$ in the case of determination of the net thickness for transversely framed plating, 6-1-2/7.3i)

ℓ = distance between frame supports in m (m, in.) (i.e., equal to the frame span as given in 6-1-2/9.9), but not reduced for any fitted end brackets, in m (m, in.). When a load-distributing stringer is fitted, the length ℓ need not be taken greater than the distance from the stringer to the most distant frame support.

FIGURE 3
Shell Framing Angle Ω *(2012)*



7.5 Changes in Plating Thickness

Changes in plating thickness in the transverse direction from the ice belt to the bottom and in the longitudinal direction within the ice belt are to be gradually tapered.

9 Framing – General

9.1 General

Framing members of Polar Class vessels are to be designed to withstand the ice loads defined in 6-1-2/5.

9.3 Application *(2014)*

The term “framing member” refers to transverse and longitudinal local frames, load-carrying stringers and load-carrying web frames in the areas of the hull exposed to ice pressure, see 6-1-2/Figure 1.

9.5 Fixity

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.

9.7 Details

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 prepared and submitted for review.

9.9 Framing Span *(1 July 2017)*

The effective span of a framing member is to be determined on the basis of its molded length. If brackets are fitted, the effective span may be reduced provided the bracket is in accordance with 3-2-9/Table 1 and rigidity of the supporting member where the bracket being attached is adequate. Brackets are to be configured to ensure stability in the elastic and post-yield response regions.

9.11 Scantlings

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.

9.13 Net Effective Shear Area *(1 July 2017)*

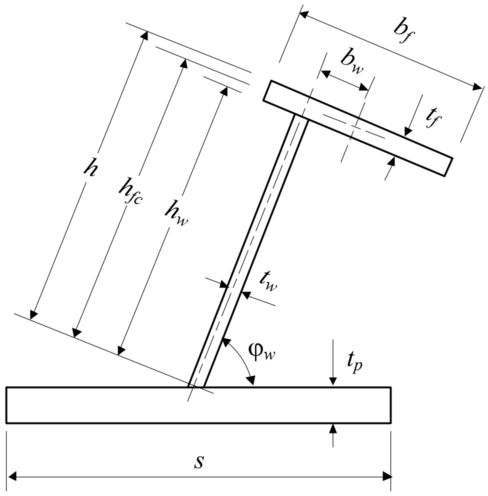
The actual net effective shear area, A_w, in cm^2 (cm^2, in^2) of a transverse or longitudinal local frame is given by:

$$A_w = h \cdot t_{wn} \cdot \sin \varphi_w / c_4^2$$

where

c_4	=	10 (10, 1)
h	=	height of stiffener, in mm (mm, in.), see 6-1-2/Figure 4
t_{wn}	=	net web thickness, in mm (mm, in.)
	=	$t_w - t_c$
t_w	=	as-built web thickness, in mm (mm, in.), see 6-1-2/Figure 4
t_c	=	corrosion deduction, in mm (mm, in.), to be subtracted from the web and flange thickness (but not less than t_s as required by 6-1-2/21.5)
φ_w	=	smallest angle between shell plate and stiffener web, measured at the midspan of the stiffener, see 6-1-2/Figure 4. The angle φ_w may be taken as 90 degrees provided the smallest angle is not less than 75 degrees.

FIGURE 4
Stiffener Geometry *(2014)*



9.15 Net Effective Plastic Section Modulus *(1 July 2017)*

When the net cross-sectional area of the attached plate flange, A_{pn}, exceeds the net cross-sectional area of the frame, A_{frn}, to which the shell plate flange is attached, the actual net effective plastic section modulus, Z_p, in cm³ (cm³, in³), of a transverse or longitudinal frame is given by:

$$Z_p = A_{frn}\, t_{pn}/(2\, c_4) + \frac{h_w^2\, t_{wn} \sin \varphi_w}{2 \cdot c_4^3} + A_{fn}(h_{fc} \sin \varphi_w - b_w \cos \varphi_w)/c_4$$

where

A_{pn}	=	net cross-sectional area of the attached plate flange, in cm² (cm², in²)
	=	$\dfrac{t_{pn} s}{c_4^2}$
A_{frn}	=	net cross-sectional area of the local frame, in cm² (cm², in²)
	=	$\dfrac{h_w\, t_{wn} + b_f t_{fn}}{c_4^2}$
t_{pn}	=	fitted net shell plate thickness, in mm (mm, in.), (is to comply with t_{net} as required by 6-1-2/7.3)
h_w	=	height of local frame web, in mm (mm, in.), see 6-1-2/Figure 4
b_f	=	width of local frame flange, in mm (mm, in.), see 6-1-2/Figure 4
t_{fn}	=	net thickness of local frame flange, in mm (mm, in.), see 6-1-2/Figure 4
A_{fn}	=	net cross-sectional area of local frame flange, in cm² (cm², in²)
	=	$\dfrac{b_f t_{fn}}{c_4^2}$
h_{fc}	=	height of local frame measured to center of the flange area, mm (mm, in.), see 6-1-2/Figure 4
	=	$h_w + \dfrac{t_{fn}}{2}$

b_w = distance from mid thickness plane of local frame web to the center of the flange area, in mm (mm, in.), see 6-1-2/Figure 4

c_4, h, t_{wn}, t_c and φ_w are as given in 6-1-2/9.13 and s is as given in 6-1-2/7.3.

When the net cross-sectional area of the local frame, A_{frn}, exceeds the net cross-sectional area of the attached plate flange, A_{pn}, the plastic neutral axis is located a distance z_{na}, in mm (mm, in.), above the attached shell plate, given by:

$$z_{na} = (c_4^2 \; A_{fn} + h_w \, t_{wn} - c_4^3 \; t_{pn} \, s)/(2 \, t_{wn})$$

and the net effective plastic section modulus, Z_p, in cm^3 (cm^3, in^3), of a transverse or longitudinal frame is given by:

$$Z_p = t_{pn} \, s(z_{na} + t_{pn}/2)\sin \varphi_w + \left[\frac{\left[(h_w - z_{na})^2 + z_{na}^2\right] t_{wn} \sin \varphi_w}{2 \cdot c_4^3} + A_{fn}\left[(h_{fc} - z_{na})\sin \varphi_w - b_w \cos \varphi_w\right] / \, c_4 \right]$$

9.17 Oblique Framing

In the case of oblique framing arrangement (70 degrees > Ω > 20 degrees, where Ω is defined as given in 6-1-2/7.3), linear interpolation is to be used.

11 Framing – Local Frames in Bottom Structures and Transverse Local Frames in Side Structures *(1 July 2017)*

11.1 Plastic Strength

The local frames in bottom structures (i.e., hull areas BI_b, 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.

11.3 Required Shear Area *(1 July 2018)*

The actual net effective shear area of the frame, A_w, as defined in 6-1-2/9.13, is to comply with the following condition: $A_w \geq A_t$ in cm^2 (cm^2, in^2) where:

$$A_t = 100^{n_1} \cdot 0.5 \cdot LL \cdot s \cdot (AF \cdot PPF \cdot P_{avg})/(0.577 \cdot \sigma_y)$$

where

n_1 = 2 (1, 0)

LL = length of loaded portion of span, the lesser of a and b, in m (cm, in.)

a = local frame span as defined in 6-1-2/9.9, in m (cm, in.)

b = height of design ice load patch according to 6-1-2/5.9.1ii) or 6-1-2/5.9.2ii), in m (cm, in.)

s = spacing of local frame, in m (m, in.)

AF = Hull Area Factor from 6-1-2/Table 4 and 6-1-2/Table 5, 6-1-2/Table 6, or 6-1-2/Table 7

PPF = Peak Pressure Factor, PPF_t or PPF_s, as appropriate from 6-1-2/Table 3

P_{avg} = average pressure within load patch according to 6-1-2/5.11.1, in MPa (kgf/mm^2, psi)

σ_y = minimum upper yield stress of the material, in N/mm^2 (kgf/mm^2, psi), but not greater than 690 N/mm^2 (70 kgf/mm^2, 100 ksi)

11.5 Required Plastic Section Modulus

The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in 6-1-2/9.15, is to comply with the following condition: $Z_p \geq Z_{pt}$, in cm^3 (cm^3, in^3) where Z_{pt} is to be the greater calculated on the basis of two load conditions:

i) Ice load acting at the midspan of the local frame, and

ii) The ice load acting near a support.

The A_1 parameter, in the equation below, reflects these two conditions.

$$Z_{pt} = 100^{n_2} \cdot LL \cdot Y \cdot s \cdot (AF \cdot PPF \cdot P_{avg}) \cdot a \cdot A_1/(4 \cdot \sigma_y)$$

where

n_2	=	3 (1, 0)
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 6-1-2/11.3, in cm^2 (cm^2, in^2)
A_w	=	net effective shear area of the local frame (calculated according to 6-1-2/9.13), in cm^2 (cm^2, in^2)
k_w	=	$1/(1 + 2 \cdot A_{fn}/A_w)$ with A_{fn} as given in 6-1-2/9.15
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, in cm^3 (cm^3, in^3)
	=	$(b_f \cdot t_{fn}^2/4 + b_{eff} \cdot t_{pn}^2/4)/c_5$
c_5	=	1000 (1000, 1)
b_f	=	flange breadth, in mm (mm, in.), see 6-1-2/Figure 4
t_{fn}	=	net flange thickness, in mm (mm, in.)
	=	$t_f - t_c$ (t_c as given in 6-1-2/9.13)
t_f	=	as-built flange thickness, in mm (mm, in.), see 6-1-2/Figure 4
t_{pn}	=	fitted net shell plate thickness, in mm (mm, in.) (not to be less than t_{net} as given in 6-1-2/7)
b_{eff}	=	effective width of shell plate flange, in mm (mm, in.)
	=	$0.5\, c_5 \cdot s$
Z_p	=	net effective plastic section modulus of the local frame (calculated according to 6-1-2/9.15), in cm^3 (cm^3, in^3)

AF, PPF, P_{avg}, LL, b, s, a and σ_y are as given in 6-1-2/11.3.

11.7 Structural Stability

The scantlings of the local frame are to meet the structural stability requirements of 6-1-2/17.

13 Framing – Longitudinal Local Frames in Side Structures *(1 July 2017)*

13.1 Plastic Strength *(1 July 2017)*

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.

13.3 Required Shear Area *(1 July 2018)*

The actual net effective shear area of the frame, A_w, as defined in 6-1-2/9.13, is to comply with the following condition: $A_w \geq A_L$, in cm^2 (cm^2, in^2) where:

$$A_L = 100^{n_3} \cdot (AF \cdot PPF_s \cdot P_{avg}) \cdot 0.5 \cdot b_1 \cdot a/(0.577 \cdot \sigma_y)$$

where

n_3 = 2 (0, 0)

AF = Hull Area Factor from 6-1-2/Table 4, 6-1-2/Table 5, 6-1-2/Table 6, or 6-1-2/Table 7

PPF_s = Peak Pressure Factor from 6-1-2/Table 3

P_{avg} = average pressure within load patch according to 6-1-2/5.11.1, in MPa (kgf/mm^2, psi)

b_1 = $k_o \cdot b_2$, in m (cm, in.)

k_o = $1 - 0.3/b'$

b' = $b/(s\ c_6)$

c_6 = 1 (100, 1)

b = height of design ice load patch from 6-1-2/5.9.1ii) or 6-1-2/5.9.2.ii), in m (cm, in.)

s = spacing of longitudinal frames, in m (m, in.)

b_2 = $b\,(1 - 0.25 \cdot b')$, in m (cm, in.) if $b' < 2$

= $s \cdot c_6$, in m (cm, in.) if $b' \geq 2$

a = effective span of longitudinal local frame as given in 6-1-2/9.9, in m (cm, in.)

σ_y = minimum upper yield stress of the material, in N/mm^2 (kgf/mm^2, psi), but not greater than 690 N/mm^2 (70 kgf/mm^2, 100 ksi)

13.5 Required Plastic Section Modulus

The actual net effective plastic section modulus of the plate/stiffener combination, Z_p, as defined in 6-1-2/9.15, is to comply with the following condition: $Z_p \geq Z_{pL}$ in cm^3 (cm^3, in^3) where:

$$Z_{pL} = 100^{n_4} \cdot (AF \cdot PPF_s \cdot P_{avg}) \cdot b_1 \cdot a^2 \cdot A_4/(8 \cdot \sigma_y)$$

where

n_4 = 3 (0, 0)

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 6-1-2/13.3, in cm^2 (cm^2, in^2)

A_w = net effective shear area of longitudinal (calculated according to 6-1-2/9.13), in cm^2 (cm^2, in^2)

k_{wl} = $1/(1 + 2 \cdot A_{fn}/A_w)$ with A_{fn} as given in 6-1-2/9.15

AF, PPF_s, P_{avg}, b_1, a and σ_y are as given in 6-1-2/13.3.

13.7 Structural Stability

The scantlings of the longitudinals are to meet the structural stability requirements of 6-1-2/17.

15 Framing – Web Frames and Load-carrying Stringers *(1 July 2017)*

15.1 General

Web frames and load-carrying stringers are to be designed to withstand the ice load patch as defined in 6-1-2/5. The load patch is to be applied at locations where the capacity of these members under the combined effects of bending and shear is minimized.

15.3 Application *(1 July 2017)*

i) Web frames and load-carrying stringers are to be dimensioned so that the combined stresses of shear and bending are to be maintained within the acceptable limit. Where these members form part of a structural grillage system, appropriate methods of analysis are to be used. Where the structural configuration is such that members do not form part of a grillage system, the appropriate peak pressure factor (PPF) from 6-1-2/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.

ii) 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, the method outlined in 6-1-2/33 is to be used, in general.

15.5 Structural Stability

The scantlings of web frames and load-carrying stringers are to meet the structural stability requirements of 6-1-2/17.

17 Framing – Structural Stability

17.1 Framing Members *(1 July 2018)*

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

- For flat bar sections: $h_w/t_{wn} \le c_7/(\sigma_y)^{0.5}$
- For bulb, tee and angle sections: $h_w/t_{wn} \le c_8/(\sigma_y)^{0.5}$

where

c_7 = 282 (90, 3396)

c_8 = 805 (257, 9695)

h_w = web height in mm (mm, in.)

t_{wn} = net web thickness in mm (mm, in.)

σ_y = minimum upper yield stress of the material, in N/mm^2 (kgf/mm^2, psi), but not greater than 690 N/mm^2 (70 kgf/mm^2, 100 ksi)

17.3 Web Stiffening *(1 July 2018)*

Framing members for which it is not practicable to meet the requirements of 6-1-2/17.1 (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, t_{wn}, in mm (mm, in.), for these framing members is given by:

$$t_{wn} = 2.63 \cdot 10^{-3} \cdot h_u \cdot [\sigma_y/(c_{11} + c_{12} \cdot (h_u/L_w)^2)]^{0.5}$$

where

h_u = $h_w - 0.8h_f$ mm (mm, in.)

h_w = web height of stringer/web frame , in mm (mm, in.) (see 6-1-2/Figure 5)

h_f = height of framing member penetrating the member under consideration (0 if no such framing member), in mm (mm, in.) (see 6-1-2/Figure 5)

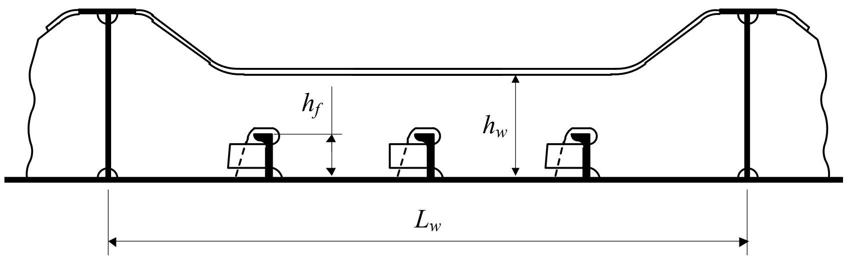
L_w = spacing between supporting structure oriented perpendicular to the member under consideration, in mm (mm, in.) (see 6-1-2/Figure 5)

c_{11} = 5.34 (0.545, 775)

c_{12} = 4 (0.41, 580)

σ_y = minimum upper yield stress of the material, in N/mm^2 (kgf/mm^2, psi), but not greater than 690 N/mm^2 (70 kgf/mm^2, 100 ksi)

FIGURE 5
Parameter Definition for Web Stiffening *(2014)*



17.5 Web Thickness

In addition, the following is to be satisfied:

$$t_{wn} \geq 0.35 \cdot t_{pn} \cdot (\sigma_y/c_{13})^{0.5}$$

where

σ_y = minimum upper yield stress of the shell plate in way of the framing member, in N/mm^2 (kgf/mm^2, psi)

c_{13} = 235 (24, 34083)

t_{wn} = net thickness of the web, in mm (mm, in.)

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

17.7 Flange Width and Outstand *(1 July 2018)*

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

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

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

$$b_{out}/t_{fn} \leq c_{14}/(\sigma_y)^{0.5}$$

where

c_{14} = 155 (49.5, 1867)

t_{fn} = net thickness of flange, in mm (mm, in.)

σ_y = minimum upper yield stress of the material, in N/mm^2 (kgf/mm^2, psi), but not greater than 690 N/mm^2 (70 kgf/mm^2, 100 ksi)

19 Plated Structures

19.1 General

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

19.3 End Fixity

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.

19.5 Stability

The stability of the plated structure is to adequately withstand the ice loads defined in 6-1-2/5.

21 Corrosion/Abrasion Additions and Steel Renewal

21.1 General

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

21.3 Corrosion/Abrasion Additions for Shell Plating *(1 July 2017)*

The values of corrosion/abrasion additions, t_s, in mm (mm, in.) to be used in determining the shell plate thickness are listed in 6-1-2/Table 8 and 6-1-2/Table 9.

21.5 Corrosion/Abrasion Additions for Internal Structures

Polar Class vessels are to have a minimum corrosion/abrasion addition of t_s = 1.0 mm (1.0 mm, 0.0394 in.) 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.

21.7 Steel Renewal

Steel renewal for ice strengthened structures is required when the gauged thickness is less than t_{net} + 0.5 mm (0.5 mm, 0.02 in.).

TABLE 8
Corrosion/Abrasion Additions for Shell Plating for Vessels Intended to Operate Ahead Only *(1 July 2017)*

Hull Area	*t_s, mm (mm, in.)*					
	With Effective Protection			*Without Effective Protection*		
	PC1 - PC3	**PC4 & PC5**	**PC6 & PC7**	**PC1 - PC3**	**PC4 & PC5**	**PC6 & PC7**
Bow; Bow Intermediate Icebelt	3.5 (3.5, 0.138)	2.5 (2.5, 0.098)	2.0 (2.0, 0.079)	7.0 (7.0, 0.276)	5.0 (5.0, 0.197)	4.0 (4.0, 0.158)
Bow Intermediate Lower; Midbody & Stern Icebelt	2.5 (2.5, 0.098)	2.0 (2.0, 0.079)	2.0 (2.0, 0.079)	5.0 (5.0, 0.197)	4.0 (4.0, 0.158)	3.0 (3.0, 0.118)
Midbody & Stern Lower; Bottom	2.0 (2.0, 0.079)	2.0 (2.0, 0.079)	2.0 (2.0, 0.079)	4.0 (4.0, 0.158)	3.0 (3.0, 0.118)	2.5 (2.5, 0.098)

TABLE 9
Corrosion/Abrasion Additions for Shell Plating for Vessels Intended to Operate Ahead and Astern *(1 July 2017)*

Hull Area	*t_s, mm (mm, in.)*					
	With Effective Protection			*Without Effective Protection*		
	PC1 - PC3	**PC4 & PC5**	**PC6 & PC7**	**PC1 - PC3**	**PC4 & PC5**	**PC6 & PC7**
Bow; Bow Intermediate Icebelt; Stern; Stern Intermediate Icebelt	3.5 (3.5, 0.138)	2.5 (2.5, 0.098)	2.0 (2.0, 0.079)	7.0 (7.0, 0.276)	5.0 (5.0, 0.197)	4.0 (4.0, 0.158)
Bow Intermediate Lower; Midbody Icebelt & Stern Intermediate Lower	2.5 (2.5, 0.098)	2.0 (2.0, 0.079)	2.0 (2.0, 0.079)	5.0 (5.0, 0.197)	4.0 (4.0, 0.158)	3.0 (3.0, 0.118)
Midbody Lower; Bottom	2.0 (2.0, 0.079)	2.0 (2.0, 0.079)	2.0 (2.0, 0.079)	4.0 (4.0, 0.158)	3.0 (3.0, 0.118)	2.5 (2.5, 0.098)

23 Materials *(1 July 2017)*

All hull structural materials are to be in accordance with the requirements of Part 2, Chapter 1 and the following paragraphs.

23.1 General *(1 July 2017)*

Steel grades of plating for hull structures are to be not less than those given in 6-1-2/Table 11 based on the as-built thickness, the Polar Class and the Material Class of structural members according to 6-1-2/23.3.

23.3 Material Classes

Material classes specified in 3-1-2/Table 2 are applicable to Polar Class vessels regardless of the vessel's length. In addition, material classes for weather and sea exposed structural members and for members attached to the weather and sea exposed plating of polar vessels are given in 6-1-2/Table 10. Where the material classes in 6-1-2/Table 10 and those in 3-1-2/Table 2 differ, the higher material class is to be applied.

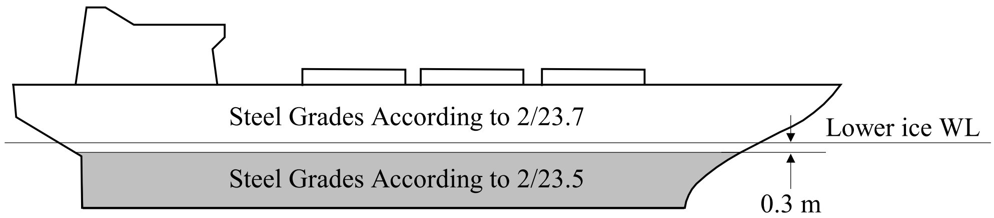
TABLE 10
Material Classes for Structural Members of Polar Class Vessels *(2012)*

Structural Members	*Material Class*
Shell plating within the bow and bow intermediate icebelt hull areas (B, BI_i)	II
All weather and sea exposed SECONDARY and PRIMARY, as defined in 3-1-2/Table 2, structural members outside 0.4*L* 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 (600 mm, 23.6 in.) of the plating	I
Weather-exposed plating and attached framing in cargo holds of vessels 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 3-1-2/Table 2, structural members within 0.2*L* from FP	II

23.5 Steel Grades

Steel grades for all plating and attached framing of hull structures and appendages situated below the level of 0.3 m (0.3 m, 12 in.) below the lower waterline, as shown in 6-1-2/Figure 6, are to be obtained from 3-1-2/Table 1 based on the Material Class for Structural Members in 6-1-2/Table 10 above, regardless of Polar Class.

FIGURE 6
Steel Grade Requirements for Submerged and Weather Exposed Shell Plating *(2012)*



23.7 Steel Grades for Weather Exposed Plating

Steel grades for all weather exposed plating of hull structures and appendages situated above the level of 0.3 m (0.3 m, 12 in.) below the lower ice waterline, as shown in 6-1-2/Figure 6, are to be not less than given in 6-1-2/Table 11.

TABLE 11
Steel Grades for Weather Exposed Plating (1, 2) *(1 July 2018)*

Thickness, t mm (in.)	*Material Class I*						*Material Class II*					
	PC1-5			**PC6 & 7**			**PC1-5**			**PC6 & 7**		
	MS	HT	XHT	MS	HT	XHT	MS	HT	XHT	MS	HT	XHT
$t \le 10$ $t \le 0.394$	B	AH	AQ	B	AH	AQ	B	AH	AQ	B	AH	AQ
$10 < t \le 15$ $0.394 < t \le 0.591$	B	AH	AQ	B	AH	AQ	D	DH	DQ	B	AH	AQ
$15 < t \le 20$ $0.591 < t \le 0.787$	D	DH	DQ	B	AH	AQ	D	DH	DQ	B	AH	AQ
$20 < t \le 25$ $0.787 < t \le 0.984$	D	DH	DQ	B	AH	AQ	D	DH	DQ	B	AH	AQ
$25 < t \le 30$ $0.984 < t \le 1.18$	D	DH	DQ	B	AH	AQ	E	EH	EQ	D	DH	DQ
$30 < t \le 35$ $1.18 < t \le 1.38$	D	DH	DQ	B	AH	AQ	E	EH	EQ	D	DH	DQ
$35 < t \le 40$ $1.38 < t \le 1.58$	D	DH	DQ	D	DH	DQ	E	EH	EQ	D	DH	DQ
$40 < t \le 45$ $1.58 < t \le 1.77$	E	EH	EQ	D	DH	EQ	E	EH	EQ	D	DH	DQ
$45 < t \le 50$ $1.77 < t \le 1.97$	E	EH	EQ	D	DH	EQ	E	EH	EQ	D	DH	DQ

TABLE 11 (continued)
Steel Grades for Weather Exposed Plating (1, 2) *(1 July 2018)*

Thickness, t mm (in.)	*Material Class III*								
	PC1-3			**PC4 & 5**			**PC6 & 7**		
	MS	HT	XHT	MS	HT	XHT	MS	HT	XHT
$t \le 10$ $t \le 0.394$	E	EH	EQ	E	EH	EQ	B	AH	AQ
$10 < t \le 15$ $0.394 < t \le 0.591$	E	EH	EQ	E	EH	EQ	D	DH	DQ
$15 < t \le 20$ $0.591 < t \le 0.787$	E	EH	EQ	E	EH	EQ	D	DH	DQ
$20 < t \le 25$ $0.787 < t \le 0.984$	E	EH	EQ	E	EH	EQ	D	DH	DQ
$25 < t \le 30$ $0.984 < t \le 1.18$	E	EH	EQ	E	EH	EQ	E	EH	EQ
$30 < t \le 35$ $1.18 < t \le 1.38$	E	EH	EQ	E	EH	EQ	E	EH	EQ
$35 < t \le 40$ $1.38 < t \le 1.58$		FH	FQ	E	EH	EQ	E	EH	EQ
$40 < t \le 45$ $1.58 < t \le 1.77$		FH	FQ	E	EH	EQ	E	EH	EQ
$45 < t \le 50$ $1.77 < t \le 1.97$		FH	FQ		FH	FQ	E	EH	EQ

MS: Ordinary strength steel, HT: High strength steel, XHT: Extra high strength steel

Notes:

1 Includes weather-exposed plating of hull structures and appendages, as well as their outboard framing members, situated above a level of 0.3 m (0.3 m, 12 in.) 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 (1.8 m, 70.9 in.) wide from 0.3 m (0.3 m, 12 in.) below the lowest ice waterline.

23.9 Castings

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

25 Longitudinal Strength

25.1 Application *(1 July 2017)*

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

ii) Intentional ramming is not considered as a design scenario for ships which are designed with vertical or bulbous bows, see 6-1-1/1.1viii) Hence the longitudinal strength requirements given in this section are not to be considered for ships with stem angle γ_{stem} equal to or larger than 80°.

iii) 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 vessel's length. In addition, sufficient local buckling strength is also to be maintained.

25.3 Design Vertical Ice Force at the Bow

The design vertical ice force at the bow, F_{IB}, in MN (tf, Ltf) is to be taken as:

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

where

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

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

K_I = indentation parameter = K_f/K_h

a) For the case of a blunt bow form:

K_f = $c_{15}[2 \cdot C \cdot (B/c_{16})^{1-e_b}/(1+e_b)]^{0.9} \cdot \tan(\gamma_{stem})^{-0.9(1+e_b)}$ MN/m (tf/cm, Ltf/in)

c_{15} = 1 (1.02, 2.54)

c_{16} = 1 (1, 3.28)

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

K_f = $[\tan(\alpha_{stem})/\tan^2(\gamma_{stem})]^{0.9}$ MN/m (tf/cm, Ltf/in)

K_h = $c_{17} A_{wp}$ MN/m (tf/cm, Ltf/in)

c_{17} = 0.01 (0.01, 0.00237) MN/m^3 [tf/(m^2-cm), Ltf/(ft^2-in)]

CF_L = Longitudinal Strength Class Factor from 6-1-2/Table 1

e_b = bow shape exponent which best describes the waterplane (see 6-1-2/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 , in degrees (buttock angle as per 6-1-2/Figure 2 measured on the centerline)

α_{stem} = waterline angle measured in way of the stem at the upper ice waterline (UIWL), in degrees (see 6-1-2/Figure 7)

C = $1/[2 \cdot (L_B/B)^{e_b}]$

B = vessel molded breadth, in m (m, ft)

L_B = bow length used in the equation $y = B/2 \cdot (x/L_B)^{e_b}$, in m (m, ft) (see 6-1-2/Figure 7 and 6-1-2/Figure 8)

D = vessel displacement, in kt (kt, kLt), not to be taken less than 10 kt (10 kt, 9.8 kLt)

A_{wp} = vessel waterplane area, in m^2 (m^2, ft^2)

CF_F = Flexural Failure Class Factor from 6-1-2/Table 1

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

FIGURE 7
Bow Shape Definition *(2012)*

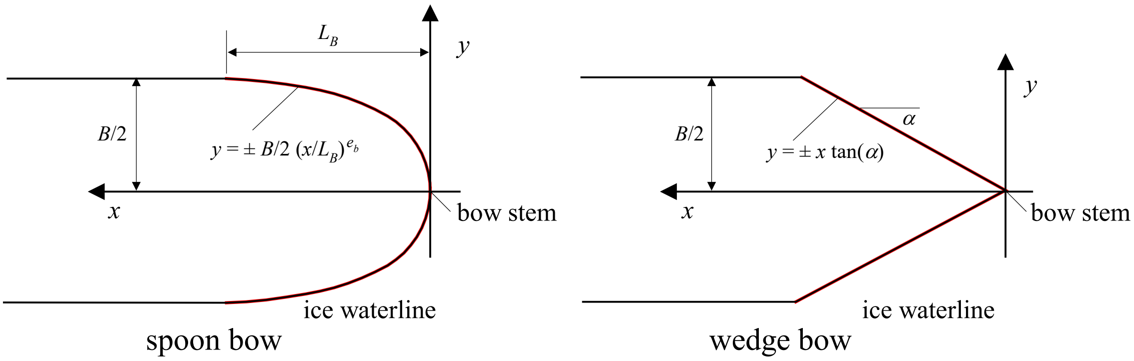
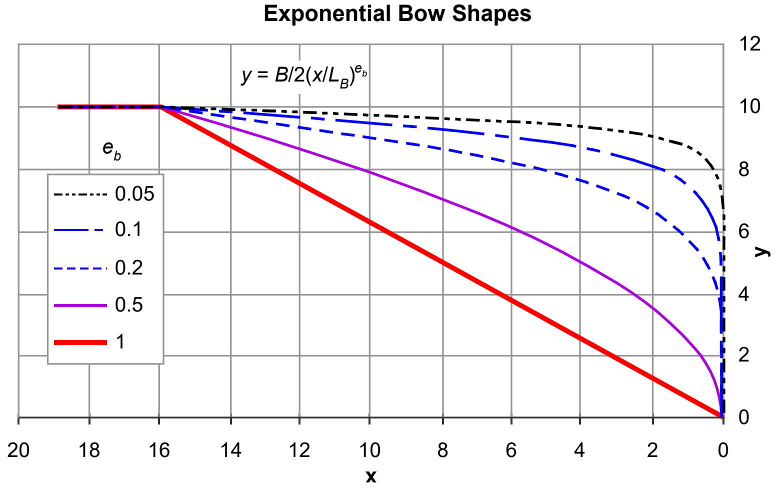


FIGURE 8
Illustration of e_b Effect on the Bow Shape for B = 20 and L_B = 16 *(2012)*



25.5 Design Vertical Shear Force

25.5.1

The design vertical ice shear force, F_I, in MN (tf, Ltf) along the hull girder is to be taken as:

$$F_I = C_f \cdot F_{IB}$$

where

C_f = longitudinal distribution factor to be taken as follows:

i) *Positive shear force*

C_f = 0.0 between the aft end of L and 0.6L from aft

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

ii) *Negative shear force*

C_f = 0.0 at the aft end of L

= –0.5 between 0.2L and 0.6L from aft

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

Intermediate values are to be determined by linear interpolation

25.5.2

The applied vertical shear stress, τ_a, is to be determined along the hull girder in a similar manner as in 3-2-1/3.9 by substituting the design vertical ice shear force for the design vertical wave shear force.

25.7 Design Vertical Ice Bending Moment

25.7.1

The design vertical ice bending moment, M_I, in MN-m (tf-m, Ltf-ft) along the hull girder is to be taken as:

$$M_I = 0.1 \cdot C_m \cdot L \cdot \sin^{-0.2}(\gamma_{stem}) \cdot F_{IB}$$

where

L = vessel length as defined in 6-1-1/5.7, in m (m, ft)

γ_{stem} = as given in 6-1-2/25.3

F_{IB} = design vertical ice force at the bow, in MN (tf, Ltf)

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

= 0.0 at the aft end of L

= 1.0 between 0.5L and 0.7L from aft

= 0.3 at 0.95L from aft

= 0.0 at the forward end of L

Intermediate values are to be determined by linear interpolation

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

25.7.2

The applied vertical bending stress, σ_a, is to be determined along the hull girder in a similar manner as in 3-2-1/3.7 by substituting the design vertical ice bending moment for the design vertical wave bending moment. The vessel still water bending moment is to be taken as the maximum sagging moment.

25.9 Longitudinal Strength Criteria *(1 July 2018)*

The strength criteria provided in 6-1-2/Table 12 are to be satisfied. The design stress is not to exceed the permissible stress.

TABLE 12
Longitudinal Strength Criteria *(1 July 2017)*

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

where

σ_a = applied vertical bending stress, in N/mm^2 (kgf/mm^2, psi)

τ_a = applied vertical shear stress, in N/mm^2 (kgf/mm^2, psi)

σ_y = minimum upper yield stress of the material, in N/mm^2 (kgf/mm^2, psi), but not greater than 690 N/mm^2 (70 kgf/mm^2, 100 ksi)

σ_u = ultimate tensile strength of material, in N/mm^2 (kgf/mm^2, psi)

σ_c = critical buckling stress in compression, according to Appendix 3-2-A4, in N/mm^2 (kgf/mm^2, psi)

τ_c = critical buckling stress in shear, according to Appendix 3-2-A4, in N/mm^2 (kgf/mm^2, psi)

η = 0.6 for ships which are assigned the additional notation **Ice Breaker**

= 0.8 otherwise

27 Stem and Stern Frames

For Polar Class **PC6** and **PC7** vessels requiring Baltic Ice Class **I AA** or **I A** equivalency of Section 6-1-6, the stem and stern requirements of the Finnish-Swedish Ice Class Rules may need to be additionally considered.

29 Appendages

29.1 General

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.

29.3 Load Definition and Response Criteria

Load definition and response criteria are to be determined on a case-by-case basis.

31 Local Details

31.1 General

For the purpose of transferring ice-induced loads to supporting structure (bending moments and shear forces), local design details are to prepared and submitted for review.

31.3 Cut-outs

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

33 Direct Calculations *(1 July 2017)*

33.1 General

i) Direct calculations are not to be utilized as an alternative to the analytical procedures prescribed for the shell plating and local frame requirements given in 6-1-2/7, 6-1-2/11 and 6-1-2/13.

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

33.3 Load Patch

Where direct calculation is used to check the strength of structural systems, the load patch specified in 6-1-2/5 is to be applied, without being combined with any other loads. The load patch is to be applied at locations where the capacity of these members under the combined effects of bending and shear is minimized. Special attention is to be paid to the shear capacity in way of lightening holes and cut-outs in way of intersecting members.

33.5 Strength Evaluation

The strength evaluation of web frames and stringers may be performed based on linear or non-linear analysis. Recognized structural idealization and calculation methods are to be applied, with detailed requirements agreed upon with ABS. In the strength evaluation, the guidance given in 6-1-2/33.7 and 6-1-2/33.9 may generally be considered.

33.7 Acceptance Criteria – Linear Analysis Methods

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

i) Web plates and flange elements in compression and shear to fulfill relevant buckling criteria

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

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

33.9 Acceptance Criteria – Non-linear Analysis Methods

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

i) The analysis is to reliably capture buckling and plastic deformation of the structure.

ii) The acceptance criteria are to ensure a suitable margin against fracture and major buckling and yielding that causes significant loss of stiffness.

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

iv) Detailed acceptance criteria are to be decided by ABS.

35 Welding

35.1 General

Hull construction welding design is to comply with Section 3-2-19. All welding within ice-strengthened areas is to be of the double continuous type.

35.3 Continuity of Strength

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

35.5 Filler Metals

When the ABS ordinary and higher strength hull steels of 2-1-2/Table 5 or 2-1-3/Table 5 are applied in accordance with 6-1-2/Table 11, approved filler metals appropriate to the grades shown in Part 2, Appendix 3 may be used.

35.7 Hull Steels Other than the ABS Grades

For the welding of hull steels other than the ABS grades in 6-1-2/Table 11, weld metal is to exhibit a Charpy V-Notch toughness value at least equivalent to the transverse base metal requirements ($^2/_3$ of longitudinal base metal requirements).

PART
6

CHAPTER **1** **Strengthening for Navigation in Ice**

SECTION **3** **Machinery Requirements for Polar Class Vessels** *(2012)*

1 Application

The contents of Section 6-1-3 apply to main propulsion, steering gear, emergency and essential auxiliary systems essential for the safety of the vessel and the survivability of the crew.

The vessel operating conditions are defined in Section 6-1-1.

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

3 Drawings and Particulars to be Submitted

3.1 Environmental Conditions

Details of the environmental conditions and the required ice class for the machinery, if different from vessel's ice class.

3.3 Drawings

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

3.5 Description Detailing

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

3.7 Calculations and Documentation *(1 July 2019)*

Calculations and documentation indicating compliance with the requirements of Section 6-1-3.

The following table shows a sample list of information and calculations required to be submitted:

General	Torsional vibration calculations addressing the effect of ice/propeller interaction (including as a minimum: i) Shaft speed drop curve (s) due to ice impact ii) Shaft response torque curve(s) etc.) Fatigue calculations for the propulsion line components considering ice loads (including: S-N curves, Miner's Rule calculations etc.)
Main Engine	Main engine power curve (power supply), Geometrical details (i.e. overall dimensions and detailed dimensions of the crankthrows) and material properties of the crankshaft, Harmonic packs (i.e. excitation tables), Torsional damping coefficients between journals.
Propeller	Propeller power curve (power demand), Main particulars, Inertial properties, Water entrained factors, Torsional and axial damping coefficients
Shaftline (including crankshaft)	Bearing offsets, Bearing clearances, Bearing radial and axial stiffnesses, Bearing torsional and axial damping coefficients,

5 System Design

5.1 General

All machinery is to be suitable for operation under the environmental conditions to which it will be exposed in service and is to include all necessary special provisions for that purpose.

5.3 Governmental Authority

Attention is directed to the appropriate governmental authorities in the intended regions of operation for additional requirements in consideration of operation in ice such as fuel capacity, refueling capability, water capacity, radio communications requirements, etc.

5.5 Damage by Freezing

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

5.7 Propeller Damage

Vessels classed **PC1**, to **PC5** inclusive shall have means provided to ensure sufficient vessel operation in the case of propeller damage including CP-mechanism (i.e., pitch control mechanism).

Sufficient vessel operation means that the vessel should be able to reach safe harbor (safe location) where repair can be undertaken in case of propeller damage. This may be achieved either by a temporary repair at sea, or by towing assuming assistance is available (condition for approval).

5.9 Turning Gear

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

7 Materials

7.1 Materials Exposed to Sea Water

Materials exposed to sea water, such as propeller blades, propeller hub, cast thrusters body shall have an elongation not less than 15% on a test specimen with a length which is five times the diameter of test specimen.

Charpy V impact tests shall be carried out for materials other than bronze and austenitic steel. Average impact energy of 20 J (20 J, 14.75 lbf-ft) taken from three Charpy V tests is to be obtained at –10°C (–10°C, +14°F).

7.3 Materials Exposed to Sea Water Temperature

Materials exposed to sea water temperature shall be of steel or other approved ductile material. Charpy V impact tests shall be carried out for materials other than bronze and austenitic steel. Average impact energy value of 20 J (20 J, 14.75 lbf-ft) taken from three Charpy V tests is to be obtained –10°C (–10°C, +14°F).

This requirement applies to blade bolts, CP-mechanisms, shaft bolts, strut-pod connecting bolts, etc. This does not apply to surface hardened components, such as bearings and gear teeth.

For definition of structural boundaries exposed to sea water temperature see 6-1-2/Figure 6.

7.5 Materials Exposed to Low Air Temperature

Materials of essential components exposed to low air temperature shall be of steel or other approved ductile material. Average impact energy value of 20 J (20 J, 14.75 lbf-ft) taken from three Charpy V tests is to be obtained at 10°C (10°C, 50°F) below the lowest design temperature.

This does not apply to surface hardened components, such as bearings and gear teeth.

For definition of structural boundaries exposed to air temperature see 6-1-2/Figure 6.

9 Ice Interaction Load

9.1 Propeller-Ice Interaction

These Rules cover open and ducted type propellers situated at the stern of a vessel having controllable pitch or fixed pitch blades. Ice loads on bow propellers shall receive special consideration to discretion of each society. The given loads are expected, single occurrence, maximum values for the whole ships service life for normal operational conditions. These loads do not cover off-design operational conditions, for example when a stopped propeller is dragged through ice. These Rules cover loads due to propeller ice interaction also for azimuth and fixed thrusters with geared transmission or 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 load case when ice block hits on the propeller hub of a pulling propeller.

The loads given in section 6-1-3/9 are total loads (unless otherwise stated) during ice interaction and are to be applied separately (unless otherwise stated) and are intended for component strength calculations only.

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

9.3 Ice Class Factors

6-1-3/Table 1 below lists the design ice thickness and ice strength index to be used for estimation of the propeller ice loads.

TABLE 1
Design Ice Thickness and Ice Strength Index *(2012)*

Ice Class	H_{ice}, m (m, ft)	S_{ice}, [-]
PC1	4.0 (4.0, 13.12)	1.2
PC2	3.5 (3.5, 11.48)	1.1
PC3	3.0 (3.0, 9.84)	1.1
PC4	2.5 (2.5, 8.20)	1.1
PC5	2.0 (2.0, 6.56)	1.1
PC6	1.75 (1.75, 5.74)	1
PC7	1.5 (1.5, 4.92)	1

where

H_{ice} = ice thickness in m (m, ft) for machinery strength design

S_{ice} = ice strength index for blade ice force

9.5 Design Ice Loads for Open Propeller

9.5.1 Maximum Backward Blade Force

The maximum backward blade force, F_b, in kN (tf, Ltf), is to be taken as:

- when $D < D_{limit}$:

$$F_b = c_0 \cdot S_{ice} [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot D^2$$

- when $D \geq D_{limit}$:

$$F_b = c_1 \cdot S_{ice} [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot [H_{ice}]^{1.4} \cdot D$$

where

D_{limit} = $c_2 \cdot (H_{ice})^{1.4}$ m (m, ft)

c_0 = 27 (2.753, 0.1096)

c_1 = 23 (2.345, 0.0580)

c_2 = 0.85 (0.85, 0.528)

n = nominal rotational speed, in rev/s, (at MCR free running condition) for CP-propeller and 85% of the nominal rotational speed (at MCR free running condition) for a FP-propeller (regardless driving engine type)

D = propeller diameter, in m (m, ft)

EAR = expanded blade area ratio

Z = number of propeller blades

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

- *Load Case 1:* From 0.6R to the tip and from the blade leading edge to a value of 0.2 chord lengths.
- *Load Case 2:* A load equal to 50% of the F_b is to be applied on the propeller tip area outside of 0.9R.
- *Load Case 5:* For reversible propellers, a load equal to 60% of the F_b is to be applied from 0.6R to the tip and from the blade trailing edge to a value of 0.2 chord lengths measured from trailing edge.

See load cases 1, 2, and 5 in 6-1-3/Table 2.

9.5.2 Maximum Forward Blade Force

The maximum forward blade force, F_f, in kN (tf, Ltf) is to be taken as:

- when $D < D_{limit}$:

$$F_f = c_3 \cdot \left[\frac{EAR}{Z}\right] \cdot D^2$$

- when $D \geq D_{limit}$:

$$F_f = 2c_3 \left[\frac{1}{1-d/D}\right] \cdot H_{ice} \cdot \left[\frac{EAR}{Z}\right] \cdot D$$

where

D_{limit} = $\left[\frac{2}{1-d/D}\right] \cdot H_{ice}$ m (m, ft)

c_3 = 250 (25.493, 2.331)

d = propeller hub diameter, in m (m, ft)

D = propeller diameter, in m (m, ft)

EAR = expanded blade area ratio

Z = number of propeller blades

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

- *Load Case 3:* From 0.6R to the tip and from the blade leading edge to a value of 0.2 chord length.
- *Load Case 4:* A load equal to 50% of the F_f is to be applied on the propeller tip area outside of 0.9R.
- *Load Case 5:* For reversible propellers, a load equal to 60% of F_f is to be applied from 0.6R to the tip and from the blade trailing edge to a value of 0.2 chord lengths measured from trailing edge.

See load cases 3, 4, and 5 in 6-1-3/Table 2.

9.5.3 Maximum Blade Spindle Torque

Spindle torque, Q_{smax}, in kN-m (tf-m, Ltf-ft), around the spindle axis of the blade fitting shall be calculated both for the load cases described in 6-1-3/9.5.1 and 6-1-3/9.5.2 for F_b and F_f. If these spindle torque values are less than the default value given below, the default minimum value to be used.

Default Value: $Q_{smax} = 0.25 \cdot F \cdot c_{0.7}$

where

$c_{0.7}$ = length of the blade chord at 0.7R radius, in m (m, ft)

F = either F_b or F_f, in kN (tf, Ltf), whichever has the greater absolute value

9.5.4 Maximum Propeller Ice Torque Applied to the Propeller

The maximum propeller ice torque, Q_{max}, in kN-m (tf-m, Ltf-ft) applied to the propeller is to be taken as:

- when $D < D_{limit}$:

$$Q_{max} = k_{open} \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot [n \cdot D]^{0.17} \cdot D^3$$

where

k_{open} = 14.7 (1.50, 0.112) for PC1 – PC5; and

k_{open} = 10.9 (1.11, 0.083) for PC6 – PC7

- when $D \geq D_{limit}$:

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

where

D_{limit} = $1.81 H_{ice}$ m (m, ft)

$P_{0.7}$ = propeller pitch at 0.7R, in m (m, ft)

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

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

where n_n is the nominal rotational speed at MCR, free running condition

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

9.5.5 Maximum Propeller Ice Thrust Applied to the Shaft

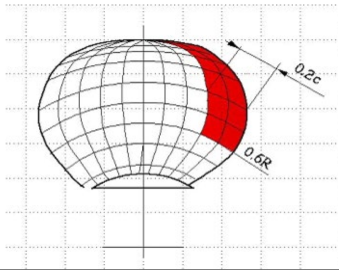
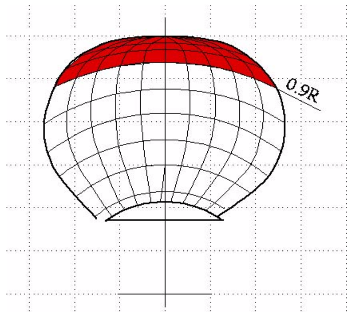
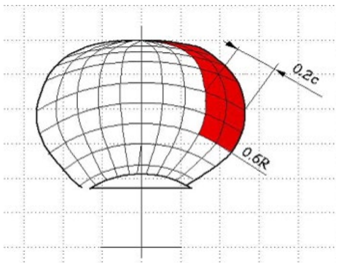
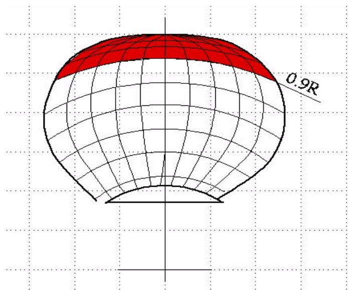
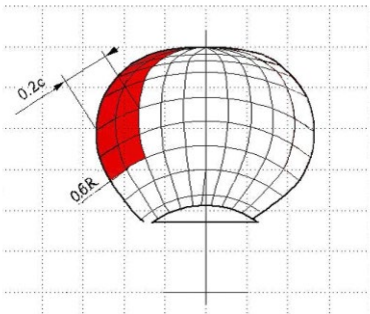
The maximum propeller ice thrust, in kN (tf, Ltf), applied to the shaft is to be taken as:

$$T_f = 1.1 \cdot F_f$$

$$T_b = 1.1 \cdot F_b$$

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

TABLE 2
Load Cases for Open Propeller *(2012)*

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

9.7 Design Ice Loads for Ducted Propeller

9.7.1 Maximum Backward Blade Force

The maximum backward blade force, F_b, in kN (tf, Ltf) is to be taken as:

- when $D < D_{limit}$:

$$F_b = c_4 \cdot S_{ice} [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot D^2$$

- when $D \geq D_{limit}$:

$$F_b = c_5 \cdot S_{ice} [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot [H_{ice}]^{1.4} \cdot D^{0.6}$$

where

D_{limit} = $4H_{ice}$ m (m, ft)

c_4 = 9.5 (0.969, 0.0386)

c_5 = 66 (6.730, 0.2679)

n is to be taken as in 6-1-3/9.5.1.

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

- *Load Case 1:* On the back of the blade from 0.6R to the tip and from the blade leading edge to a value of 0.2 chord lengths.
- *Load Case 5:* For reversible rotation propellers, a load equal to 60% of F_b is applied on the blade face from 0.6R to the tip and from the blade trailing edge to a value of 0.2 chord lengths measured from trailing edge.

See load cases 1 and 5 in 6-1-3/Table 3.

9.7.2 Maximum Forward Blade Force

The maximum forward blade force, F_f, in kN (tf, Ltf), is to be taken as:

- when $D \leq D_{limit}$:

$$F_f = c_3 \cdot \left[\frac{EAR}{Z}\right] \cdot D^2$$

- when $D > D_{limit}$:

$$F_f = 2c_3 \left[\frac{1}{1 - d/D}\right] \cdot H_{ice} \cdot \left[\frac{EAR}{Z}\right] \cdot D$$

where

D_{limit} = $\dfrac{2}{1-(d/D)} \cdot H_{ice}$ m (m, ft)

c_3 = 250 (25.493, 2.331)

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

- *Load Case 3:* On the blade face from 0.6R to the tip and from the blade leading edge to a value of 0.5 chord lengths.
- *Load Case 5:* A load equal to 60% F_f is to be applied from 0.6R to the tip and from the blade leading edge to a value of 0.2 chord lengths measured from trailing edge.
- See load cases 3 and 5 in 6-1-3/Table 3.

9.7.3 Maximum Blade Spindle Torque for CP-mechanism Design

Spindle torque, Q_{smax}, in kN-m (tf-m, Ltf-ft), around the spindle axis of the blade fitting is to be calculated for the load case described in 6-1-3/9.1. If these spindle torque values are less than the default value given below, the default value is to be used.

Default Value: $Q_{smax} = 0.25 \cdot F \cdot c_{0.7}$

where

$c_{0.7}$ = length of the blade chord at 0.7R radius, in m (m, ft)

F = either F_b or F_f, in kN (tf, Ltf) whichever has the greater absolute value.

9.7.4 Maximum Propeller Ice Torque Applied to the Propeller

Q_{max}, in kN-m (tf-m, Ltf-ft), is the maximum torque on a propeller due to ice-propeller interaction.

- when $D \le D_{limit}$:

$$Q_{max} = k_{ducted} \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot [n \cdot D]^{0.17} \cdot D^3$$

where

k_{ducted} = 10.4 (1.0605, 0.07923) for PC1 – PC5

k_{ducted} = 7.7 (0.785, 0.05866) for PC6 – PC7

- when $D > D_{limit}$:

$$Q_{max} = 1.9 \cdot k_{ducted} \cdot \left[1 - \frac{d}{D}\right] \cdot \left[H_{ice}\right]^{1.1} \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot [n \cdot D]^{0.17} \cdot D^{1.9}$$

where

D_{limit} = $1.8H_{ice}$ m (m, ft)

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

Propeller Type	*n*
CP propellers	n_n
FP propellers driven by turbine or electric motor	N_n
FP propellers driven by diesel engine	$0.85n_n$

where n_n is the nominal rotational speed at MCR, free running condition

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

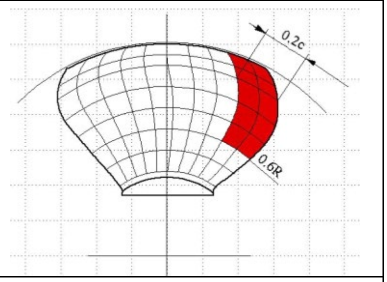
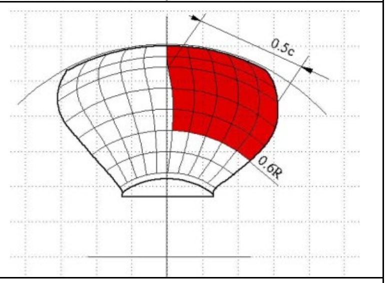
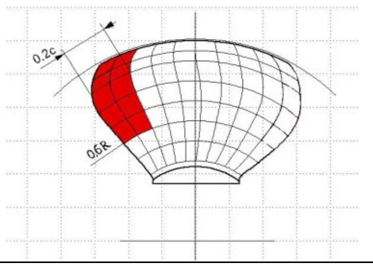
9.7.5 Maximum Propeller Ice Thrust (applied to the shaft at the location of the propeller)

The maximum propeller ice thrust in kN (tf, Ltf), (applied to the shaft at the location of the propeller) is:

T_f = $1.1 \cdot F_f$

T_b = $1.1 \cdot F_b$

TABLE 3
Load Cases for Ducted Propeller *(2012)*

	Force	*Loaded Area*	*Right handed propeller blade seen from back*
Load case 1	F_b	Uniform pressure applied on the back of the blade (suction side) to an area from 0.6*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 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	

9.9 Propeller Blade Loads and Stresses for Fatigue Analysis

9.9.1 Blade Stresses

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 6-1-3/9.5 and 6-1-3/9.7.

The peak stresses are those determined due to F_f and F_b. The peak stress range $\Delta\sigma_{max}$ and the maximum load amplitude F_{Amax} are determined on the basis of:

$$\Delta F_{max} = 2 \cdot F_{Amax} = |F_f| + |F_b|$$

9.11 Design Loads on Propulsion Line

9.11.1 Torque Excitation *(1 July 2019)*

The propeller ice torque excitation for shaft line dynamic analysis shall be described by a sequence of blade impacts which are of half sine shape and occur at the blade. The torque due to a single blade ice impact as a function of the propeller rotation angle is then:

$$Q(\varphi) = C_q \cdot Q_{max} \cdot \sin[\varphi(180/\alpha_i)] \qquad \text{when } \varphi = 0 \ldots \alpha_i$$

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

where C_q and α_i are parameters given in 6-1-3/Table 4 below.

TABLE 4
Parameters C_q and α_i (2012)

Torque Excitation	*Propeller-Ice Interaction*	C_q	α_i
Case 1	Single ice block	0.75	90
Case 2	Single ice block	1.0	135
Case 3	Two ice blocks with 45 degree phase in rotation angle	0.5	45

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

$$N_Q = c_6 \cdot H_{ice}$$

where

c_6 = 2 rev/s/m

= 0.6096 rev/s/ft

The number of impacts during one milling sequence for blade order excitation is $Z \cdot N_Q$.

In addition, the impacts are to ramp up over 270 degrees and subsequently ramp down over 270 degrees.

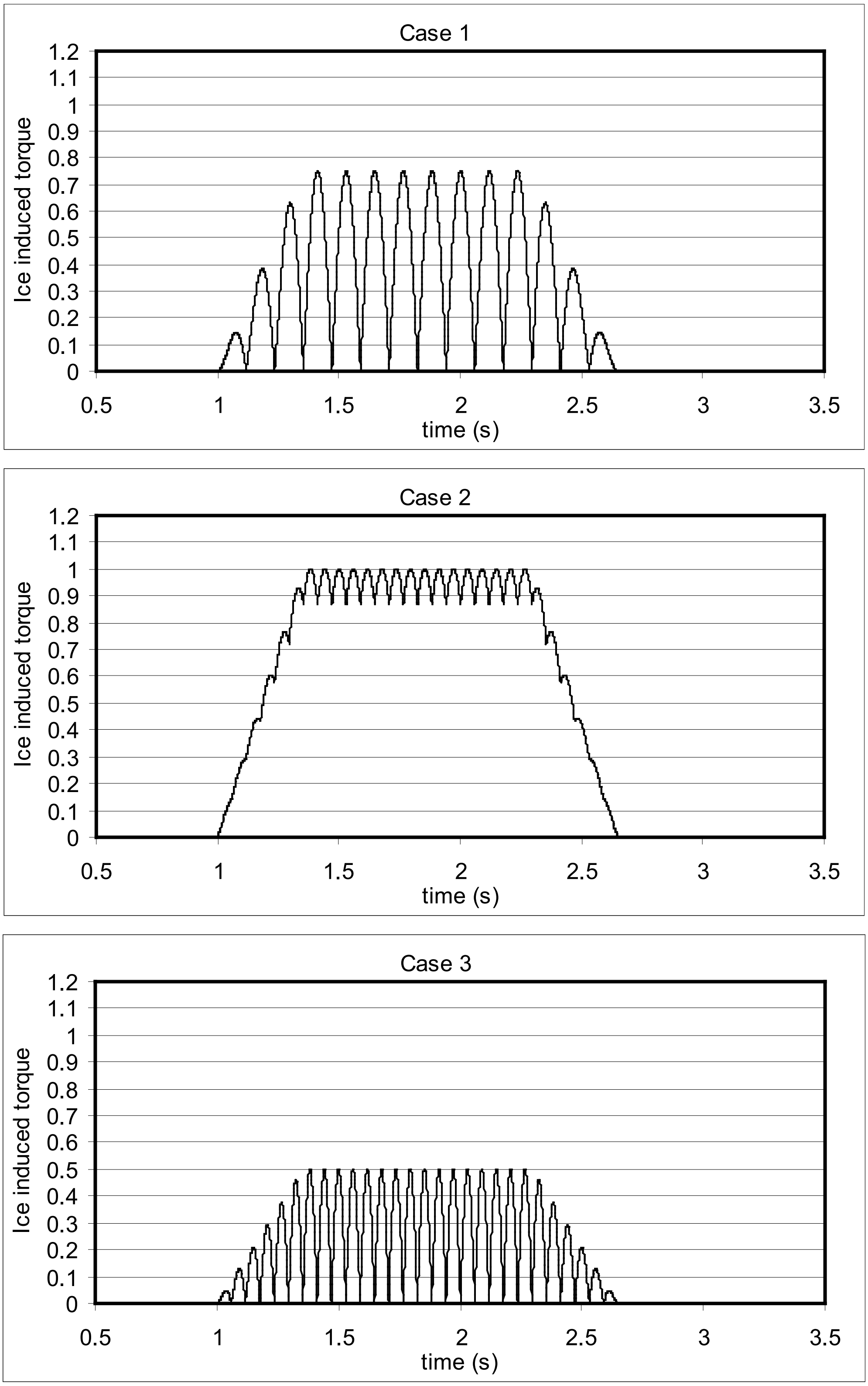
The total excitation torque from the 3 cases will then look like 6-1-3/Figure 1 below.

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

The response torque at any shaft component is to be analyzed considering excitation torque $Q(\varphi)$ at the propeller, actual engine torque, Q_e, and the mass elastic system.

Q_e = actual maximum engine torque at considered speed

FIGURE 1
Shape of the Propeller Ice Torque Excitation for 90° and 135° Single Blade Impact Sequences and 45° Double Blade Impact Sequence (Figures Apply for Propellers with Four Blades) *(2012)*



9.11.2 Response Torque in the Propulsion System *(1 July 2019)*

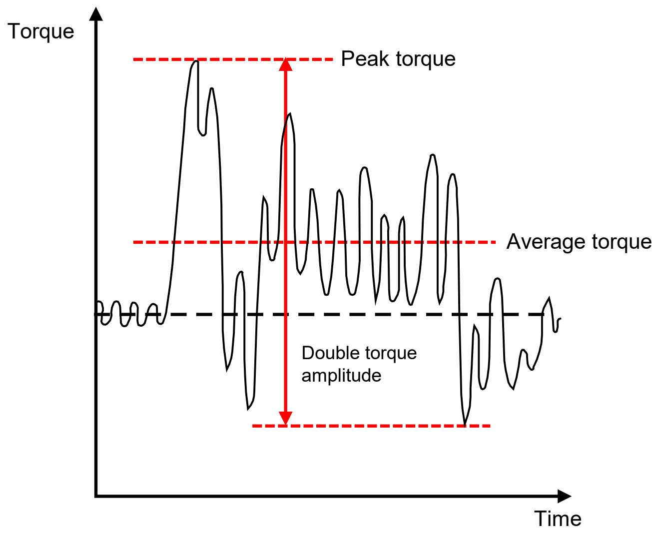
The response torque ($Q_r(t)$) in all components shall be determined by means of transient torsional vibration analysis of the propulsion line. Calculations are to be carried out for all excitation cases given above and the response is to be applied on top of the mean hydrodynamic torque in bollard condition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

The results of the 3 cases are to be used in the following way as illustrated in 6-1-3/Figure 2:

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

ii) 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 $Q_{A\max}$.

FIGURE 2
Definitions of Peak Torque and Torque Amplitude *(2012)*



Note: For transient torsional vibration analysis (time domain), the model should include the ice excitation at the propeller, the mean torque values provided by the prime mover and the hydrodynamic mean torque produced by the propeller as well as any other relevant excitations. The aim of torsional vibration calculations is to estimate the torsional loads for individual shaft line components in order to determine scantlings for safe operation.

9.11.3 Maximum Response Thrust

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

Maximum Shaft Thrust Forwards: $T_r = T + 2.2 \cdot T_f$

Maximum Shaft Thrust Backwards: $T_r = 1.5 \cdot T_b$

where

T = propeller bollard thrust, in kN (tf, Ltf)

T_f = maximum forward propeller ice thrust, in kN (tf, Ltf)

T_b = maximum backward propeller ice thrust, in kN (tf, Ltf)

If hydrodynamic bollard thrust, T, is not known, T is to be taken as given in 6-1-3/Table 5:

TABLE 5
Propeller Bollard Thrust *(2012)*

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

where

T_n = nominal propeller thrust at MCR at free running open water conditions, in kN (tf, Ltf)

For pulling type propellers ice interaction loads on propeller hub must be considered in addition to the above.

9.11.4 Blade Failure Load for both Open and Nozzle Propellers

The force is acting at $0.8R$ in the weakest direction of the blade and at a spindle arm of $^1/_3$ of the distance of axis of blade rotation of leading and trailing edge whichever is the greatest.

The blade failure load in kN (tf, Ltf) is:

$$F_{ex} = c_7 \cdot \frac{c \cdot t^2 \cdot \sigma_{ref}}{0.8 \cdot D - 2 \cdot r} \cdot 10^3$$

where

σ_{ref} = $0.6\sigma_{0.2} + 0.4\sigma_u$ in MPa (kgf/mm^2, psi)

σ_u = specified maximum ultimate tensile strength in MPa (kgf/mm^2, psi)

$\sigma_{0.2}$ = specified maximum yield or 0.2% proof strength in MPa (kgf/mm^2, psi)

c_7 = 0.3 (0.3, 1.9286E-5)

c = actual chord length in m (m, ft)

t = thickness, in m (m, ft), of the cylindrical root section of the blade at the weakest section outside root fillet, typically at the termination of the fillet into the blade profile

r = radius, in m (m, ft), of the cylindrical root section of the blade at the weakest section outside root fillet, typically at the termination of the fillet into the blade profile

σ_u and $\sigma_{0.2}$ are representative values for the blade material. Representative in this respect means values for the considered section. These values may either be obtained by means of tests, or commonly accepted "thickness correction factors" approved by the classification society. If not available, maximum specified values shall be used.

Alternatively the F_{ex} can be determined by means of FEA of the actual blade. Blade bending failure shall take place reasonably close to the root fillet end and normally not more 20% of R outside fillet. The blade bending failure is considered to occur when equivalent stress reach σ_{ref1} times 1.5 in elastic model.

11 Design

11.1 Design Principles *(1 July 2019)*

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

11.1.1 Fatigue Design in General

The design loads shall be based on the ice excitation and where necessary (shafting) dynamic analysis, as described by a sequence of blade impacts (6-1-3/9.11.1). The shaft response torque shall be determined by means of transient torsional vibration analysis of the propulsion line.

The components are to be designed so as to prevent accumulated fatigue failure when considering the loads according to 6-1-3/9.9 and 6-1-3/9.11 using the linear elastic Miner's rule.

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

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 $\sigma_{ref.}$ The load distribution (spectrum) is to be in accordance with the Weibull distribution.

11.1.2 Propeller Blades

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.

The blade stress amplitude distribution is therefore simplified (and at the same time disregarding mean stresses for fatigue purpose) and assumed to be as:

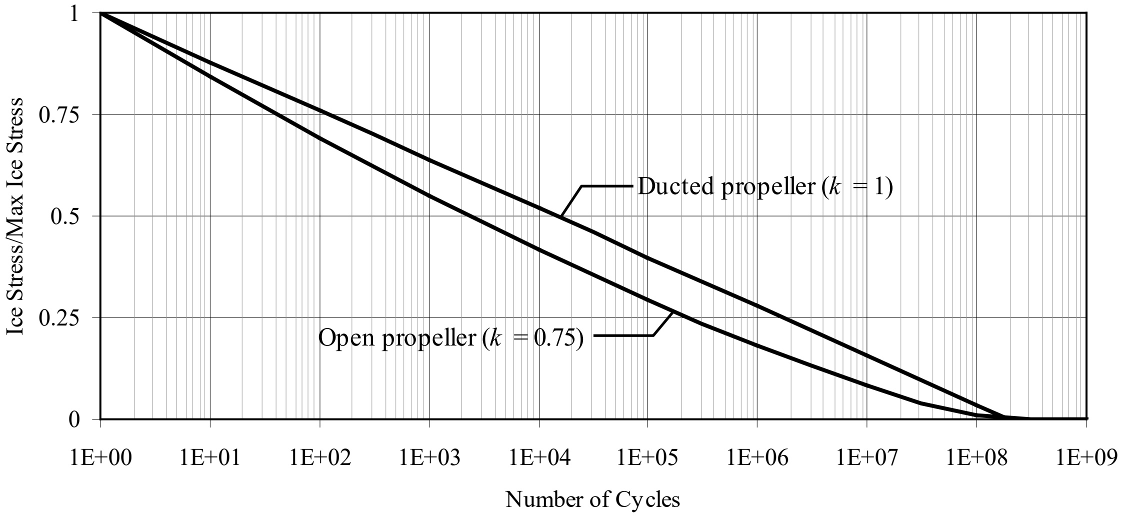
$$\sigma_A(N) = \sigma_{A\max} \cdot \left\{ 1 - \frac{\log(N)}{\log(N_{ice})} \right\}^{\frac{1}{k}}$$

where

k	=	Weibull exponent	
	=	0.75	for open propeller
	=	1.0	for nozzle propeller

This is illustrated in the cumulative stress spectrum in 6-1-3/Figure 3.

FIGURE 3
Ice Load Distribution for Ducted and Open Propeller *(2012)*



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

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

where

N_{class}	=	reference number of impacts per propeller rotation speed for each ice class as indicated in 6-1-3/Table 6		
k_1	=	1 for centre propeller		
	=	2 for wing propeller		
	=	3 for pulling propeller (wing and centre)		
	=	for pulling bow propellers number of load cycles is expected to increase in range of 10 times		
k_2	=	$0.8 - f$	when	$f < 0$
	=	$0.8 - 0.4 \cdot f$	when	$0 \leq f \leq 1$
	=	$0.6 - 0.2 \cdot f$	when	$1 < f \leq 2.5$
	=	0.1	when	$f > 2.5$
f	=	immersion function		
	=	$\frac{h_o - H_{ice}}{D/2} - 1$		
h_o	=	depth, in m (m, ft), of the propeller centerline at the minimum ballast waterline in ice (LIWL) of the ship.		

TABLE 6
Reference Number of Impacts Per Propeller Rotation Speed for Each Ice Class *(2012)*

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

11.1.3 Propulsion Line Components

The strength of the propulsion line shall be designed

i) For maximum loads in Subsection 6-1-3/9.5 and 6-1-3/9.7 (for open and ducted propellers respectively)

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

iii) With sufficient fatigue strength as determined by the following criteria:

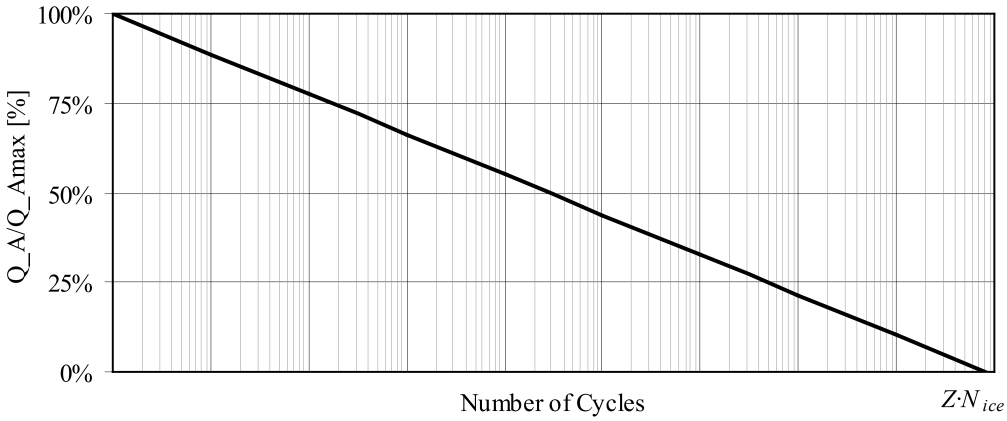
Cumulative fatigue calculations shall be made according to the Miner's rule.

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

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

This is illustrated by the example in 6-1-3/Figure 4.

FIGURE 4
Cumulative Torque Distribution *(2012)*



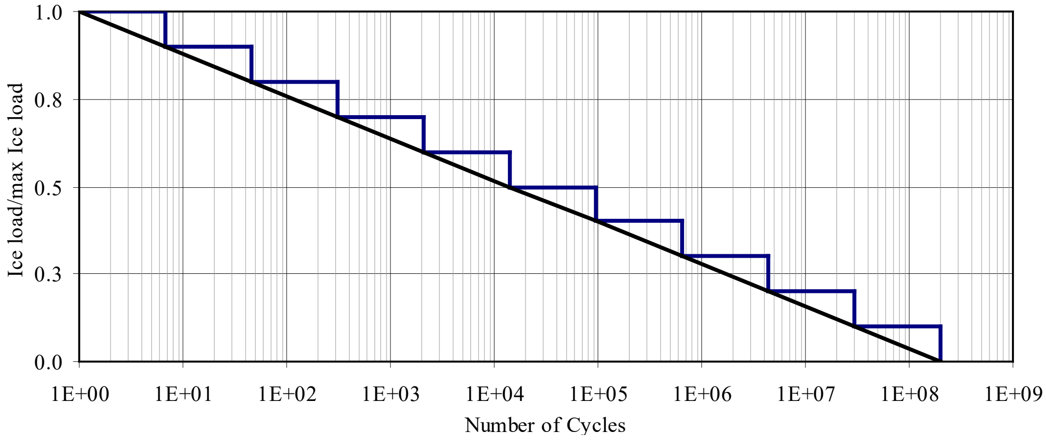
$Q_{A\max}$ is the average response torque shown in 6-1-3/Figure 2, calculated by means of transient torsional vibration analysis.

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

The Weibull exponent is $k = 1.0$ both for open propeller torque and for ducted propeller torque (and bending forces). The load distribution is an accumulated load spectrum, and the load spectrum is divided into minimum ten load blocks for the Miner summarizing method.

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

FIGURE 5
Example of Ice Load Distribution for the Shafting (k = 1), Divided Into Load Blocks *(2012)*



The load spectrum is divided into z-number of load blocks for the Miner summarizing method. The following formula can be used for calculation of the number of cycles for each load block.

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

where

i = single load block

z = number of load blocks

11.3 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 loading cases has to reflect the way of operation of the ship and the thrusters. In this respect, for example, the loads caused by the impacts of ice blocks on the propeller hub of a pulling propeller have to be considered. Furthermore, loads resulting from the thrusters operating at an oblique angle to the flow have to 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 has to stand the loads obtained when the maximum ice blocks, with the dimensions $H_{ice} \cdot 2H_{ice} \cdot 3H_{ice}$, 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 6-1-3/9.3.

11.3.1 Design Criteria for Azimuthing Propulsors

Azimuth propulsors shall be designed for following loads:

i) Ice pressure on strut based on defined location area of the strut / ice interaction as per 6-1-2/29.

ii) Ice pressure on pod based on defined location area of thruster body / ice interaction as per 6-1-2/29.

iii) Plastic bending of one propeller blade in the worst position (typically top-down) without consequential damages to any other part

iv) Steering gear design torque, Q_{SG}, in kN-m (tf-m, Ltf-ft), shall be minimum 60% of steering torque expected at propeller ice milling condition defined as $Q_{\max}$

$$Q_{SG} = 0.6 \cdot \left(\frac{Q_{\max}}{0.8R}\right) \cdot \ell$$

where

ℓ = distance from propeller plane to steering (azimuth) axis, in m (m, ft)

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

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

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

vi) Steering gear shall be ready for operation after above load, *v)a)* or *v)b)* has gone

11.5 Propeller Blade Design

11.5.1 Maximum Blade Stresses

Blade stresses (equivalent and principal stresses) are to be calculated using the backward and forward loads given in section 6-1-3/9.5 and 6-1-3/9.7. The stresses shall be calculated with recognized and well documented FE-analysis or other acceptable alternative method. The stresses on the blade shall not exceed the allowable stresses for the blade material given below.

Calculated blade equivalent stress for maximum ice load is to comply with the following:

$$\sigma_{calc} < \sigma_{all} = \sigma_{ref}/S$$

where

S = 1.5

σ_{ref} = reference stress, defined as:

= $0.7\sigma_u$ or

= $0.6\sigma_{0.2} + 0.4\sigma_u$, whichever is less

σ_u, $\sigma_{0.2}$ = minimum specified representative values for the blade material according to approved maker's specification

11.5.2 Blade Fatigue Design

Propeller blades are to be designed so as to prevent accumulated fatigue when considering the loads according to 6-1-3/11.1.2 and using the Miner's rule.

For simplification purpose it is permitted to arrange the blade stress distribution into a frequency spectrum having min. ten classes (every 10% load).

The S-N curve characteristics are based on two slopes, the first slope 4.5 is from 10^0 to 10^8 load cycles; the second slope 10 is above 10^8 load cycles.

i) The maximum allowable stress is limited by σ_{ref}/S

ii) The fatigue strength $\sigma_{Fat\text{-}E7}$ is the fatigue limit at 10 million load cycles.

The geometrical size factor, K_{size}, is:

$$K_{size} = 1 - a \cdot \ln\left(\frac{t}{C_8}\right)$$

where

a = given in 6-1-3/Table 7

t = actual blade thickness at considered section, in mm (mm, in.)

c_8 = 25 (25, 0.98425)

The mean stress effect (K_{mean}) is

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

The fatigue limit for 10 million load cycles is then:

$$\sigma_{E7} = \frac{\sigma_{Fat-E7}}{S} \cdot K_{size} \cdot K_{mean}$$

where

S = 1.5

The S-N curve is extended by using the first slope (4.5) to 100 million load cycles due to the variable loading effect.

$\sigma_{Fat\text{-}E7}$ can be defined from fatigue test results from approved fatigue tests at 50% survival probability and stress ratio $R = -1$.

TABLE 7
Mean Fatigue Strength, $\sigma_{Fat\text{-}E7}$, for Different Material Types *(2012)*

Bronze and Brass (a = 0.10)		*Stainless Steel (a = 0.05)*	
Type	$\sigma_{Fat\text{-}E7}$ *MPa (kgf/mm², psi)*	*Type*	$\sigma_{Fat\text{-}E7}$ *MPa (kgf/mm², psi)*
Mn-Bronze, CU1 (high tensile brass)	80 (8.158, 11603)	Ferritic (12Cr 1Ni)	120 (12.237, 17405)
Mn-Ni-Bronze, CU2 (high tensile brass)	80 (8.158, 11603)	Martensitic (13Cr 4Ni/13Cr 6Ni)	150 (15.296, 21756)
Ni-Al-Bronze, CU3	120 (12.237, 17405)	Martensitic (16Cr 5Ni)	165 (16.825, 23931)
Mn-Al-Bronze, CU4	105 (10.707, 15229)	Austenitic (19Cr 10Ni)	130 (13.256, 18855)

11.7 Blade Flange, Bolts and Propeller Hub and CP Mechanism

11.7.1 General

The blade bolts, the CP mechanism, the propeller boss, and the fitting of the propeller to the propeller shaft shall be designed to withstand the maximum and fatigue design loads, as defined in 6-1-3/9. The safety factor against yielding shall be greater than 1.3 and that against fatigue greater than 1.5. In addition, the safety factor for loads resulting from loss of the propeller blade through plastic bending, as defined in 6-1-3/9.11.4, shall be greater than 1 against yielding.

Blade bolts shall withstand following bending moment, M_{bolt}, in kN-m (tf-m, Ltf-ft) considered around bolt pitch circle, or another relevant axis for not circular joints, parallel to considered root section:

$$M_{bolt} = SF_{ex}\left(0.8\frac{D}{2} - r_{bolt}\right)$$

where

r_{bolt} = radius to the bolt plan, in m (m, ft)

S = 1.0

Blade bolt pre-tension shall be sufficient to avoid separation between mating surfaces with maximum forward and backward ice loads in 6-1-3/9.5 and 6-1-3/9.7 (open and ducted respectively).

Separate means (e.g. dowel pins) have to be provided in order to withstand a spindle torque resulting from blade failure Q_{sex} (6-1-3/9.11.4) or ice interaction $Q_{s\max}$ (6-1-3/9.7.3), whichever is greater. A safety of $S = 1$ is required. d in mm (mm, in.) is:

$$d = c_8 \sqrt{\frac{Q_s \cdot 8 \cdot \sqrt{3}}{PCD \cdot i \cdot \pi \cdot \sigma_{0.2}}}$$

where

c_8 = 1000 (1000, 163.95)

S = 1.3 for $Q_{s\max}$

= 1.0 for Q_{sex}

PCD = pitch circle diameter, in mm (mm, in.)

i = number of pins

Q_s = max $(SQ_{s\max}; SQ_{sex}) - Q_{fr1} - Q_{fr2}$ kN-m (tf-m, Ltf-ft)

Q_{sex} = $F_{ex} \frac{1}{3} L_{ex}$ kN-m (tf-m, Ltf-ft)

Q_{fr1} = friction torque in blade bearings caused by the reaction forces due to F_{ex}, in kN-m (tf-m, Ltf-ft)

Q_{fr2} = friction between connected surfaces resulting from blade bolt pretension forces, in kN-m (tf-m, Ltf-ft)

L_{ex} = maximum of distance from spindle axis to the leading, or trailing edge at radius 0.8R, in m (m, ft)

Friction coefficient = 0.15 may normally be applied in calculation of Q_{fr}.

The blade failure spindle torque Q_{sex} shall not lead to any consequential damages.

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

$$Q_{sa\max} = \frac{Q_{sb} + Q_{sf}}{2} \quad \text{kN-m (tf-m, Ltf-ft)}$$

Provided that calculated stresses duly considering local stress concentrations are less than yield strength, or maximum 70% of σ_u of respective materials, detailed fatigue analysis is not required. In opposite case components shall be analyzed for cumulative fatigue. Similar approach as used for shafting may be applied.

11.7.2 Servo Pressure

Design pressure for servo system shall be taken as a pressure caused by $Q_{s\max}$ or Q_{sex} when not protected by relief valves, reduced by relevant friction losses in bearings caused by the respective ice loads. Design pressure shall in any case be less than relief valve set pressure.

11.9 Propulsion Line Components *(1 July 2019)*

The main propulsion line's components (i.e. propulsion shafts, couplings etc.) are to be reviewed by applying the loads determined in 6-1-3/9.5.5, 6-1-3/9.7.5, 6-1-3/9.11.1, 6-1-3/9.11.2, 6-1-3/9.11.3 and 6-1-3/11.1.

The strength evaluation under the applied loads is to verify the loads corresponding to the propeller blade failure load shall not cause damage or deformation in the remaining propulsion line components.

The fatigue strength evaluation is to be based on the cumulative fatigue analyses according to Miner's Rule, as applicable. The applicable highest peak torque and the corresponding load spectrum are to be determined for each of the components or connections in question, as applicable.

The requirements in this section are complementary to those described in Section 4-3-2 of the Rules. The loads considered in this section do not need to be considered for shaft alignment or other calculations of normal operational conditions.

11.9.1 Propeller Fitting to the Shaft

11.9.1(a) Keyless Cone Mounting. The friction capacity (at 0°C (0°C, 32°F)) shall be at least 2.0 times the highest peak torque, Q_{peak}, as determined in 6-1-3/9.11.2, without exceeding the permissible hub stresses.

The necessary surface pressure in MPa (kgf/mm², psi) can be determined as:

$$p_{0°C} = c_9 \frac{2 \cdot 2.0 \cdot Q_{peak}}{\pi \cdot \mu \cdot D_S^2 \cdot L}$$

where

c_9	=	0.001 (0.001, 15.556)	
μ	=	0.14	for steel-steel
	=	0.13	for steel-bronze
D_S	=	is the shrinkage diameter at mid-length of taper, in m (m, ft)	
L	=	is the effective length of taper, in m (m, ft)	

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

11.9.1(b) Key Mounting. Key mounting is not permitted.

11.9.1(c) Flange Mounting.

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

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

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

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

v) The bolts are to be designed so that the blade failure load F_{ex} (6-1-3/9.11.4) does not cause yielding

11.9.2 Propeller Shaft

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

i) The blade failure load F_{ex} (6-1-3/9.11.4) applied parallel to the shaft (forward or backwards) shall not cause yielding. Bending moment need not to be combined with any other loads. The diameter d, in mm (mm, in.), in way of the aft stern tube bearing shall not be less than:

$$d = c_{10} \cdot \sqrt[3]{\frac{F_{ex} \cdot D}{\sigma_{0.2} \cdot \left(1 - \frac{d_i^4}{d^4}\right)}}$$

where

c_{10} = 160 (160, 48)

$\sigma_{0.2}$ = minimum specified yield or 0.2% proof strength of the propeller shaft material, in MPa (kgf/mm^2, psi)

d = propeller shaft diameter, in mm (mm, in.)

d_i = propeller shaft inner diameter, in mm (mm, in.)

Forward from the aft stern tube bearing the diameter may be reduced based on direct calculation of actual bending moments, or by the assumption that the bending moment caused by F_{ex} is linearly reduced to 50% 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 below the stresses due to the blade failure load.

ii) The stresses due to the peak torque, Q_{peak}, in kN-m (tf-m, Ltf-ft), shall have a minimum safety factor of 1.5 against yielding in plain sections and 1.0 in way of stress concentrations in order to avoid bent shafts.

Minimum diameter of:

Plain shaft: $$d = c_{11} \cdot \sqrt[3]{\frac{Q_{peak}}{\sigma_{0.2} \cdot \left(1 - \frac{d_i^4}{d^4}\right)}} \quad \text{mm (mm, in.)}$$

Notched shaft: $$d = c_{12} \cdot \sqrt[3]{\frac{Q_{peak} \cdot \alpha_t}{\sigma_{0.2} \cdot \left(1 - \frac{d_i^4}{d^4}\right)}} \quad \text{mm (mm, in.)}$$

where

c_{11} = 237 (237, 71)

c_{12} = 207 (207, 62)

α_t = the local stress concentration factor in torsion. Notched shaft diameter shall in any case not be less than the required plain shaft diameter.

iii) The torque amplitudes with the foreseen number of cycles shall be used in an accumulated fatigue evaluation where the safety factors are as defined in 6-1-3/11.1. If the plant also has high engine excited torsional vibrations (e.g. direct coupled 2-stroke engines), this has also to be considered.

iv) For plants with reversing direction of rotation the stress range $\Delta\tau \cdot \alpha_t$ resulting from forward Q_{peakf} to astern Q_{peakb} shall not exceed twice the yield strength (in order to avoid stress-strain hysteresis loop) with a safety factor of 1.5, i.e.:

$$\Delta\tau \cdot \alpha_t \le \frac{2 \cdot \sigma_y}{\sqrt{3} \cdot 1.5} \quad \text{MPa (kgf/mm}^2\text{, psi)}$$

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:

$$\sigma_F = 0.436 \cdot \sigma_{0.2} + 77 = \tau_F \cdot \sqrt{3} \quad \text{MPa (kgf/mm}^2\text{, psi)}$$

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 1.5.

The low cycle fatigue (LCF) representing 10^3 cycles is to be based on the lower value of either half of the stress range criterion [see *iv)*] or the smaller value of yield or 0.7 of tensile strength/$\sqrt{3}$. Both criteria utilize a safety factor of 1.5.

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.

11.9.3 Intermediate Shafts

The intermediate shafts are to be designed to fulfill the following:

i) The stresses due to the peak torque Q_{peak}, in kN-m (tf-m, Ltf-ft), shall have a minimum safety factor of 1.5 against yielding in plain sections and 1.0 in way of stress concentrations in order to avoid bent shafts.

Minimum diameter of:

Plain shaft: $$d = c_{11} \cdot \sqrt[3]{\frac{Q_{peak}}{\sigma_{0.2} \cdot \left(1 - \frac{d_i^4}{d^4}\right)}} \quad \text{mm (mm, in.)}$$

Notched shaft: $$d = c_{12} \cdot \sqrt[3]{\frac{Q_{peak} \cdot \alpha_t}{\sigma_{0.2} \cdot \left(1 - \frac{d_i^4}{d^4}\right)}} \quad \text{mm (mm, in.)}$$

where

c_{11} = 237 (237, 71)

c_{12} = 207 (207, 62)

α_t = local stress concentration factor in torsion.

$\sigma_{0.2}$ = minimum specified yield or 0.2% proof strength of the shaft material, in MPa (kgf/mm^2, psi)

d = shaft diameter, in mm (mm, in.)

d_i = shaft inner diameter, in mm (mm, in.)

ii) The torque amplitudes with the foreseen number of cycles shall be used in an accumulated fatigue evaluation where a minimum safety factor of 1.5 is required. If the plant also has high engine excited torsional vibrations (e.g., direct coupled 2-stroke engines), this has also to be considered.

iii) For plants with reversing direction of rotation the stress range $\Delta\tau \cdot \alpha_t$ resulting from forward Q_{peakf} to astern Q_{peakb} shall not exceed twice the yield strength (in order to avoid hysteresis) with a safety factor of 1.5, i.e.:

$$\Delta\tau \cdot \alpha_t \le \frac{2 \cdot \sigma_y}{\sqrt{3} \cdot 1.5} \quad \text{MPa (kgf/mm}^2\text{, psi)}$$

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:

$$\sigma_F = 0.436 \cdot \sigma_{0.2} + 77 = \tau_F \cdot \sqrt{3} \quad \text{MPa (kgf/mm}^2\text{, psi)}$$

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 and the required safety factor of 1.5.

The low cycle fatigue (LCF) representing 10^3 cycles is to be based on the lower value of either half of the stress range criterion [see *iii)*] or the smaller value of yield or 0.7 of tensile strength/$\sqrt{3}$. Both criteria utilize a safety factor of 1.5.

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.

11.9.4 Shaft Connections

11.9.4(a) Shrink Fit Couplings (Keyless). The friction capacity shall be at least 1.8 times the highest peak torque, Q_{peak}, in kN-m (tf-m, Ltf-ft), as determined in 6-1-3/9.11.2, without exceeding the permissible hub stresses.

The necessary surface pressure can be determined as:

$$p = c_9 \frac{2 \cdot 1.8 \cdot Q_{peak}}{\pi \cdot \mu \cdot D_S^2 \cdot L} \quad \text{MPa (kgf/mm}^2\text{, psi)}$$

where

c_9 = 0.001 (0.001, 15.556)

μ = 0.14 for steel to steel with oil injection (0.18 if glycerine injection)

D_S = is the shrinkage diameter at mid-length of taper, in m (m, ft)

L = is the effective length of taper, in m (m, ft)

11.9.4(b) Key Mounting. Key mounting is not permitted.

11.9.4(c) Flange Mounting.

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

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

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

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

v) The bolts are to be designed so that the blade failure load (6-1-3/9.11.4) in backwards direction does not cause yielding.

11.9.5 Gear Transmissions

11.9.5(a) 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). Maximum permissible deflection in order to maintain sufficient tooth contact pattern is to be considered for the relevant parts of the gear shafts.

11.9.5(b) Gearing (1 July 2019): The gearing shall fulfill following 3 acceptance criteria:

i) Tooth root stresses

ii) Pitting of flanks

iii) 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 twin input single output gears).

The load spectrum (see 6-1-3/11.1) 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 with higher speed the numbers of load cycles are found by multiplication with the gear ratios. The peak torque (Q_{peak}) is also to be considered.

Cylindrical gears can be assessed on the basis of the international standard ISO 6336 Pt.1–6, provided that "methods B" are used. Other acceptable alternative methods may also be considered on a case-by-case basis, provided that they are reasonably equivalent.

Bevel gears should be assessed on the basis of standards within the classification societies.

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

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 Pt 1, 2 and 6)

The scuffing safety (flash temperature method – ref. ISO-TR 13989) 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 society.

11.9.6 Clutches

Clutches shall have a static friction torque of at least 1.3 times the peak torque 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.

11.9.7 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.

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 log-log torque-cycle diagram where $T_{K\max1}$ respectively $\Delta T_{K\max}$ refer to 50,000 cycles and T_{KV} refer to 10^6 cycles. See illustration in 6-1-3/Figures 6, 7 and 8.

FIGURE 6
Log-log Torque-cycle Diagram Defining $T_{K\max 1}$ *(2012)*

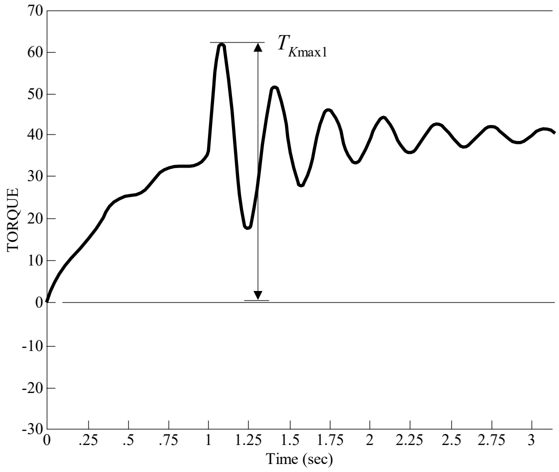


FIGURE 7
Log-log Torque-cycle Diagram Defining $\Delta T_{K\max}$ *(2012)*

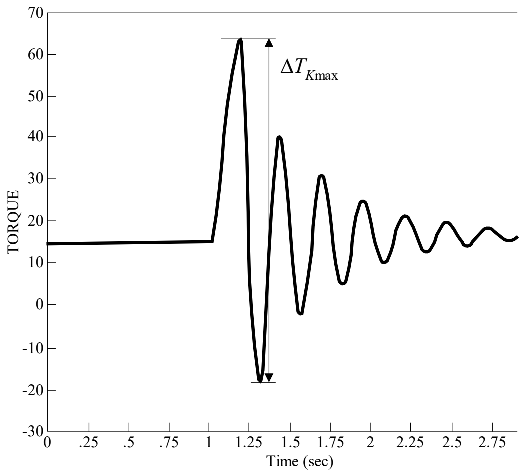
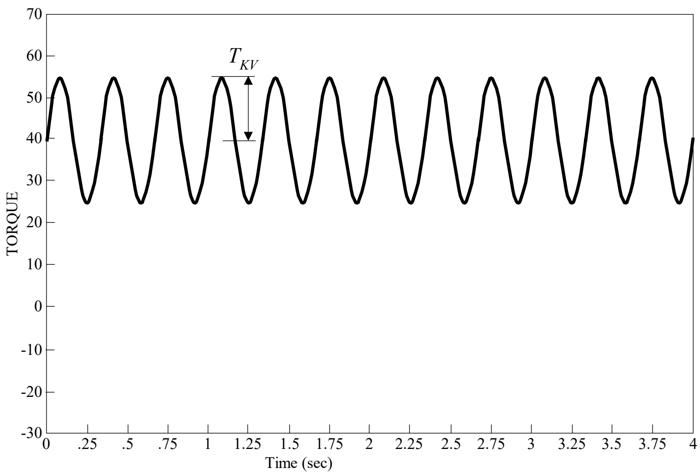


FIGURE 8
Log-log Torque-cycle Diagram Defining T_{KV} *(2012)*



11.9.8 Crankshafts

Special considerations apply for plants with large inertia (e.g., flywheel, tuning wheel or PTO) in the non-driving end of the engine.

11.9.9 Bearings

All shaft bearings are to be designed to withstand the propeller blade ice interaction loads according to 6-1-3/9.5 and 6-1-3/9.7. For the purpose of calculation, the shafts are assumed to rotate at rated speed. Reaction forces due to the response torque (e.g., in gear transmissions) are to be considered.

Additionally, the aft stern tube bearing as well as the next shaftline bearing are to withstand F_{ex} as given in 6-1-3/9.11.4, in such a way that the ship can maintain operational capability.

Rolling bearings are to have a L_{10a} lifetime of at least 40 000 hours according to ISO-281.

Thrust bearings and their housings are to be designed to withstand maximum response thrust 6-1-3/9.11.3 and the force resulting from the blade failure force F_{ex} in 6-1-3/9.11.4. 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 propeller hub.

11.9.10 Seals

Basic requirements for 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 are to be of proven design.

11.11 Prime Movers

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

11.11.2 Emergency Power Units

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

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

13 Machinery Fastening Loading Accelerations

13.1 General

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

13.3 Longitudinal Impact Accelerations

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

$$a_\ell = g_c \cdot (F_{IB}/\Delta) \cdot \left\{ \left[1.1 \cdot \tan(\gamma + \varphi)\right] + \left[7 \cdot \left(\frac{H}{L}\right)\right] \right\} \quad \text{m/s}^2 \text{ (m/s}^2\text{, ft/s}^2\text{)}$$

where

g_c = 1 (9.80665, 32.174)

13.5 Vertical Acceleration

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

$$a_v = g_c \cdot 2.5 \cdot (F_{IB}/\Delta) \cdot F_X \quad \text{m/s}^2 \text{ (m/s}^2\text{, ft/s}^2\text{)}$$

where

g_c	=	1 (9.80665, 32.174)	
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	

13.7 Transverse Impact Acceleration

Combined transverse impact acceleration, a_t, at any point along hull girder:

$$a_t = g_c \cdot 3 \cdot F_i \cdot \frac{F_X}{\Delta} \quad \text{m/s}^2 \text{ (m/s}^2\text{, ft/s}^2\text{)}$$

where

g_c	=	1 (9.80665, 32.174)	
F_X	=	1.5	at FP
	=	0.25	at midships
	=	0.5	at AP
	=	1.5	at AP for vessels conducting ice breaking astern
		intermediate values to be interpolated linearly	
ϕ	=	maximum friction angle between steel and ice, normally taken as 10, in degrees	
γ	=	bow stem angle at waterline, in degrees	
Δ	=	displacement in (tonnes, tonnes, Lton)	
L	=	length between perpendiculars, in m (m, ft)	
H	=	distance from the water line to the point being considered, in m (m, ft)	
F_{IB}	=	vertical impact force in kN (tf, Ltf), defined in 6-1-2/25.3	
F_i	=	total force in kN (tf, Ltf) normal to shell plating in the bow area due to oblique ice impact, defined in 6-1-2/5.5	

15 Auxiliary Systems

15.1 Machinery Protection

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.

15.3 Freezing

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

15.5 Vent and Discharge Pipes

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

17 Sea Inlets and Cooling Water Systems

17.1 Cooling Water Systems for Machinery

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

17.3 Sea Chests

At least two sea chests are to be arranged as ice boxes for Polar Class **PC1** to **PC5** inclusive where the calculated volume for each of the ice boxes shall be at least 1 m^3 (1 m^3, 35.314 ft^3) for every 750 kW (750 kW, 1005 HP) of the total installed power.

For Polar Classes **PC6** and **PC7**, at least one ice box for supplying water for cooling and fire-fighting purposes is to be connected to the cooling-water discharge by a branch pipe having the same cross sectional area as the main pipe-line, in order to stay free from ice and slush ice. As far as practicable, the sea inlet chest is to be situated well aft, adjacent to the keel, located preferably near the centerline.

17.5 Ice Boxes

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

17.7 Sea Inlet Valves

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

17.9 Vent Pipes

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

17.11 Sea Bays Freezing Prevention

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

17.13 Cooling Seawater Re-circulation

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.

17.15 Ice Boxes Access

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.

17.17 Openings in Vessel Sides

Openings in vessel 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 (20 mm, 0.787 in.). Gratings of the ice boxes are to be provided with a means of clearing. Clearing pipes are to be provided with screw-down type non return valves.

19 Ballast Tanks

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.

21 Ventilation System

21.1 Air Intakes Location

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

21.3 Air Intakes Heating

Accommodation and ventilation air intakes are to be provided with means of heating.

21.5 Machinery Air Intakes

The temperature of inlets air provided to machinery from the air intakes is to be suitable for the safe operation of the machinery

23 Steering Systems

23.1 General

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 6-1-2/29.

23.3 Rudder Actuator Holding Torque

The effective holding torque of the rudder actuator, at safety valve set pressure, is obtained by multiplying the open water requirement at design speed [maximum 9.26 m/s (9.26 m/s, 18 knots)] by the factors given in 6-1-3/Table 8.

TABLE 8
Rudder Actuator Holding Torque Multipliers *(2012)*

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

23.5 Torque Relief Arrangements

The rudder actuator is to be protected by torque relief arrangements, assuming the turning speeds given in 6-1-3/Table 9 without undue pressure rise (ref UR M42 for undue pressure rise):

TABLE 9
Assumed Turning Speeds for Torque Relief Arrangements *(2012)*

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

23.7 Fast Acting Torque Relief Arrangements

Additional fast acting torque relief arrangements (acting at 15% higher pressure than set pressure of safety valves in 6-1-3/23.5) are to provide effective protection of the rudder actuator in case of the rudder is pushed rapidly hard over against the stops assuming turning speeds given in 6-1-3/Table 10.

TABLE 10
Rudder Actuator Holding Torque Multipliers *(2012)*

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

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

25 Alternative Designs

As an alternative to 6-1-3, a comprehensive design study may be submitted and may be requested to be validated by an agreed test program.

PART

6

CHAPTER **1 Strengthening for Navigation in Ice**

SECTION **4 Requirements for Enhanced Polar Class Notation** *(1 July 2017)*

1 General

1.1 Application

Vessels that comply with the requirements of this Section, 6-1-1, 6-1-2, and 6-1-3 can be considered for a Polar Class notion as listed in 6-1-1/Table 2 followed by **Enhanced**. (e.g., **Ice Class PC3, Enhanced**)

3 Transverse Framing *(2014)*

3.1 Main and Intermediate Frames

3.1.1 Upper Ends of Frames

Main and intermediate frames are to extend up to the first deck or platform above the ice belt. They are to be welded and bracketed to the deck beams or to the deck longitudinals, as shown in 6-1-4/Figure 1a and 6-1-4/Figure 1b.

For ice classes **PC4** through **PC7**, where the lowest or only deck, or the lowest platform, is situated above the ice belt so that the distance between the deck, or platform, and the upper boundary of the ice belt exceeds "*d*" meters (feet), given in 6-1-4/Table 1, the upper ends of intermediate frames in the midbody and stern areas may terminate at a deep stringer situated at least 0.6 m (2 ft) above the ice belt.

For ice classes **PC6** and **PC7** in tween deck spaces, where the tween deck is 0.5 m (1.6 ft) or more above the upper ice waterline but within the ice belt, the upper ends of intermediate frames may terminate for ice class **PC6** and **PC7** at a stringer situated at least 0.5 m (1.6 ft) above the ice belt.

The upper ends of the frames terminated at a deep stringer are to be welded and bracketed to it as shown in 6-1-4/Figure 1c.

The intermediate frames terminated at an intercostal stringer or longitudinal are to be welded to it as shown in 6-1-4/Figure 1d.

TABLE 1
Distance *d*, m (ft) *(2012)*

Ice Class	*Where Web Frames are Fitted*
PC4	5.2 (17)
PC5	4.0 (13)
PC6	3.0 (10)
PC7	3.0 (10)

3.1.2 Lower Ends of Frames

Main and intermediate frames are to extend down to the inner bottom or to the double bottom margin plate. For ice classes **PC4** through **PC7**, the intermediate frames may terminate at a deck 1.0 m (3.3 ft) below the ice belt. The main and intermediate frames are to be attached and bracketed either to the inner bottom or to the double bottom margin plate or to the deck beams, or deck or to the stringer as shown in 6-1-4/Figure 2.

For vessels not having a double bottom, the intermediate frames are to extend down to a point below the top of the bottom transverses and are to terminate at an intercostal longitudinal. For ice classes **PC6** and **PC7**, the intermediate frames need not extend below the top of the floors, provided they terminate on an intercostal longitudinal not less than 0.8 m (2.6 ft) below the ice belt. The intermediate frames are to be attached to the bottom intercostal longitudinals.

3.1.3 Connection to Stringers and Decks

Main and intermediate frames are to be attached and bracketed to each supporting (deep) stringer, deck and deck beam within the ice belt.

FIGURE 1
Upper End Terminations of Frames *(2012)*

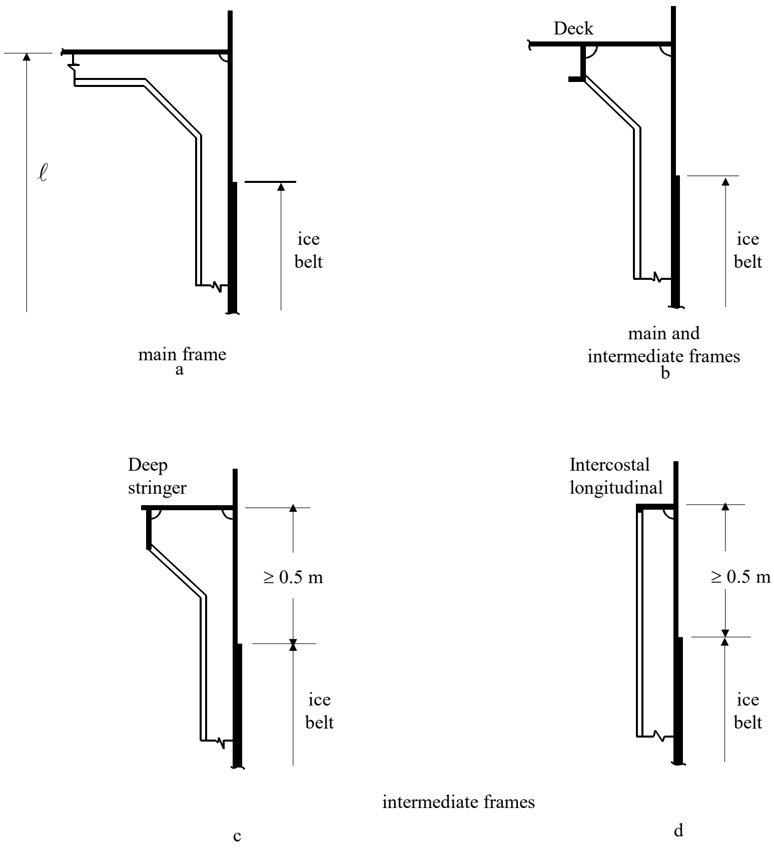
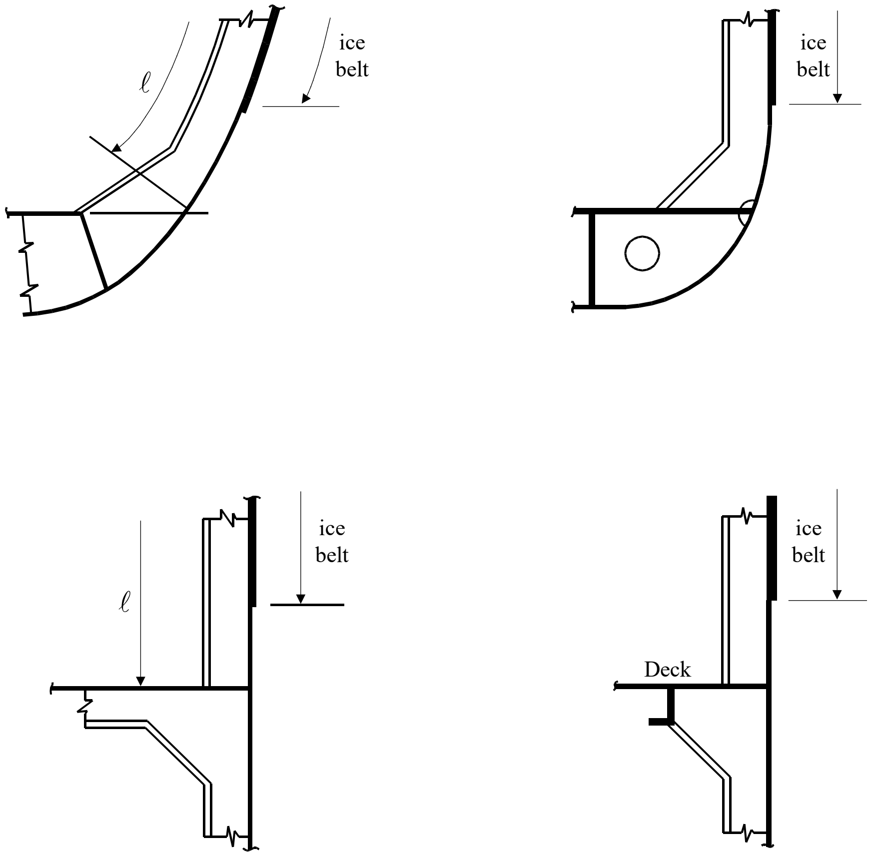


FIGURE 2
Lower End Terminations of Frames *(2012)*



3.3 Web Frames

The web frames are to be attached and bracketed to the solid floors and the beams at each ice deck.

3.5 Ice Stringers

3.5.1 Arrangements

Deep continuous or intercostal stringers are to be fitted within the ice belt throughout the length of the vessel. The spacing between adjacent stringers, or between the stringer and a deck or the inner bottom, measured along the shell is to be not more than indicated in 6-1-4/Table 2. One of the ice stringers is to be fitted about 200 to 400 mm (8 to 16 in.) below the upper ice waterline, if there is no deck in this area. For ice classes **PC1** through **PC7**, another stringer is to be fitted about 100 to 300 mm (4 to 12 in.) below the lower ice waterline, if there is no deck or similar support in this area.

TABLE 2
Maximum Stringer Spacing, m (ft) *(2012)*

Ice Class	*For Framing without Web Frames*	*System with Web Frames*
PC1 through **PC4**	1.5 (5)	2.1 (7)
PC5 through **PC7**	1.5 (5)	2.7 (9)

3.5.2 Scantlings and Connections *(2014)*

Where ice stringers are intercostal, the following criteria shall be met,

i) The intercostal stringers shall be fitted between frames and their scantlings are to be not less than those for main frames.

ii) The intercostal stringers are to be welded to the main and intermediate frames

iii) The web plate and the flange, or face, of intercostal ice stringers are to be attached to those of the main and intermediate frames.

iv) The intercostal stringers are to be bracketed to the bulkheads, side transverses, or web frames

Where deep ice stringers are fitted, the following criteria shall be met:

i) The shear area of the deep ice stringer within one frame space from the web frame is to be not less than that of the web frames.

ii) The depth of the ice stringer at the midspan between the web frames is to be not less than twice the depth of the main frame.

iii) The face, or flange, area of the deep stringer is to be not less than that of the web frame.

iv) The web plate and the face, or flange, of deep ice stringers are to be attached to those of the web frames.

v) *(1 July 2017)* The deep stringer referred to in 6-1-4/3.1.1 at which the upper ends of frames are terminated, is to have the scantlings as required in 6-1-4/3.5.

vi) The deep stringers are to be bracketed to the bulkheads or side transverses, so that the shear area at the bulkhead is twice that of the ice stringer web.

Stiffeners or tripping brackets are to be fitted as required in 3-2-6/3.7 and 3-2-6/3.9.

5 Longitudinal Framing *(2014)*

5.1 Struts *(1 July 2018)*

Where one or more struts are fitted as an effective supporting system for the ice belt structure, they are to be located within the ice belt and spaced so as to divide the supported web into spans of approximately equal length. Inboard ends of the struts are to be supported sufficiently by longitudinal bulkhead transverses having a section modulus not less than 0.9 of that required by 6-1-2/15.3. The sectional area of the strut is to be obtained from the following equation:

$$A = (bs_1/K)(P/\sigma_y)K_o \quad \text{cm}^2 \text{ (in}^2\text{)}$$

where

b = as defined in 6-1-2/5.9 for particular area of the ice belt, in m (ft)

s_1 = distance between web frames in mm (in.) measured along lower ice waterline in way of compartment being considered

K = $0.04 - 0.0175(\ell/r)$ for SI & MKS units

= $0.0333 - 0.00175(\ell/r)$ for US units

ℓ = unsupported span of the strut, m (ft)

r = least radius of gyration, cm (in.)

P = $C_1 P_{ave} AF$

C_1 = 0.60 for bow area as defined in 6-1-2/3

= 0.50 for all other areas

P_{ave} = as defined in 6-1-2/5.11 for particular area of the ice belt

AF = Hull Area Factor from 6-1-2/Table 4, 6-1-2/Table 5, 6-1-2/Table 6 or 6-1-2/Table 7

σ_y = minimum upper yield stress of the material, in N/mm^2, but not greater than 690 N/mm^2 (70 kgf/mm^2, 100 ksi)

K_o = $(2.44/\ell)^{1/2}$ (ℓ in m)

= $(8/\ell)^{1/2}$ (ℓ in ft), but not less than 0.4

7 Peak Frames *(2014)*

Main and intermediate frames in forepeaks are to extend down to the floors or the bottom transverses or the stem. The section modulus of each peak frame is to be as given in 6-1-2/11.5. The spacing between the deep ice stringers or platforms measured along the shell is to be not more than 1.5 m (5 ft) for forepeaks of ice classes **PC1** through **PC4**. For the forepeaks of ice classes **PC5** through **PC7**, the distance is to be not more than 2.1 m (7 ft).

For ice classes **PC1** through **PC4**, transverse peak frames are to be fitted so that the angle between the web of the transverse frame and the shell plating, φ_w, is not less than 40 degrees at any waterline within the ice belt. If this angle is less than 60 degrees, the section modulus of the transverse peak frames is to be increased by the factor.

$K = 2\cos\varphi_w$ where 40 degrees $\leq \varphi_w \leq$ 60 degrees

For all ice classes, the intermediate frames are to extend down to the bottom structure and up to the first deck above the ice belt

9 Double Bottom

9.1 Inner Bottom

An inner bottom is to be fitted between the peaks in all vessels of ice classes **PC1** to **PC3** and in **PC4** ice class vessels of lengths of 61 m (200 ft) and over.

9.3 Transversely Framed Bottom

For ice classes **PC1** through **PC5**, solid floors are to be fitted at each web frame along the length of the vessel, and, in addition, at each main frame within the bow, lower intermediate and lower stern areas of the ice belt. Spacing of the solid floors is to be not more than required by 3-2-4/5 or the appropriate sections of Part 5, as applicable. Open floors or bilge brackets extending to longitudinals or side girders are to be fitted at each intermediate frame that extends to the inner bottom. The distance between bottom side girders is to be not more than 2.4 m (8 ft) for the bow area of ice classes **PC1** through **PC3** and 3.0 m (10 ft) elsewhere for ice classes **PC1** through **PC5**. Spacing of the side girders is to be not more than required by 3-2-4/3.7.

9.5 Longitudinally Framed Bottom

For ice classes **PC1** through **PC5**, solid bottom transverses or solid floors are to be fitted at each web frame along the length of the vessel, but at not more than 1.8 m (6 ft) within the bow, lower intermediate and lower stern areas of the ice belt. Spacing of the solid floors is to be not more than required by 3-2-4/5 or the appropriate sections of Part 5, as applicable. Special consideration will be given to wider spacings.

Open floors or bilge brackets extending to the outboard longitudinals are to be fitted throughout at each frame that extends to the inner bottom. The spacing of the bottom longitudinals within the bow, lower intermediate and lower stern areas of the ice belt is to be not more than 0.6 m (2 ft) for ice classes **PC1** through **PC3** and 0.7 m (2.3 ft) for ice classes **PC4** through **PC7**.

11 Ice Decks

11.1 General

The following requirements apply to decks or parts of decks situated within the ice belt as defined in 6-1-1/5.1. For vessels not having decks in the ice belt and for vessels of ice classes **PC1** through **PC4** having only one deck in the ice belt, the following requirements apply also to decks or parts of decks above and below the ice belt to which the main and intermediate frames extend.

11.3 Deck Plating *(1 July 2017)*

The thickness of the stringer plate is to be not less than:

$$t = k(s^2 b\, P)^{1/3} \quad \text{mm (in.)}$$

where

k	=	0.12 (0.257, 0.00523)
s	=	distance between the deck beams, in mm (in.)
b	=	as defined in 6-1-2/5.9, in m (ft), for the particular area of the ice belt
P	=	$C_1 P_{ave} AF$
C_1	=	0.60 for bow area as defined in 6-1-2/3
	=	0.50 for all other areas
P_{ave}	=	as defined in 6-1-2/5.11 for the particular area of the ice belt
AF	=	Hull Area Factor from 6-1-2/Table 4, 6-1-2/Table 5, 6-1-2/Table 6 or 6-1-2/Table 7 for the particular area of the ice belt

The width of the stringer plate is to be not less than five times the depth of the main frame for ice classes **PC1** and **PC2** and four times the main frame depth for **PC3** to **PC7** ice classes. For ice classes **PC1** through **PC7**, the thickness of the deck plating is to be not less than 0.75 times the required thickness of the stringer plate.

11.5 Deck Transverses and Deck Beams

11.5.1 Transversely Framed Decks

Partial beams or brackets are to be fitted at every intermediate frame for ice classes **PC1** to **PC5**. These partial beams or brackets are to be extended from the frames to a deck longitudinal or deck girder. The length of these partial beams or brackets is to be not less than the width of the stringer plate.

11.5.2 Longitudinally Framed Decks

Deck transverses are to be fitted at every web frame and, in addition, not less than at every second main frame for ice classes **PC1** to **PC4**, at every third main frame for ice classes **PC5** to **PC7**.

Partial beams or brackets are to be fitted at all other main frames and at every intermediate frame for ice classes **PC1** to **PC7.** The partial beams or brackets are to be extended from the frames to a deck longitudinal or deck girder situated not less than 1.5s from the inboard edge of the frames, where s is as defined in 6-1-4/11.3.

11.5.3 Scantlings *(1 July 2018)*

The sectional area of the beams and deck transverses is to be not less than:

$$A = K_1 sb(P/\sigma_y)\cos\beta \quad \text{cm}^2$$

$$A = 1.2K_1 sb(P/\sigma_y)\cos\beta \quad \text{in}^2$$

The moment of inertia of the beams is to be not less than:

$MI = kK_2 s\ell^2 bP\cos\beta$ cm^4 (in^4)

where

k	=	1.0 (9.81, 0.1191)	
P	=	$C_1 P_{ave} AF$	
C_1	=	0.60	for bow area as defined in 6-1-2/3
	=	0.50	for all other areas
P_{ave}	=	as defined in 6-1-2/5.11, in N/mm^2 (kgf/mm^2, ksi), for the particular area of the ice belt	
AF	=	Hull Area Factor from 6-1-2/Table 4, 6-1-2/Table 5, 6-1-2/Table 6 or 6-1-2/Table 7	
b	=	as defined in 6-1-2/5.9, in m (ft), for the particular area of the ice belt	
s	=	distance between the beams, in mm (in.)	
ℓ	=	the span of the beam, measured in m (ft), between the inboard edge of the frame and the deck longitudinal or deck girder supporting the beam	
σ_y	=	minimum upper yield stress of the material, in N/mm^2, but not greater than 690 N/mm^2 (70 kgf/mm^2, 100 ksi)	
β	=	as defined in 6-1-2/5.5, in degrees, for the particular area of the ice belt	
K_1	=	8.5	for ice classes **PC1** to **PC5**
	=	6.6	for ice classes **PC6** and **PC7**
K_2	=	0.24	for ice classes **PC1** to **PC5**
	=	0.13	for ice classes **PC6** and **PC7**

The sectional area and the moment of inertia of the partial beams and of the brackets are to be not less than required above. The beams and the partial beams are to be bracketed to the deck longitudinals or deck girders. Beams or partial beams or brackets fitted at the web frames are to be reinforced so that their section modulus, *SM* is to be not less than:

$SM = K_3 SM_{wf} \ell_{wf} / \ell$ cm^3 (in^3)

where

SM_{wf}	=	section modulus of the web frame in cm^3 (in^3)	
ℓ_{wf}	=	span of the web frame, measured in m (ft), between supports, with no reduction for fitted end brackets, if any	
K_3	=	0.8	for ice classes **PC1** through **PC5**
	=	0.5	for ice classes **PC6** and **PC7**

When calculating the section modulus and the moment of inertia of a framing member, net thicknesses of the web, flange (if fitted) and attached shell plating are to be used.

11.7 Decks with Wide Openings *(1 July 2017)*

Within the bow intermediate and midbody areas of the ice belt, the cross sectional area of the deck outside the line of openings is to be not less than:

$$A = Kb\ell(P/\sigma_y)\cdot 10^3 \quad \text{cm}^2$$

$$A = 14.4Kb\ell(P/\sigma_y) \quad \text{in}^2$$

where

K	=	8.2	for ice classes **PC1** to **PC5**
	=	6.2	for ice classes **PC6** and **PC7**

b = as defined in 6-1-2/5.9, in m (ft), for the particular area of the ice belt

ℓ = the length of the opening, in m (ft), but need not be taken as more than $0.1L$

P = $C_1 P_{ave} AF$

C_1	=	0.60	for bow area as defined in 6-1-2/3
	=	0.50	for all other areas

P_{ave} = as defined in 6-1-2/5.11, for the particular area of the ice belt

AF = Hull Area Factor from 6-1-2/Table 4, 6-1-2/Table 5, 6-1-2/Table 6 or 6-1-2/Table 7

σ_y = as defined in 6-1-4/11.5.3

L = as defined in 6-1-1/5.7, in m (ft)

13 Bulkheads

13.1 General

For ice classes **PC1** to **PC5**, those parts of transverse bulkheads situated within the ice belt are not to be vertically corrugated.

13.3 Scantlings *(1 July 2017)*

For ice classes **PC1** to **PC7**, the thickness of that part of the bulkhead adjacent to the side shell and within the ice belt is to be not less than the thickness of the adjacent frames or of the stringers connected to the bulkhead, whichever is greater. The width of these parts of the bulkhead is to be not less than shown in 6-1-4/Table 3. These parts of the bulkhead adjacent to the shell within the ice belt are to be fitted with stiffeners normal to the shell plating. The stiffeners are to be welded to a vertical bulkhead stiffener and welded and bracketed to the side longitudinals. Where the shell is transversely framed, brackets are to be welded to the shell and extended and attached to adjacent frames.

TABLE 3
Minimum Width of Reinforced Bulkhead Plating *(2012)*

Ice Class	*Area of the Ice Belt*			
	Peak Bulkheads m (ft)	*Bow and Bow Intermediate Areas m (ft)*	*Midbody Area m (ft)*	*Stern Area m (ft)*
PC1 through **PC4**	1.6 (5.2)	1.4 (4.6)	1.2 (4.0)	1.4 (4.6)
PC5 through **PC7**	1.2 (4.0)	1.2 (4.0)	1.0 (3.3)	1.0 (3.3)

If a vessel is intended to operate astern in ice regions, the width of the reinforced parts of the bulkhead adjacent to the Stern and Stern Intermediate ice belt areas is to be not less than that required for Bow and Bow Intermediate Areas shown in 6-1-4/Table 3.

15 Stem and Stern Frames

15.1 General *(2014)*

The requirements of Section 3-2-13 of the Rules are to be complied with. The stem and stern frame for ice class **PC1** through **PC5**, and for ice class **PC6** and **PC7** vessels of displacements more than 50,000 tonnes (49,200 Lt), are to be constructed of rolled bar, cast or forged steel. Shaped plate stem may be used for **PC6** and **PC7** vessels of and less than 50,000 tonnes (49,200 Lt). The shaped plate stem used in other cases is to be specially considered. All joints and connections are to fully develop the strength of the stem and stern frame. All rudders are to be protected against ice impacts for going astern.

15.3 Stem

15.3.1 Solid Stem

The cross sectional area of a stem made of rolled bar, cast or forged steel from the center vertical keel to 0.01L above the ice belt is to be not less than:

$$A = K_1 D^{1/3} (L - 61) + A_o \quad \text{cm}^2$$

$$A = 0.0473 K_1 D^{1/3} (L - 200) + A_o \quad \text{in}^2$$

where

K_1 and A_o = as given in 6-1-4/Table 4

D = as defined in 6-1-1/5.5

L = as defined in 6-1-1/5.7, in m (ft), but is not to be taken less than 61 m (200 ft)

For vessels of displacements less than 2,500 tonnes (2,460 Lt) the cross sectional area given by the above equation may be reduced 10%. The cross sectional area of the stem above the ice belt may be reduced gradually to the value given in Section 3-2-13.

TABLE 4
Solid Stem Bar Coefficients *(2012)*

Ice Class	A_o *cm² (in²)*	K_1
PC1	750 (116.2)	0.28
PC2	750 (116.2)	0.28
PC3	700 (108.5)	0.27
PC4	500 (77.5)	0.24
PC5	200 (31.0)	0.18
PC6	62 (9.6)	0.13
PC7	62 (9.6)	0.13

15.3.2 Shaped Plate Stem *(1 July 2018)*

Thickness of shaped plate stems within the bow area of the ice belt is to be not less than

$$t = 0.8s(P/\sigma_y)^{1/2} + t_s \quad \text{but not less than } 0.04R.$$

where

t = required thickness of plate stem, in mm (in.)

s = distance between frames, brackets (breast hooks) or stiffeners, in mm (in.)

P = $0.75P_{bow}$, as defined in 6-1-2/5.5

σ_y = minimum upper yield stress of the material, in N/mm², but not greater than 690 N/mm² (70 kgf/mm², 100 ksi)

t_s = corrosion/abrasion addition for the bow area, as defined in 6-1-2/21, in mm

R = the inside radius of the stem at the given section, in mm (in.). Need not be taken greater than 625 mm (24.6 in.) for ice classes **PC6** and **PC7**

At any section, the fore and aft length of the stem plate is to be not less than $15t$.

15.3.3 Arrangement

The outer surface of connections of the shell plating to the stem is to be flush. The stem is to be supported by floors, webs, frames, breasthooks or brackets spaced not more than 610 mm (24 in.). In addition, shaped plate stems are to be supported on the centerline by a plate, web or bulkhead having the same thickness as the center vertical keel and a width not less than 610 mm (24 in.).

15.5 Stern Frame

The stern post is to be of size obtained from 3-2-13/3.5 through 3-2-13/3.11, with all thicknesses increased by coefficient K, as given in 6-1-4/Table 5. In addition, factors C_f and C_c in 3-2-13/3.5 are to be multiplied by K^2.

TABLE 5
Stern Post Coefficient *(2012)*

Ice Class	*K*
PC1	2.0
PC2	1.9
PC3	1.8
PC4	1.6
PC5	1.4
PC6	1.2
PC7	1.2

17 Towing Arrangements

17.1 Bow

Polar Class vessels intended to be escorted by a higher ice class leading vessel, are to be fitted with a tow chock pipe and a tow bitt on the bow. The chock and the bitt are to be properly connected to the stem frame. The portions of the decks at which the chock and the bitt are attached are to meet requirements of 6-1-4/11. The shell plating and framing below and 1.5 m (5 ft) around the chock are to be as required by 6-1-2/7, 6-1-2/9 and 6-1-2/11 for the bow area of the ice belt for ice classes **PC6** and **PC7** and for the intermediate area of the ice belt for ice classes **PC2** through **PC5** and where the corrosion and abrasion allowance, t_s, is as given in 6-1-4/Table 6. The stem frame below the connections with the chock is to be as required by 6-1-4/15.3 for the portion of the stem within the ice belt.

TABLE 6
Corrosion/Abrasion Additions for Shell Plating Around Chock *(2012)*

Hull Area	*t_s, mm*					
	With Effective Protection			*Without Effective Protection*		
	PC1 - PC3	**PC4 & PC5**	**PC6 & PC7**	**PC1 - PC3**	**PC4 & PC5**	**PC6 & PC7**
Shell plating below and 1.5 m (5 ft) around the chock	1.0	1.0	1.0	3.0	2.0	1.5

Where a bulbous bow is fitted, the bulb is not to extend beyond the fore end of the lower ice waterline specified by 6-1-1/5.3.

17.3 Stern

Vessels of ice classes **PC1** through **PC6** intended to be used as leading vessels assisting passage of a lower ice class vessel as listed in 6-1-1/Table 2 are to be equipped with a towing system. Both the arrangement of the towing system and the shape of the stern are to be suitable for towing the assisted vessel in immediate contact. The portion of the upper deck which the towed vessel may contact is to be as required by 6-1-4/11. The shell plating and framing adjacent to this portion of the upper deck are to be as required by 6-1-2/7, 6-1-2/9 and 6-1-2/11 for the stern area of the ice belt.

19 Machinery Arrangements

19.1 Propulsion Arrangements

In addition to the regular governor, all propulsion engines and turbines are to be fitted with a separate overspeed device so adjusted that the speed cannot exceed the maximum rated speed by more than 20%.

19.3 Electric Propulsion

Propulsion motors are to be fitted with automatic protection against excessive torque, overloading and temperature. This protection is to automatically limit these parameters, but is not to cause loss of propulsion power.

19.5 Boilers

Vessels propelled by steam machinery are to be fitted with at least two boilers of equal capacity.

19.7 Protection Against Excessive Torques

For vessels of all classes, if torsionally flexible couplings or torque-limiting devices are fitted in the propulsion system, positive means are to be provided for transmitting full torque to the propeller in the event of failure of the flexible element. In addition, for vessels of classes **PC1** through **PC4**, couplings of the elastomer-in-shear type are not to be fitted in those portions of the propulsion system which are subject to shock loading from the propeller.

19.9 Propeller Arrangements

Propeller arrangements, the shape of the stern and the propeller protecting structures are to be adequate for the intended service. Special consideration is to be given to the propeller protection when moving astern. For **PC1** through **PC5** ice class vessels, the following condition is to be complied with.

$$0.5B_x - b_x \geq kd$$

where

B_x	=	breadth of the lower ice waterline, as defined in 6-1-1/5.3, at the hull section in way of the propeller tips, in m (ft)	
b_x	=	distance from the vessel centerline to the outermost propeller blade tip, in m (ft)	
k	=	0.25	for open propellers
	=	0.10	for ducted propellers
d	=	propeller diameter, in m (ft)	

19.11 Tunnel Thrusters

The mechanical components of a tunnel thruster (i.e., propellers, gears, shafts, couplings, etc.) are to meet the applicable requirements of Section 4-3-5, for a theoretical input torque of twice (2) the prime mover output torque in order to simulate, in a conservative way, the effect of ice on all the torque transmitting components.

Alternatively, a comprehensive study to determine the effect of ice – propeller interaction and the resultant ice torque may be considered. In this way the mechanical components of the tunnel thruster are to meet the applicable requirements of the present Section using the so determined ice-torque.

19.13 Cooling Water Arrangements

The following apply to vessels of ice classes **PC1** through **PC5**.

19.13.1 Sea Bay or Tank

The suctions for cooling water for all machinery essential to the propulsion of the vessel and for fire-fighting purposes are to be taken from a sea bay or tank located as close as practicable o the keel. The sea bay or tank is to be supplied with water from at least two independent sea suctions with at least one on each side of the hull. The area of each sea suction opening is to be not less than six times the total cross-sectional area of all pump suctions connected to the sea bay.

19.13.2 Sea Suctions

Suitable strainers are to be provided between the sea suctions and the sea bay. Valves are to be provided to permit isolation of the strainers, both from the sea suctions and from the sea bay. The cross-sectional area of such valves and strainers and associated piping for each sea suction is not to be less than the total cross-sectional area of all pump suctions connected to the sea bay.

19.13.3 Sea Water Pumps

Each sea water pump serving machinery essential to the propulsion of the vessel is to draw sea water directly from the sea bay. Design flow velocity in any suction line is not to exceed 2 m (6.6 ft) per second.

19.13.4 Cooling Water Recirculation

The discharge line from the cooling system is to be provided with suitable piping, valves and fittings to permit the discharge flow to be recirculated. The recirculation piping is to connect with the suction piping at a point on the seaward side of the strainer sea shut-off valves.

Piping, valves and fittings for the recirculation line are to be of at least the same cross-sectional area as the overboard discharge line.

19.15 Starting-air System

For vessels of Ice Class **PC1** through **PC5**, in addition to the applicable requirements of 4-6-5/9, starting-air systems are to comply with the following.

i) At least two independently driven starting-air compressors are to be provided. The total capacity of the compressors is to be sufficient to charge the air receiver from empty to maximum pressure in not more than 30 minutes.

ii) The smallest of the starting air compressors is to have not less than two-thirds the capacity of the largest.

21 Power of Propulsion Machinery

For Polar Classes **PC1** through **PC7**, the total ahead power delivered to the propellers, is to be sufficient for the vessel to maintain a design service speed under the ice conditions described in 6-1-1/Table 2, as related to the appropriate vessel notation.

An appropriate analytical approach or ice model testing results, are to be submitted for review. Where the design is in an early stage or ice model testing is not planned, the requirement for minimum power/astern power, as specified in this section of the Rules, may be used for an assessment of power of propulsion machinery, unless otherwise any specific methodology is provided by the cognizant authorities having jurisdiction over the water in which the vessel is intended to operate. The requirements of the cognizant authorities or administrations may also need to be recognized or complied with.

21.1 Minimum Powering Criteria

The total propulsion power delivered to the propellers is recommended to satisfy either of two criteria, namely:

i) The thickness of consolidated level ice passable by a vessel of ice classes **PC1** though **PC7** in stable continuous icebreaking is to be as defined in 6-1-4/21.3

ii) The total power delivered to the propellers at the maximum continuous rate has to be as defined in 6-1-4/21.5

21.3 Maximum Thickness of Consolidated Level Ice

The maximum thickness, h_{max}, of consolidated level ice (in the absence of wind/current driven ice compressions) passable in stable continuous icebreaking is not to be less than the value of h_0 given in 6-1-4/Table 7, i.e.:

$$h_{max} \geq h_0$$

The value of h_{max} can be determined at design stages by the following formula:

$$h_{max} = f_u f_s f_p \frac{\left(N_p d_{pr}\right)^{1/3} \Delta^{1/6}}{B^{0.5}} \geq h_0 \quad \text{m}$$

where

f_u = 0.615

f_s = factor of hull shape

$$= \frac{\cos^{1.5}\varphi \cdot \sin^{0.5}\left(\frac{\alpha_0 + \beta_0 + \beta_2}{3}\right)}{\sin^{1.5}(90^\circ - \beta_{10})} \cdot \left(\frac{B}{L}\right)^{0.2}$$

f_p = factor of propellers arrangement, as follows:

= 0.88 for single screw ships

= 0.99 for twin screw ships

= 1.06 for triple screw ships with all three propellers of the same diameter

For vessels with azimuthing propellers only

= 0.9 for single azimuthing pod

= 1.0 for two azimuthing pods

N_p = total power delivered to the propellers, MW

B = maximum breadth of ship at DWL, m

L = LBP, m

Δ = displacement of ship at DWL or at the deepest WL for ice conditions, whichever is greater, tons

d_{pr} = diameter of the propellers, m

φ = stem inclination angle to the waterplane at DWL, deg

α_0 = angle between DWL and CL at FP, deg

β_0 = flare angle between side shell line and CP at DWL at STA 0 (FP), deg

β_2 = flare angle between side shell line and CP at DWL at STA 2, deg

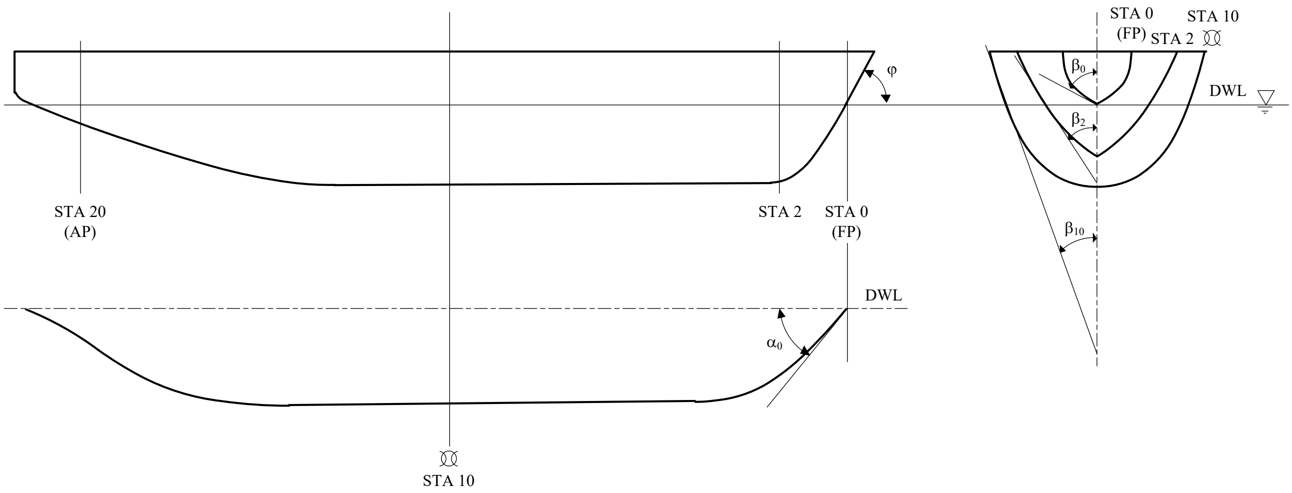
β_{10} = flare angle between side shell line and CP at DWL at the midship (STA 10), deg

Special consideration will be given to vessels with other arrangements of propellers including the use of both bow and stern shaft-line propellers operating jointly, or a mix of shaft-line and azimuthing propellers, or other.

TABLE 7
Nominal Values of Powering Criteria *(2012)*

Ice Class	*Nominal Ice Thickness h_0,* m	
	Ice Breakers	*Ice Class Vessels*
PC1	2.8	2.5
PC2	2.2	1.9
PC3	1.6	1.3
PC4	1.2	1.0
PC5	0.8	0.6
PC6	0.6	0.5
PC7	--	0.5

FIGURE 3
Flare Angle Between Side Shell Line and CP at DWL *(2014)*



21.5 Total Power Delivered to Propellers *(2014)*

The total ahead maximum continuous rated power, N, delivered to the propellers is to be not less than the values obtained as follows:

i) For Ice Class **PC1** through **PC4** and vessels assigned the **Ice Breaker** notation:

$$N = kA(B)^{0.8} (L)^{0.4}[1 + me^{-5\Delta \times 10^{-6}}] \quad \text{kW}$$

ii) For Ice Class **PC5** through **PC7**:

The smaller of the values obtained from the following two equations:

$$N = kA(B)^{0.8}(L)^{0.4}(1 + me^{-5\Delta \times 10^{-6}}) \quad \text{kW}$$

$$N = k(C + K\,\Delta/1000) \quad \text{kW}$$

where

N = total propulsion power delivered to propellers

e = base of natural logarithms

k = unit system factor

= 0.735

B, *L*, and *Δ* are as defined above.

A, *m*, *C*, and *K* are coefficients given in 6-1-4/Table 8.

TABLE 8
Power Coefficients *(2014)*

Ice Class	*A* in SI units	*m*	*C*	*K*
PC1	360	1.3	---	---
PC2	270	1.0	---	---
PC3	200	0.8	---	---
PC4	136	0.6	---	---
PC5	107	0.6	1500	400
PC6	93	0.6	1000	350
PC7	93	0.6	1000	350

21.7 Powering Criteria Obtained from Ice Model Tests

At later design and construction stages, when results of ice model tests are available, the value of h_{max} calculated in 6-1-4/21.3 or/and the value of N_p calculated in 6-1-4/21.5 can be superseded by the results of self-propelled model tests in an ice model testing basin. The model tests have to be conducted according to a standard procedure approved by International Association of Ice Model Testing Basins. Using standard series model propellers will be approved in the self-propelled model tests used to produce the required values of h_{max} and N_p provided that the standard series model propellers are most similar to the actual full-scale propellers approved for the vessel.

21.9 Astern Power *(2014)*

The following requirements apply to all main propulsion systems fitted to a vessel with an Enhanced Polar Class notation

i) **PC1, Enhanced** through **PC6, Enhanced** with **Ice Breaker** notation:

Total astern power delivered to the propellers is to be not less than that required in 6-1-4/21.1.

ii) All vessels with Enhanced Polar Class Notation intended to operate astern in accordance with 6-1-2/3.1vii):

Total astern power delivered to the propellers is to be not less than that required in 6-1-4/21.1.

iii) **PC1, Enhanced** through **PC3, Enhanced**:

Total astern power delivered to the propellers is not to be less than 90% of that required in 6-1-4/21.1.

iv) **PC4, Enhanced**:

Total astern power delivered to the propellers is not to be less than 85% of that required in 6-1-4/21.1.

v) **PC5, Enhanced** through **PC7, Enhanced**:

Total astern power delivered to the propellers is not to be less than 70% of that required in 6-1-4/21.1.

23 Flexible Couplings

Flexible couplings which may be subject to damage from overheating are to be provided with temperature-monitoring devices or equivalent means of overload protection with alarms at each engine control station.

25 Bossings

The bossings are to be designed to withstand the design ice forces F_n, as specified by 6-1-4/29.3, where d_1 is the diameter of the bossing. The bossing plating thickness is to be not less than required by 6-1-2/7.3 for the stern ice belt area, where s is the distance between stiffeners.

27 Rudder and Steering Arrangements

27.1 General

27.1.1 Multiple Rudders

Where two or more rudders are provided, they are to be mechanically independent.

27.1.2 Pintles

Rudders are to have at least two pintles.

27.1.3 Rudder Stops

Rudders are to be protected by strong and effective external rudder stops and provided with mechanical means of locking the rudder parallel to the centerline for use in the astern condition.

27.1.4 Ice Knife

Rudders are to be protected by ice knives or other similar structures located abaft the rudder. Clearance between the ice knife and the rudder is not to exceed 100 mm (4 in.)

27.3 Rudder Stocks, Couplings and Pintles

27.3.1 Ice Classes **PC1** through **PC5**

In addition to the requirements in Section 3-2-14, the rudder stocks, couplings and pintles are to meet the ice strengthening requirements, using equations in Section 3-2-14 in association with V_i, A_i and r_i, as defined below, in lieu of V, A, A_1, A_2, r, r_1, and r_2.

V_i = the greater of V, as defined in Section 3-2-14, or the minimum design speed in 6-1-4/Table 9

A_i = that part of the total projected area, A, A_1 or A_2, as defined by Section 3-2-14, that is abaft the rudder stock centerline

r_i = distance from the centerline of the rudder stock to the centroid of A_i

27.3.2 Ice Classes **PC6** and **PC7**

For ice classes **PC6** and **PC7**, rudder stocks, pintles, gudgeons and other bolting arrangements to the stern frames are to meet the requirements in Section 3-2-14 in association with V_i, as specified in 6-1-4/27.3.1.

TABLE 9
Design Speed for Rudders, Couplings and Pintles *(2012)*

Ice Class	*Minimum Design Speed, knots*
PC1	29
PC2	29
PC3	28
PC4	26
PC5	23
PC6	20
PC7	20

27.3.3 Ice Classes **PC1** through **PC7**

The stresses in these members with the load F applied as follows are not to exceed the shear yielding strength which may be taken as 0.577 times the specified yield point of the material.

$$F = 2K_3(Dt)^{1/2} \quad \text{kN (tf, Ltf)}$$

where

K_3 = as given in 6-1-4/Table 10

D = ship displacement, in tonnes (long tons), as specified in 6-1-2/5.7

t = thickness of the rudder, in m (ft), measured at the level of F and at 10% of the rudder length from the trailing edge.

F is to be applied to the after edge of the rudder in a direction parallel to the centerline of the vessel at all locations below the ice waterline within the middle 40% of the rudder height to determine the most severe requirements. Alternatively, F may be spread over any 60% of the rudder height as a uniform load. No other force need be considered simultaneously with F.

27.5 Double Plate Rudder

For double plate rudders, the minimum thickness of plates is to be not less than required by 6-1-4/29.5.

29 Propeller Nozzles

29.1 General

This Subsection applies to fixed nozzles. Special consideration will be given to steering nozzles. The nozzles are to be supported at least at the upper and lower ends. The strength, rigidity and resistance to buckling of the nozzle are to be adequate for the design ice forces given in 6-1-4/29.3. All of the critical loading cases are to be considered. In no case under the design ice forces are the normal and axial displacements of the inside ring to exceed 10% of the clearance between the inside plating of the nozzle and the propeller blade tips, or 0.5% of the inside ring diameter, whichever is less. Nozzles are to be protected by stern structures as much as possible against direct impacts with large ice features.

29.3 Design Ice Forces

The design ice forces are to be not less than those obtained from the following equations:

$$F_n = K_1K_2(D\,d_1)^{1/2} \quad \text{kN (tf, Ltf)}$$

$$F_f = K_3K_4[D\,(d_1 - d_2)]^{1/2} \quad \text{kN (tf, Ltf)}$$

where

F_n = the design ice force applied normal to the outside surface of the nozzle in the most critical location

K_2 = 1 for the external sides of a single nozzle of a single screw vessel

= 1.1 for the outboard external sides of the outermost nozzles of vessels with two or more screws

= 0.25 for the external sides of nozzles situated between the outermost ones and for the internal sides of any nozzles

= 0.8 for bottoms of the nozzles

D = ship displacement, in tonnes (long tons), as specified in 6-1-2/5.7

d_1 = maximum outer diameter of the nozzle, in m (ft)

d_2 = minimum internal diameter of the nozzle, in m (ft)

F_f = the design ice force applied to the ends of the nozzle, parallel to the propeller axis, in the most critical locations

K_4 = 1 for aft end face of the nozzle having no rudder behind

= 0.7 for the aft end face of the nozzle with a rudder behind

= 0.6 for the fore end face of the nozzle

K_1 and K_3 are as given in 6-1-4/Table 10.

Values of K_2 and K_4 less than above will be approved, provided the stern and bottom hull structures effectively protect the nozzle against large ice fragments.

TABLE 10
Design Ice Force Coefficient *(2012)*

Ice Class	K_1	K_3
	SI units (MKS, US)	*SI units (MKS, US)*
PC1	55 (5.6, 3.1)	294 (30.0, 16.4)
PC2	53 (5.4, 3.0)	286 (29.2, 16.0)
PC3	49 (5.0, 2.7)	243 (24.8, 13.6)
PC4	43 (4.4, 2.4)	188 (19.2, 10.0)
PC5	32 (3.3, 1.8)	110 (11.2, 6.1)
PC6	20 (2.1, 1.1)	59 (6.0, 3.3)
PC7	20 (2.1, 1.1)	59 (6.0, 3.3)

29.5 Plate Thickness

The plate thickness of both inner and outer surfaces of the nozzle is to be not less than required by 6-1-2/7.1 for the stern ice belt area where the corrosion and abrasion allowance, t_s, is as given in 6-1-4/Table 11.

TABLE 11
Corrosion/Abrasion Additions for Nozzle Surface Plating *(2012)*

Hull Area	t_s, *mm*					
	With Effective Protection			*Without Effective Protection*		
	PC1 - PC3	**PC4 & PC5**	**PC6 & PC7**	**PC1 - PC3**	**PC4 & PC5**	**PC6 & PC7**
Nozzle Surface Plating	0	0	0	2.0	1.8	1.0

31 Hull Structural Materials *(2012)*

31.1 Inspection

In addition to the nondestructive inspection requirements of the other sections of the Rules, all intersections of full penetration welds within the ice belt structure of ice class vessels **PC1 Enhanced** to **PC4 Enhanced** are to be inspected by radiographic or ultrasonic methods and are to meet the Class A requirements of the ABS *Rules for Nondestructive Inspection of Hull Welds*. Additional inspections may also be required by the Surveyor for other locations including block connection joints.

PART
6

CHAPTER 1 Strengthening for Navigation in Ice

SECTION 5 Requirements for Vessels Intended for Navigation in First-year Ice

1 General *(2015)*

1.1 Application

Vessels to be distinguished in the *Record* by **Ice Class** followed by ice class **A0**, **B0**, **C0**, **D0** or **E0** as specified in 6-1-5/3.1 are to meet the applicable requirements of this Chapter.

Non-self-propelled vessels are to comply with the requirements in 6-1-5/31. Vessels requiring ice breaker assistance are to comply with the additional requirements in 6-1-5/37.1.

3 Selection of Ice Class *(2012)*

3.1 Ice Class *(2015)*

The requirements in this Section are intended primarily for vessels intended for navigation in first-year ice. The ice classes are as follows:

Ice Class A0

Ice Class B0

Ice Class C0

Ice Class D0

Ice Class E0

3.3 Guide for Selection

For the guidance of the Owner in selecting the most suitable ice class, ice conditions suitable for respective ice classes are shown in 6-1-5/Table 1. The conditions of first-year ice are shown in 6-1-5/Table 2.

TABLE 1
Regions and Periods for Navigation in Ice for Selecting Ice Class *(2015)*

Ice Class	*Navigating independently or when escorted by an ice breaker of the following ice classes*	*Year around navigation in water with first-year ice with the ice conditions given in 6-1-5/Table 2*
A0	Escorted by **PC4** or Higher Ice Class Vessel	Extreme
B0	Escorted by **PC3** or Higher Ice Class Vessel	Extreme
A0, B0, C0	Escorted by **PC5** or Higher Ice Class Vessel	Very Severe
A0	Independently	Severe
B0	Independently	Medium
C0	Independently	Light
D0	Independently	Very Light
E0	Independently	Very Light drift ice [in coastal areas]

TABLE 2
Ice Conditions of First-Year Ice Versus Concentration and Thickness of Ice Cover

Thickness of First-Year Ice Cover in m (ft)	*Concentration of Ice* (1)			
	Very Close and Consolidated Ice, Fast Ice (from 10/10 to 9/10 or from 8/8 to 7/8)	*Close Ice (from 9/10 to 6/10 or from 7/8 to 5/8)*	*Open Ice (from 6/10 to 3/10 or from 5/8 to 2/8) and Fresh Channel*(2) *in Fast Ice (more than 6/10 or 5/8)*	*Very Open Ice (less than 3/10 or 2/8), Fresh Channel*(2) *in Fast Ice (6/10 or 5/8 and less) and Brash Ice*
1.0 (3.3) and above	Extreme	Extreme	Very severe	Severe
from 0.6 (2) to 1.0 (3.3)	Extreme	Very severe	Severe	Medium
from 0.3 (1) to 0.6 (2)	Very severe	Severe	Medium	Light
less than 0.3 (1)	Severe	Medium	Light	Very light

Notes

1 These ratios of mean area density of Ice in a given area are from the "World Meteorological Organization Sea Ice Nomenclature", Appendix B.7, and give the ratio of area of Ice concentration to the total area of sea surface within some large geographic locales.

2 Provided the channel is wider than the ship

5 Definitions

5.1 Ice Belt

The *Ice Belt* is that part of the shell plating and hull appendages defined in 6-1-5/7 for self-propelled vessels and in 6-1-5/31.5 for non-self-propelled vessels.

5.3 Upper Ice Waterline

The *Upper Ice Waterline* is the deepest waterline at which the vessel is intended to operate in ice. The upper ice waterline is to be clearly indicated on the shell expansion drawing.

5.5 Lower Ice Waterline

The *Lower Ice Waterline* is the lightest waterline at which the vessel is intended to operate in ice. Generally, it is to be located so that propellers are fully submerged. The lower ice waterline is to be clearly indicated on the shell expansion drawing.

5.7 Displacement

The *Displacement*, *D*, is the molded displacement in metric tons (long tons) at the upper ice waterline. For the purposes of this section, the displacement may be calculated using a specific gravity of 1.00.

5.9 Length

The *Length, L* is the length at the upper ice waterline.

7 Extent and Length of Ice Belt Areas *(2015)*

The ice belt for self-propelled vessels is subdivided into the following areas:

- *For ice class* **A0** *through* **C0**

  Bow, midbody and stern areas.

- *For ice class* **D0** *and* **E0**

  Bow area.

For all first-year ice classes, the lowest extent of the bow area need not extend below a line drawn between Q m (ft) below the lower ice waterline at the stem and B m (ft) below the lower ice waterline at the stern. (See 6-1-5/Table 3 for values of Q and B.) The extent and length of each area is shown in 6-1-5/Figure 1 and 6-1-5/Table 3.

TABLE 3
Dimensions of Ice Belt Areas, m (ft) *(2015)*

Ice Class	*A*	*B*	*C*	*D*	*F**	*S*	*Q*
A0	0.8 (2.6)	0.6 (2.0)	$0.5D$	$0.2 + 0.004L$ $(0.7 + 0.004L)$	$0.3L$	$0.10L$	10.0 (33.0)
B0	0.6 (2.0)	0.5 (1.6)	0	$0.1 + 0.003L$ $(0.3 + 0.003L)$	$0.3L$	$0.10L$	9.0 (30.0)
C0	0.6 (2.0)	0.5 (1.6)	0	$0.0025L$	$0.3L$	$0.10L$	6.6 (22.0)
D0, E0	0.5 (1.6)	0.5 (1.6)	0	$0.002L$	$0.3L$	0	4.5 (15.0)

* For ships with upper ice waterline parallel to centerline, F is to be as shown in 6-1-5/Figure 1c. In any case, the bow area is to extend aft not less than to a section at:

M = $0.2L$ abaft the fore-end of the lower ice waterline, or

N = $0.05L$ abaft point where the molded stem line crosses the baseline, whichever is located aft.

FIGURE 1
Ice Belt Areas *(2012)*

FIGURE 1a
Ice Class A0 through C0

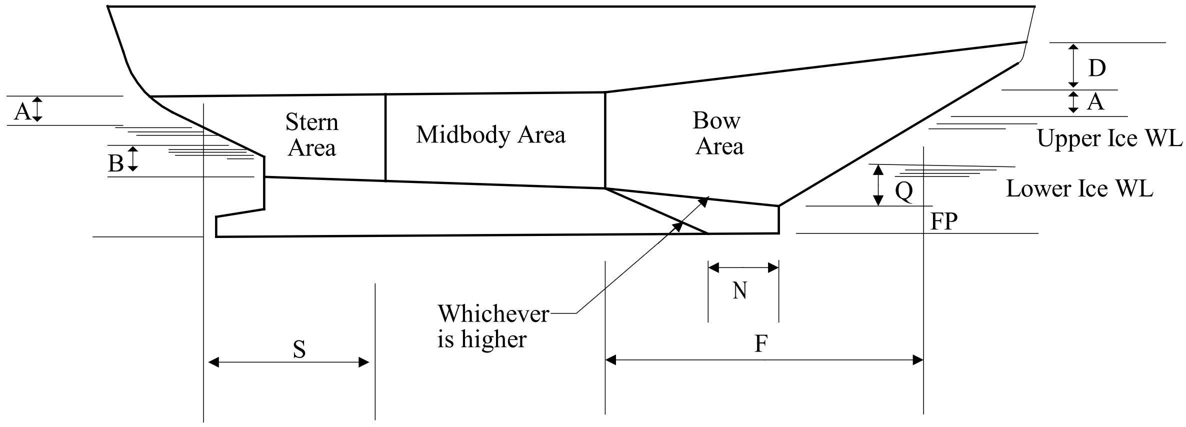


FIGURE 1b
Ice Class D0 and E0 *(2015)*

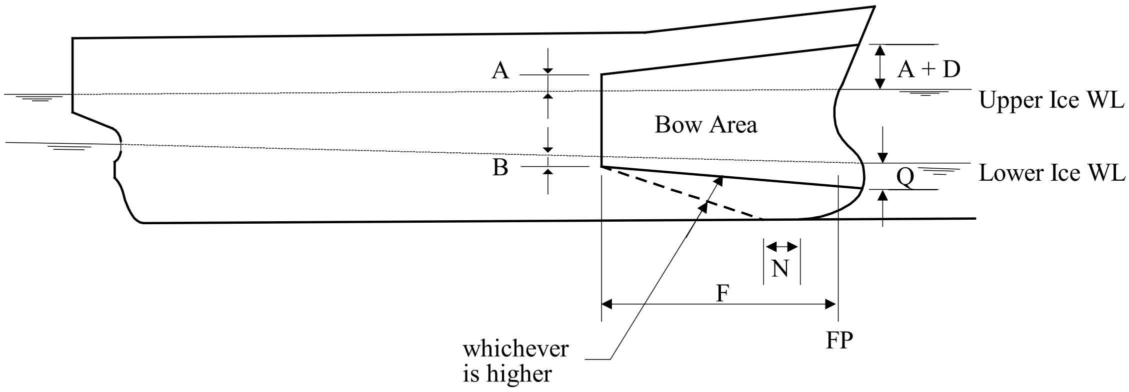
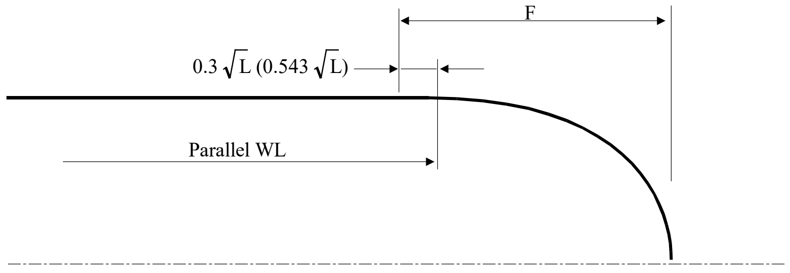
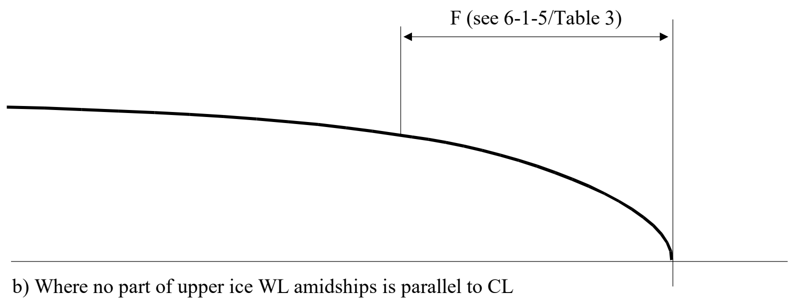


FIGURE 1c
Definition of F



a) Where midship part of upper ice WL is parallel to CL



b) Where no part of upper ice WL amidships is parallel to CL

9 Design Ice Loads *(2012)*

9.1 Design Ice Pressure on the Bow Area *(2015)*

The design ice pressure on the bow area is to be not less than that obtained from the following equations:

$P_b = P_o F_b$

P_b = design ice pressure on the bow area, in N/mm^2 (kgf/mm^2, ksi)

- *For all first-year ice classes*

$P_o = B(D/n)^{0.2}$

where

B = coefficient, as given in 6-1-5/Table 4

D = displacement, as defined in 6-1-5/5.7

n = 1000 (1000, 984)

F_b = $(F_{b1})\ (F_{b2})$

F_{b1} = coefficient is given in 6-1-5/Figure 2. It is to be determined for each bow section at the upper and lower ice waterlines depending on α_b and β_b and the maximum value obtained is to be used; if the values of coefficient F_{b1} obtained for the different sections and at different ice waterlines vary by more than 15%, different coefficients F_{b1} and, correspondingly, different design ice pressures may be used for the appropriate parts of the bow area.

F_{b1} is not to be taken less than 0.80, but need not be taken as more than 1.25 for vessels with conventional bows; for vessels fitted with bulbous bows, the F_{b1} coefficient within the bulb area is to be as given in 6-1-5/Table 4

F_{b2} = $1 + i\,(1.3 + 0.001D)^{-2}$

i = coefficient given in 6-1-5/Table 4

α_b = angle between the centerline and a tangent to the ice waterline being considered at the bow section being considered

β_b = angle between the vertical and tangent to the bow section at the level of the ice waterline being considered.

TABLE 4
Bow Area Ice Pressure Coefficients *(2015)*

Ice Classes	*B* *N/mm² (kgf/mm², ksi)*	*i*	F_{b1} *
A0	0.997 (0.102, 0.142)	2	1.35
B0	0.750 (0.076, 0.109)	0	1.25
C0	0.60 (0.061, 0.086)	0	1.25
D0	0.50 (0.051,0.071)	0	1.25
E0	0.30 (0.031,0.043)	0	1.25

* Within the bulbous bow area

9.3 Design Ice Pressures on Other Ice Belt Areas

Design ice pressures on other parts of the ice belt are to be obtained from the following equations:

- *For the midbody*

  $P_m = K_m P_o$ or $P_m = K_m P_b$, whichever is less

- *For the stern*

  $P_s = K_s P_b$

P_m and P_s = design ice pressures on corresponding area, in N/mm² (kgf/mm², ksi)

K_s = coefficient, as given in 6-1-5/Table 5

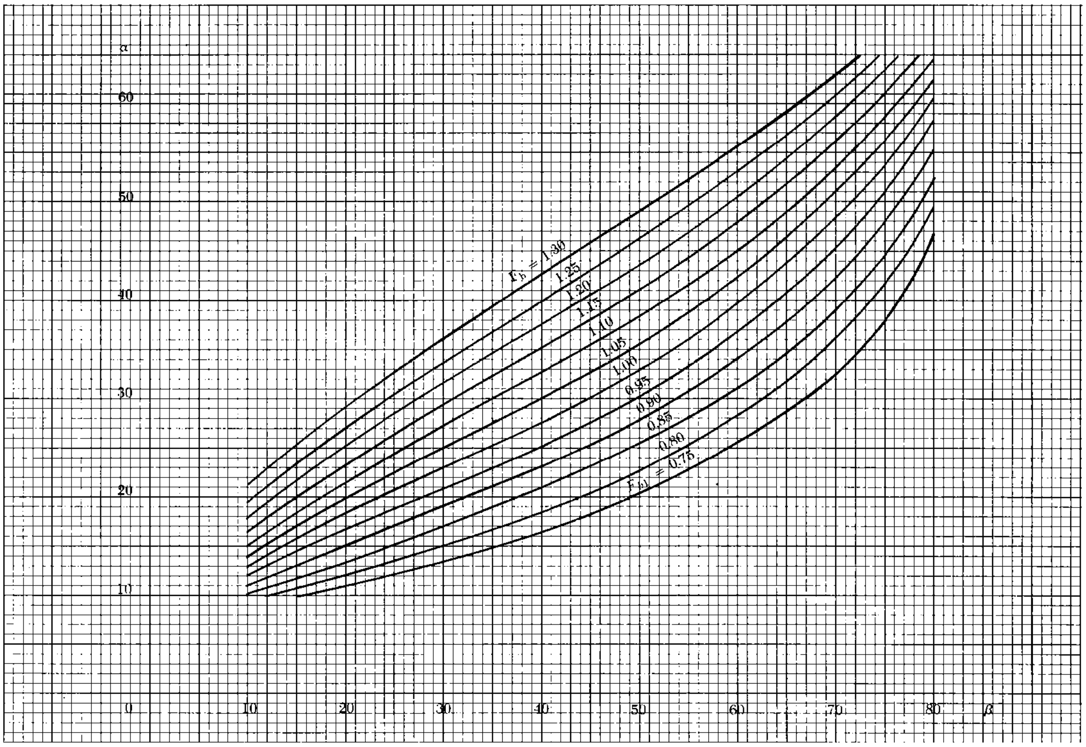
K_m = coefficient, as given in 6-1-5/Table 5 or by $2(3 + 4\sin\beta_m)^{-1}$, whichever is less

β_m = as defined for β_b (see 6-1-5/9.1), but for the section at amidships

TABLE 5
Ice Pressure Coefficients in Other Areas *(2012)*

Ice Class	K_s	K_m
A0	0.35	0.45
B0	0.22	0.35
C0	0.11	0.22

FIGURE 2
Coefficients F_{b1} Versus angles α_b and β_b



9.5 Extent of Design Ice Load *(2014)*

In a vertical direction, the design ice pressure is considered to be uniformly distributed on the side structure. The vertical extent of the design ice pressure is to be obtained from the following equations:

- *For the bow*

  $b_b = 0.61 + b_o F_{b1}$ m

  $b_b = 2 + b_o F_{b1}$ ft

- *For the midbody*

  $b_m = 0.65 + 0.5 b_o$ m

  $b_m = 2.13 + 0.5 b_o$ ft

- *For the stern*

  $b_s = 0.61 + 0.7 b_o$ m

  $b_s = 2 + 0.7 b_o$ ft

where b_b, b_m and b_s are the vertical extent of the design ice pressure, in m (ft)

$b_o = R\,(N/k)^{0.25}\,(D/n)^{0.2}$

R = coefficient, as given in 6-1-5/Table 6

F_{b1} = coefficient, as given in 6-1-5/9.1

k = 746 (100, 986)

N = total maximum continuous power delivered to the propellers, in kW (mhp, hp)

For **A0** to **C0** ice class vessels fitted with bulbous bows, the extent b_b within the bulbous area of the ice belt is to be 30% more.

D and *n* are as defined in 6-1-5/9.1.

TABLE 6
Extent of Ice Load Coefficients *(2015)*

Ice Class	*R, m (ft)*
A0	0.020 (0.066)
B0	0
C0	0
D0	0
E0	0

11 Shell Plating

11.1 Ice Belt with Transverse Framing *(2014)*

The thickness of the ice belt shell plating is to be not less than that obtained from the following equation:

$t = 0.60s\,(P/Y)^{1/2} + Ct_o$ mm (in.)

where

t = thickness of the shell plating, in mm (in.)

s = distance measured along the shell between adjacent frames, in mm (in.)

P = design ice pressure in appropriate region, as given in 6-1-5/9, in N/mm^2 (kgf/mm^2, ksi)

Y = minimum yield strength of the material, in N/mm^2 (kgf/mm^2, ksi)

C = 1 for the bow area

= 0.80 for the midbody area

= 0.65 for the stern area

t_o = as given in 6-1-5/Table 7

In no case is the thickness of the bow, mid and stern areas of the ice belt plating to be less than given in 6-1-5/Table 7.

TABLE 7
Minimum Thickness and Abrasion Allowance of Ice Belt Plating *(2015)*

Ice Class	*Minimum Thickness*	t_o * *mm (in.)*
A0	12 (0.47)	3 (0.118)
B0	10 (0.39)	3 (0.118)
C0	8 (0.315)	3 (0.118)
D0	8 (0.315)	1 (0.04)
E0	8 (0.315)	1 (0.04)

* Values of t_o may be reduced down to $0.3t_o$, if an abrasive-resistant coating is used for the ice belt plating. Special approval of this will be based on necessary evidence including submission of results of operational experience in ice.

11.3 Ice Belt with Longitudinal Framing *(2014)*

The thickness of ice belt shell plating is to be not less than that obtained from the following equation:

$$t = 0.7s\,(P/Y)^{1/2} + Ct_o \quad \text{m (in.)}$$

where

s = distance between longitudinal frames, in mm (in.)

t, P, Y, t_o, C are as defined in 6-1-5/13.1.

The thickness of ice belt plating is also to be not less than the thickness given in 6-1-5/Table 7, plus 1 mm (0.04 in.).

11.5 Changes in Plating Thickness

Plating thickness in the transverse direction from the ice belt to the bottom and in the longitudinal direction within the ice belt is to be gradually tapered.

13 Transverse Framing

13.1 Definitions

13.1.1 Main Frames

Main Frames are the hold, tween deck and peak frames referred to in Section 3-2-5.

13.1.2 Intermediate Frames

Intermediate Frames are the additional frames fitted within the ice belt between the main frames, to comply with 6-1-5/13.3.

13.1.3 Standard Frame Spacing

Standard Frame Spacing is the frame spacing specified by 3-2-5/1.7 and is measured along the centerline.

13.3 Ice Belt Frame Spacing *(2015)*

Except for the midbody and stern areas of ice class **C0** and the bow area of ice class **E0**, spacing between any adjacent frames measured along the centerline is in general not to exceed one half of the standard frame spacing defined in 6-1-5/13.1.3. A larger spacing between any adjacent frames may be approved if the intermediate frames have end fixity similar to that of the main frames. In no case is the spacing between any adjacent frames measured alongside plating to exceed 0.75 of the standard frame spacing given in 6-1-5/13.1.3.

13.5 Main and Intermediate Frames

13.5.1 Section Modulus *(2015)*

The section modulus, SM, of each transverse main and intermediate frame in association with the width of plating, s, to which it is attached is to be not less than that obtained from the following equation:

$$SM = Ks\ell b(P/Y) \quad \text{cm}^3$$

$$SM = 0.144Ks\ell b(P/Y) \quad \text{in}^3$$

where

K = $(160 - 100b/\ell)K_1K_2$

s = distance between adjacent frames, in mm (in.), measured along the lowest ice waterline in way of the compartment being considered

ℓ = span of the main frame, in m (ft), measured along the frame between decks or between deck and inner bottom

b = vertical extent of the design ice pressure, as defined in 6-1-5/9.5, in m (ft)

P = the design ice pressure, as defined in 6-1-5/9

Y = minimum yield strength of the material, in N/mm^2 (kgf/mm^2, ksi)

For framing system with supporting stringers in accordance with 6-1-5/13.9, coefficient K_1 is to be obtained from the equation:

$K_1 = 2/(3 + j)$

where j = number of the supporting stringers.

For framing system without supporting stringers, coefficient K_1 is to be as given in 6-1-5/Table 8a.

K_2 = 1.1 for the midship area of the ice belt for ice classes **A0** through **C0**

= 1 elsewhere

The web thickness, t, of the main and intermediate frames is to be not less than:

$t = 0.013h + 6$ mm

$t = 0.013h + 0.24$ in.

where h is the depth of the main and intermediate frame, in mm (in.).

In no case is the web thickness t to be less than the following:

Ice class **A0**	9 mm (0.35 in.)
Ice class **B0**	8.5 mm (0.34 in.)
Ice class **C0**, **D0** and **E0**	8.0 mm (0.31 in.)

13.5.2 Upper End of Frames *(2015)*

Main and intermediate frames are to extend up to the first deck or platform above the ice belt. They are to be welded and bracketed to the deck beams or to the deck longitudinals, as shown in 6-1-5/Figure 3a and 6-1-5/Figure 3b.

For ice classes **A0** through **E0**, where the lowest or only deck, or the lowest platform, is situated above the ice belt so that the distance between the deck, or platform, and the upper boundary of the ice belt exceeds "*d*" meters (feet), given in 6-1-5/Table 8b, the upper ends of intermediate frames in the midbody and stern areas (**A0** through **C0**) or bow area (**D0** and **E0**) may terminate at a deep stringer situated at least 0.6 m (2 ft) above the ice belt.

For ice classes **A0**, **B0**, **C0**, **D0** and **E0** in tween deck spaces, where the tween deck is 0.5 m (1.6 ft) or more above the upper ice waterline but within the ice belt, the upper ends of intermediate frames may terminate for ice class **A0** at a stringer situated at least 0.5 m (1.6 ft) above the ice belt, and for ice classes **B0**, **C0**, **D0** and **E0** at an intercostal longitudinal at least 0.5 m (1.6 ft) above the ice belt.

The upper ends of the frames terminated at a deep stringer are to be welded and bracketed to it as shown in 6-1-5/Figure 3c.

The intermediate frames terminated at an intercostal stringer or longitudinal are to be welded to it as shown in 6-1-5/Figure 3d.

TABLE 8a
Coefficient K_1 for the Framing System without Supporting Stringers *(2014)*

Termination of the upper & lower ends of the main & intermediate frames	*At the upper deck (or platform) of the adjacent upward spaces*	*Other*
At bottom structures or at the lower deck of the adjacent downward spaces (hold, tween deck, tank, etc.)	0.9	1
Other	1	1.15

TABLE 8b
Distance, m (ft) *(2015)*

Ice Class	*Where Web Frames are Fitted*	*No Web Frames are Fitted*
A0	3.0 (10)	—
B0	2.1 (7)	3 (10)
C0	1.2 (4)	1.8 (6)
D0	1.2 (4)	1.8 (6)
E0	1.2 (4)	1.8 (6)

13.5.3 Lower End of Frames *(2015)*

Main and intermediate frames are to extend down to the inner bottom or to the double bottom margin plate. For ice class **A0**, the intermediate frames may terminate at a deck 1.0 m (3.3 ft) below the ice belt. For ice classes **B0**, **C0**, **D0** and **E0**, the intermediate frames may terminate at a stringer or intercostal longitudinal situated at least 1.0 m (3.3 ft) below the ice belt. The main and intermediate frames are to be attached and bracketed either to the inner bottom or to the double bottom margin plate or to the deck beams, or deck or to the stringer as shown in 6-1-5/Figure 4.

For vessels not having a double bottom, the intermediate frames are to extend down to a point below the top of the bottom transverses and are to terminate at an intercostal longitudinal. For ice classes **A0**, **B0**, **C0**, **D0** and **E0**, the intermediate frames need not extend below the top of the floors, provided they terminate on an intercostal longitudinal not less than 0.8 m (2.6 ft) below the ice belt. The intermediate frames are to be attached to the bottom intercostal longitudinals.

13.5.4 Connection to Stringers and Decks

Main and intermediate frames are to be attached and bracketed to each supporting (deep) stringer, deck and deck beam within the ice belt.

13.7 Web Frames *(2015)*

The section modulus, *SM*, of each web frame, in association with the plating to which it is attached, is to be not less than that obtained from the following equation:

$SM = Ks_1 \ell b(P/Y)$ cm^3

$SM = 0.144Ks_1 \ell b(P/Y)$ in^3

where

K = $(96 - 36b/\ell)K_1K_2K_3$

K_1 = $1.06 - 0.0024i^2$, but not less than 0.4

i = number of the main and intermediate frames between adjacent web frames

K_2 = 1 for the bow and stern areas of the ice belt

= 1.2 for the midship area of the ice belt for ice classes **A0** through **C0**

K_3 = 1 if there is one supporting (deep) stringer

= 0.90 if there are two supporting (deep) stringers

= 0.85 if there are three or more supporting (deep) stringers

s_1 = distance between the web frames, in mm (in.), measured along lower ice waterline in way of the compartment being considered

ℓ = span, in m (ft), measured in a straight line along the hold web frame from the line of the inner bottom (extended to the side of the vessel) to the lowest deck of the hold, or for the tween deck web frame measured between the decks as shown in 6-1-5/Figure 3a or 6-1-5/Figure 3b and 6-1-5/Figure 4

b = as defined in 6-1-5/9.5, in m (ft)

P = as defined in 6-1-5/9, in N/mm^2 (kgf/mm^2, ksi)

Y = as defined in 6-1-5/11.1, in N/mm^2 (kgf/mm^2, ksi)

In determining the section modulus, the effective width of the plating is to be the distance between the webs or 0.125ℓ, whichever is less.

Thickness of the web plate, t, is to be not less than that obtained from the following equation:

$t = 0.01h + 8$ mm

$t = 0.01h + 0.32$ in.

where h is the depth of the web frame; t need not exceed 15 mm (0.59 in.).

The web frames are to be attached and bracketed to the solid floors and the beams at each ice deck.

13.9 Ice Stringers

13.9.1 Arrangements *(2015)*

Deep continuous or intercostal stringers are to be fitted in the bow area of the ice belt for ice class **C0**, **D0** and **E0** vessels and within the ice belt throughout the entire length of the vessel for ice class **A0** and **B0** vessels.

The spacing between adjacent stringers or between the stringer and a deck or the double bottom measured along the shell is to be not more than indicated in 6-1-5/Table 9. One of the ice stringers is to be fitted about 200 to 400 mm (8 to 16 in.) below the upper ice waterline, if there is no deck in this area. For ice class **A0**, another stringer is to be fitted about 100 to 300 mm (4 to 12 in.) below the lower ice waterline, if there is no deck or similar support in this area.

FIGURE 3
Upper End Terminations of Frames

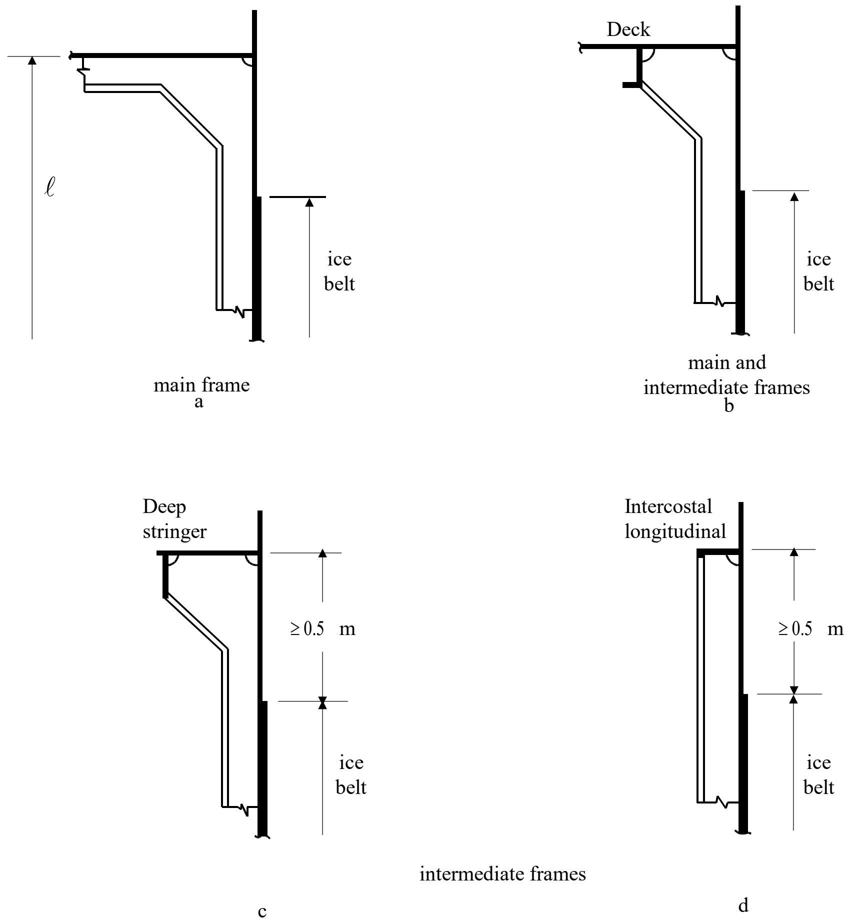


FIGURE 4
Lower End Terminations of Frames

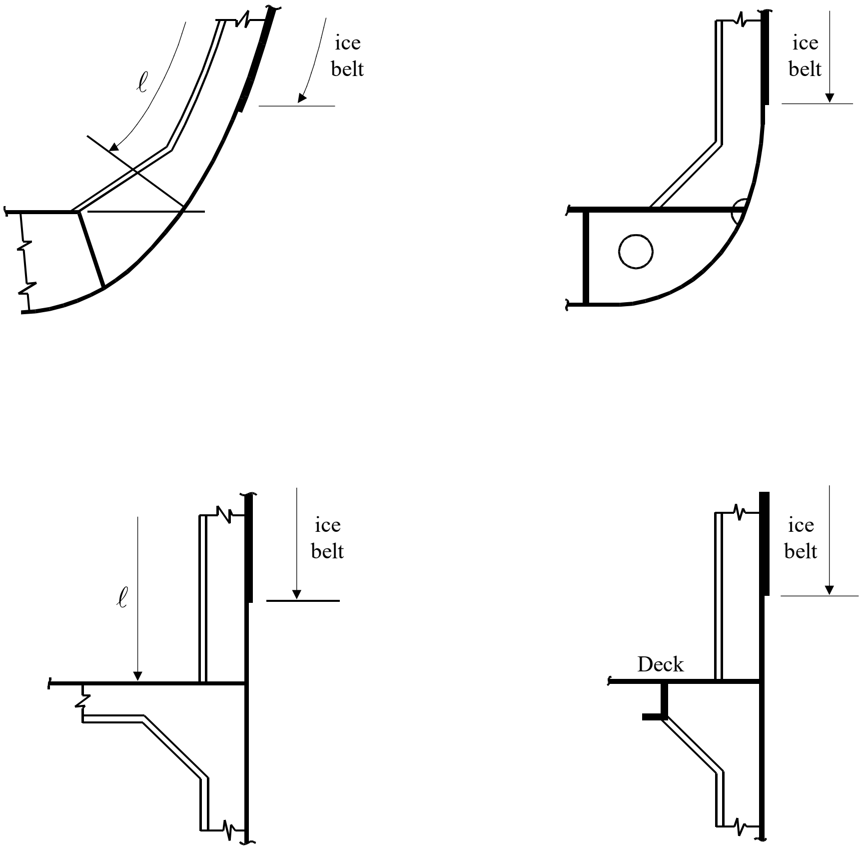


TABLE 9
Maximum Stringer Spacing, m (ft) *(2015)*

Ice Class	*For Framing without Web Frames*	*System with Web Frames*
A0 through **E0**	1.5 (5)	2.7 (9)

13.9.2 Scantlings and Connections *(2014)*

Where ice stringers are intercostal, the following criteria shall be met,

i) The intercostal stringers shall be fitted between frames and their scantlings are to be not less than those for main frames.

ii) The intercostal stringers are to be welded to the main and intermediate frames

iii) The web plate and the flange, or face, of intercostal ice stringers are to be attached to those of the main and intermediate frames.

iv) The intercostal stringers are to be bracketed to the bulkheads, side transverses, or web frames

Where deep ice stringers are fitted, the following criteria shall be met:

i) The shear area of the deep ice stringer within one frame space from the web frame is to be not less than that of the web frames.

ii) The depth of the ice stringer at the midspan between the web frames is to be not less than twice the depth of the main frame.

iii) The face, or flange, area of the deep stringer is to be not less than that of the web frame.

iv) The web plate and the face, or flange, of deep ice stringers are to be attached to those of the web frames.

v) The deep stringer referred to in 6-1-5/13.5.2 at which the upper ends of frames are terminated, is to have the scantlings as required in-6-1-5/13.9.

vi) The deep stringers are to be bracketed to the bulkheads or side transverses, so that the shear area at the bulkhead is twice that of the ice stringer web.

Stiffeners or tripping brackets are to be fitted as required in 3-2-6/3.7 and 3-2-6/3.9.

15 Longitudinal Framing

15.1 Spacing of Longitudinals

The spacing measured along the shell between adjacent longitudinals and between the longitudinal and the double bottom or a deck within the ice belt is not to exceed one half of the spacing as given in 3-2-5/1.7.

15.3 Section Modulus

The section modulus, *SM*, of each longitudinal, in association with the width of plating, *s*, to which it is attached, is to be not less than that obtained from the following equation:

$$SM = 70s\ell^2 K_o(P/Y) \quad \text{cm}^3$$

$$SM = 10s\ell^2 K_o(P/Y) \quad \text{in}^3$$

where

s = spacing of longitudinals, as defined in 6-1-5/15.1, in mm (in.)

ℓ = span, in m (ft), of the longitudinals measured at the lower ice waterline

K_o = $(2.44/\ell)^{1/2}$ (ℓ in m)

= $(8/\ell)^{1/2}$ (ℓ in ft), but not less than 0.4

P = design ice pressure, as defined in 6-1-5/9

Y = minimum yield stress of the material, in N/mm^2 (kgf/mm^2, ksi)

The longitudinals are to be attached and bracketed to the webs and to the bulkheads to provide a shear area at least twice the net shear area of the longitudinal.

15.5 Web Frames *(2014)*

The section modulus, *SM*, of the web frame, in association with the plating to which it is attached, is to be not less than that obtained from the following equation:

$$SM = K_o K s_1 \ell b(P/Y) \quad \text{cm}^3$$

$$SM = 0.144 K_o K s_1 \ell b(P/Y) \quad \text{in}^3$$

where

SM = required section modulus of the web frame, in cm^3 (in^3)

K_o = as defined in 6-1-5/15.3

K = 165 without struts

= 100 with one horizontal strut

= 80 with two struts

= 70 with three struts

s_1 = as defined in 6-1-5/13.7, in mm (in.)

ℓ = as defined in 6-1-5/13.7, in m (ft)

b = as defined in 6-1-5/9.5 for particular area of the ice belt, in m (ft)

P = as defined in 6-1-5/9 for particular area of the ice belt

Y = as defined in 6-1-5/11.1

In determining the section modulus, the effective width of plating is to be the distance between the webs or 0.125ℓ, whichever is less.

The net sectional area of the web plate including effective end brackets, where applicable, is to be not less than that obtained from the following equation:

$A = K_1 SM/\ell$ cm^2

$A = 8.33 K_1 SM/\ell$ in^2

where

K_1 = 0.009 without struts

= 0.015 with one or more struts

Plate thickness is to be not less than given in 6-1-5/13.7.

15.7 Struts

Where one or more struts are fitted as an effective supporting system for the ice belt structure, they are to be located within the ice belt and spaced so as to divide the supported web into spans of approximately equal length. Inboard ends of the struts are to be supported sufficiently by longitudinal bulkhead transverses having a section modulus not less than 0.9 of that required by 6-1-5/15.5. The sectional area of the strut is to be obtained from the following equation:

$A = (bs_1/K)(P/Y)K_o$ cm^2 (in^2)

where

b = as defined in 6-1-5/9.5 for particular area of the ice belt, in m (ft)

s_1 = as defined in 6-1-5/13.7, in mm (in.)

K = $0.04 - 0.0175(\ell/r)$ for SI & MKS units

= $0.0333 - 0.00175(\ell/r)$ for US units

ℓ = unsupported span of the strut, m (ft)

r = least radius of gyration, cm (in.)

P = as defined in 6-1-5/9 for particular area of the ice belt

Y = as defined in 6-1-5/11.1

K_o = as defined in 6-1-5/15.3

17 Alternative Framing Arrangements

Where framing arrangements differing from those given in 6-1-5/13 and 6-1-5/15 are used for the ice belt structures, special approval of the framing members will be based on submitted stress analysis of the structure.

19 Peak Frames *(2015)*

Main and intermediate frames in forepeaks are to extend down to the floors or the bottom transverses or the stem. The section modulus of each peak frame is to be as given in 6-1-5/13.5.1 where ℓ, in m (ft), is measured between deep ice stringers and K_1 = 1. For the forepeaks of ice classes **A0** and **B0**, the distance is not to be more than 2.1 m (7 ft).

For ice classes **A0** and **B0**, the intermediate frames are to extend down to the bottom structure and up to the first deck above the ice belt. Intermediate frames in the forepeak for ice class **C0**, **D0** and **E0** may terminate at the first stringer above the ice belt.

21 Double Bottom *(2012)*

21.1 Longitudinally Framed Bottom *(2015)*

Open floors or bilge brackets extending to the outboard longitudinals are to be fitted throughout at each frame that extends to the inner bottom, except ice classes **B0**, **C0**, **D0** and **E0**, where only the bow area is to comply with this requirement. The spacing of the bottom longitudinals within the bow, lower intermediate and lower stern areas of the ice belt is to be not more than 0.7 m (2.3 ft) for ice class **A0**.

23 Ice Decks *(2012)*

23.1 General

The following requirements apply to decks or parts of decks situated within the ice belt. For vessels not having decks in the ice belt, the following requirements apply also to decks or parts of decks above and below the ice belt to which the main and intermediate frames extend. Where there are three or more decks within the ice belt, the deck or parts of the deck situated within the upper area of the ice belt, defined in 6-1-5/7, need not comply with these requirements.

23.3 Deck Plating

The thickness of the stringer plate is to be not less than:

$$t = k(s^2 b\ P)^{1/3} \quad \text{mm (in.)}$$

where

k = 0.12 (0.257, 0.0523)

s = distance between the deck beams, in mm (in.)

b = as defined in 6-1-5/9.5, in m (ft), for the particular area of the ice belt

P = as defined in 6-1-5/9.1 or 6-1-5/9.3, for the particular area of the ice belt

The width of the stringer plate is to be not less than four times the main frame depth for **A0** ice class. For ice class **A0**, the thickness of the deck plating is to be not less than 0.75 times the required thickness of the stringer plate.

23.5 Deck Transverses and Deck Beams

23.5.1 Longitudinally Framed Decks *(2015)*

Deck transverses are to be fitted at every third main frame for ice class **A0** and at every fourth main frame for ice class **B0**.

Partial beams or brackets are to be fitted at all other main frames and at every intermediate frame for ice class **A0**, and at all other main frames for ice classes **B0**, **C0**, **D0** and **E0**. The partial beams or brackets are to be extended from the frames to a deck longitudinal or deck girder situated not less than 1.5s from the inboard edge of the frames, where s is as defined in 6-1-5/23.3.

23.5.2 Scantlings

The sectional area of the beams and deck transverses is to be not less than:

$A = K_1 sb(P/Y)\cos\beta$ cm^2

$A = 1.2K_1 sb(P/Y)\cos\beta$ in^2

The moment of inertia of the beams is to be not less than:

$MI = kK_2 s\ell^2 bP\cos\beta$ cm^4 (in^4)

where

k = 1.0 (9.81, 0.1191)

P = as defined in 6-1-5/9.1 or 6-1-5/9.3, in N/mm^2 (kgf/mm^2, ksi), for the particular area of the ice belt

b = as defined in 6-1-5/9.5, in m (ft), for the particular area of the ice belt

s = distance between the beams, in mm (in.)

ℓ = span of the beam, measured in m (ft), between the inboard edge of the frame and the deck longitudinal or deck girder supporting the beam

Y = as defined in 6-1-5/11.1

β = as defined in 6-1-5/9.1 and 6-1-5/9.3, in degrees, for the particular area of the ice belt

K_1 = 6.6

K_2 = 0.13

The sectional area and the moment of inertia of the partial beams and of the brackets are to be not less than required above. The beams and the partial beams are to be bracketed to the deck longitudinals or deck girders. Beams or partial beams or brackets fitted at the web frames are to be reinforced so that their section modulus, SM is to be not less than:

$SM = K_3 SM_{wf}\ell_{wf}/\ell$ cm^3 (in^3)

where SM_{wf} and ℓ_{wf} are the section modulus and the span of the web frame, as defined in 6-1-5/13.7, respectively.

K_3 = 0.5

23.7 Decks with Wide Openings

Within the midbody area of the ice belt, the cross sectional area of the deck outside the line of openings is to be not less than:

$A = Kb\ell(P/Y)\cdot 10^3$ cm^2

$A = 14.4Kb\ell(P/Y)$ in^2

where

K = 6.2 for ice classes **A0** and **B0**

b = as defined in 6-1-5/9.5, in m (ft), for the particular area of the ice belt

ℓ = length of the opening, in m (ft), but need not be taken as more than $0.1L$

P = as defined in 6-1-5/9.3, for the particular area of the ice belt

Y = as defined in 6-1-5/11.1

L = as defined in 6-1-5/5.9, in m (ft)

25 Bulkheads *(2012)*

25.1 Scantlings

For ice class **A0**, the thickness of that part of the bulkhead adjacent to the side shell and within the ice belt is to be not less than the thickness of the adjacent frames or of the stringers connected to the bulkhead, whichever is greater. The width of these parts of the bulkhead is to be not less than shown in 6-1-5/Table 10. These parts of the bulkhead adjacent to the shell within the ice belt are to be fitted with stiffeners normal to the shell plating. The stiffeners are to be welded to a vertical bulkhead stiffener and welded and bracketed to the side longitudinals. Where the shell is transversely framed, brackets are to be welded to the shell and extended and attached to adjacent frames.

TABLE 10
Minimum Width of Reinforced Bulkhead Plating *(2012)*

Ice Class	*Area of the Ice Belt*			
	Peak Bulkheads m (ft)	*Bow and Intermediate Areas m (ft)*	*Midbody Area m (ft)*	*Stern Area m (ft)*
A0	1.2 (4.0)	1.2 (4.0)	1.0 (3.3)	1.0 (3.3)

27 Stem and Stern Frame

27.1 General *(2012)*

The requirements of Section 3-2-13 of the Rules are to be complied with. The stem and stern frame for ice class **A0** vessels of displacements more than 50,000 tonnes (49,200 Lt) are to be constructed of rolled bar, cast or forged steel. Shaped plate stem may be used elsewhere. All joints and connections are to fully develop the strength of the stem and stern frame. All rudders are to be protected against ice impacts for going astern.

27.3 Stem

27.3.1 Solid Stem

The cross sectional area of a stem made of rolled bar, cast or forged steel from the center vertical keel to $0.01L$ above the ice belt is to be not less than:

$$A = K_1 D^{1/3} (L - 61) + A_o \quad \text{cm}^2$$

$$A = 0.0473 K_1 D^{1/3} (L - 200) + A_o \quad \text{in}^2$$

where

K_1 and A_o = as given in 6-1-5/Table 11

D = as defined in 6-1-5/5.7

L = as defined in 6-1-5/5.9, in m (ft), but is not to be taken less than 61 m (200 ft)

For vessels of displacements less than 2,500 tonnes (2,460 Lt) the cross sectional area given by the above equation may be reduced 10%. The cross sectional area of the stem above the ice belt may be reduced gradually to the value given in Section 3-2-13.

TABLE 11
Solid Stem Bar Coefficients *(2015)*

Ice Class	A_o cm^2 (in^2)	K_1
A0	62 (9.6)	0.13
B0	50 (7.8)	0.705
C0	45 (7.0)	0.095
D0	45 (7.0)	0.095
E0	45 (7.0)	0.095

27.3.2 Shaped Plate Stem *(2015)*

Thickness of shaped plate stems within the bow area of the ice belt is to be not less than

$$t = 0.8s(P/Y)^{1/2} + t_o \text{ but not less than } 0.04R.$$

where

t = required thickness of plate stem, in mm (in.)

s = distance between frames, brackets (breast hooks) or stiffeners, in mm (in.)

P, Y and t_o are as defined in 6-1-5/11.1.

R = inside radius of the stem at the given section, in mm (in.). Need not be taken greater than 625 mm (24.6 in.) for ice classes **A0** through **E0**

At any section, the fore and aft length of the stem plate is to be not less than 15t.

27.3.3 Arrangement

The outer surface of connections of the shell plating to the stem is to be flush. The stem is to be supported by floors, webs, frames, breasthooks or brackets spaced not more than 610 mm (24 in.). In addition, shaped plate stems are to be supported in the centerline by a plate, web or bulkhead having the same thickness as the center vertical keel and a width not less than 610 mm (24 in.).

27.5 Stern Frame

The stern post is to be of size obtained from 3-2-13/3.5 through 3-2-13/3.11, with all thicknesses increased by coefficient K, as given in 6-1-5/Table 12. In addition, factors C_f and C_c in 3-2-13/3.5 are to be multiplied by K^2.

TABLE 12
Stern Post Coefficient *(2015)*

Ice Class	K
A0	1.2
B0	1.12
C0	1.07
D0	1.05
E0	1.05

29 Power of Propulsion Machinery *(2012)*

29.1 Minimum Power

For ice classes **A0** through **C0**, the total ahead horsepower delivered to the propellers, *N*, is to be not less than the lesser of the values obtained from the following two equations:

i) $N = kA(B)^{0.8}(L)^{0.4}(1 + me^{-5D \times 10^{-6}})$ kW (mhp, hp)

ii) $N = k(C + KD \times n/1000)$

where

B = maximum breadth of the vessel, in m (ft), at the upper ice waterline

L = length of the vessel, in m (ft), as defined in 6-1-5/5.9

e = base of natural logarithms

D = as defined in 6-1-5/5.7

n = 1 (1.016)

k = 0.735 (1, 0.986)

A, m, C and K are as given in 6-1-5/Table 13.

For vessels with unconventional features, the power delivered to the propellers may also be less than given in equation *i)*, if the particular vessel is able to progress continuously in any ice condition corresponding to its ice class. Special approval of this will be based on necessary evidence including the submission of results of full-scale and model tests. Special consideration will be given when the value of N determined from the equations in 6-1-5/29.1 is less than N_o in 6-1-5/Table 13.

TABLE 13
Power Coefficients *(2012)*

The power given may be reduced up to 10% for vessels fitted with controllable pitch propellers

Ice Class	*A* *SI & MKS (US units)*	*m*	*C*	*K*	N_o *kW (mhp, hp)*
A0	93 (22.4)	0.6	1000	350	1,490 (2,030, 2,000)
B0	79 (19.0)	0.6	500	300	746 (1,040, 1,000)
C0	64 (15.4)	0.6	0	250	373 (507, 500)

29.3 Astern Power *(2015)*

The astern power delivered to the propellers for ice classes **A0** to **C0** is to be not less than 70% of that required in 6-1-5/29.1. For ice class **D0** and **E0**, see 4-1-1/7.5, as applicable.

31 Non-self-propelled Vessels

31.1 General

Barges designed for being towed and/or pushed in broken ice and built to the requirements of this Section and related sections of the ABS *Rules for Building and Classing of Steel Barges* will be designated by ice classes **A0**, **B0**, **C0** and **D0**. Non-self-propelled vessels other than barges covered by these Rules will be subject to special consideration.

31.3 Ice Classes

For the guidance of the Owner, the ice conditions considered appropriate for towing or pushing barges are shown below:

TABLE 14
Ice Conditions for Towing or Pushing Barges *(2012)*

Ice Class	*Towed/Pushed*	*Towed by ice class **PC5** vessel**	*Towed by ice class **PC4** vessel**
A0	severe	very severe	extreme
B0	medium	severe	
C0	light	medium	
D0	very light	light	

* Breadth of towed barge not to exceed the breadth of towing vessels.

Barges intended to be pushed in "very severe" or "extreme" ice conditions will be subject to special consideration.

31.5 Ice Belt

The ice belt is divided into three parts: bow, midbody and aft areas, except that for class **D0**, the ice belt applies to bow area only. For barges designed for tow by either end, bow area requirements apply to both ends. For such barges, the midbody and two bow areas of the ice belt are to be used. The bow area of the ice belt is to extend forward from the section 0.025L aft of either the point where the rake reaches the bottom or where the lightest ice waterline reaches its greatest breadth, whichever is greater. The aft area of the ice belt is to extend aft of the section 0.025L forward of the point where the lightest ice waterline reaches its greatest breadth. The midbody area of the ice belt extends between the bow and aft areas.

Upper boundary of the ice belt throughout the length of the barge is to be not less than 0.75 m (30 in.) above the deepest ice waterline for ice class **A0** and not less than 0.6 m (24 in.) above the deepest ice waterline for ice classes **B0** and **C0** and not less than 0.5 m (20 in.) above the deepest ice waterline for ice class **D0**. The lower boundary of the ice belt is to be not less than 0.6 m (24 in.) below the lightest ice waterline for the midbody and aft areas of ice class **A0**. In the bow area of ice class **A0**, the ice belt is to extend to the bottom of the side shell and is to include the bottom shell in way of the rake. For ice classes **B0**, **C0** and **D0**, the lower boundary of the ice belt is to be not less than 0.5 m (20 in.) below the lightest ice waterline throughout the length of the barge.

31.7 Design Ice Loads

The design ice pressure on the bow area, P_{bow}, is to be as given for P_b in 6-1-5/9.1, where F_{b1} = 1.25 for vertical structures and F_{b1} = 1 for the rakes. The design ice pressures on the midship and aft areas, P_{mid} and P_{aft} are to be:

$$P_{mid} = K_m P_{bow}$$

$$P_{aft} = K_s P_{bow}$$

where K_m and K_s are as given in 6-1-5/Table 5.

The vertical extent of the design ice pressure for all of the ice belt areas is to be:

0.61 m (24 in.) for ice class **A0**

0.51 m (20 in.) for ice class **B0**

0.45 m (18 in.) for ice class **C0**

0.40 m (16 in.) for ice class **D0**

31.9 Structural Arrangements

The thickness of the shell plating within the ice belt areas is to be as required by 6-1-5/11.1 or 6-1-5/11.3. Structural arrangements and scantlings of the ice belt framing members are to be as required by 6-1-5/13, 6-1-5/15 and 6-1-5/19. Decks and bulkheads situated within the ice belt and, where there are no decks within the ice belt, the deck above and below the ice belt to which the main and intermediate frames are extended are to comply with the requirements of 6-1-5/23 and 6-1-5/25.

33 Hull Structural Materials

33.1 General

All hull structural materials are to be in accordance with the requirements of Part 2, Chapter 1. In addition, material grades for ice belt structures and exposed shell and main strength deck structures are to be selected based on the design service temperature and material class, as defined as follows.

33.3 Design Service Temperature

The design service temperature is to be taken in accordance with 6-1-5/Table 15. Design service temperature for insulated members will be specially considered upon submission of substantiating data.

TABLE 15
Design Service Temperature, degrees C (degrees F) *(2015)*

		Ice Class		
	Zones	***A0***	***B0** and **C0***	***D0** and **E0***
a.	Ice Belt Structures (other than Area c)			
	1. External plating	-30 (-22)	-20 (-4)	-10 (14)
	2. Framing [(1)] for all items above	-20 (-4)	-10 (14)	0 (32)
b.	Above Ice Belt [(3)]			
	1. External plating	-30 (-22)	-20 (-4)	-10 (14)
	2. Framing [(1)] for external plating	-20 (-4)	-10 (14)	0 (32)
	3. Plating [(2)] and framing in enclosed spaces			
	i) Heated space	0 (32)	0 (32)	0 (32)
	ii) Unheated space	-10 (14)	0 (32)	0 (32)
c.	More than 0.3 m (1 ft) below the lower ice waterline.	0 (32)	0 (32)	0 (32)

Notes:

1 Includes bulkheads and decks attached to the external plating within 600 mm (23.5 in.) from the plating.

2 Excludes those portions covered by Note 1 above.

3 *(2015)* Above Area c for class **D0** and **E0** excluding the bow area.

33.5 Material Class of Structural Members

The material class of hull structural members is to be in accordance with 6-1-5/Table 16.

TABLE 16
Material Class of Structural Members *(2015)*

Material class given in this table refers to the classes in 6-1-5/Table 17 or in 6-1-5/33.9, as applicable.

	Structural Members	*Ice Classes* **A0**	***B0** and below*
a.	Within Ice Belt (other than Area c)		
	1. Bottom and side shell plating-bow, intermediate and lower intermediate areas	III	I
	2. Bottom and side shell plating-other ice belt areas	II	I
	3. Framing (1) – bow and intermediate areas	II	I
	4. Framing (1) – other ice belt areas	I	I
	5. Stem, ice knife, propeller nozzle, shaft bracket, rudder, stern frame and rudder horn	III	I
	6. Other structures	I	I
b.	Above Ice Belt		
	1. Sheer strake and deck stringer		
	i) within 0.4*L* amidships	III	III
	ii) outside 0.4*L* amidships	II	II
	2. Side shell (4) and strength deck plating (2),(3)(5)	I	I
	3. Other structures (2),(3)	I	I
c.	More than 0.3 m (1 ft) below the lower ice waterline.	No additional requirements for ice class. See 3-1-2/Table 2	

Notes:

1 Includes bulkheads and decks attached to the external plating within 600 mm (23.5 in.) from the plating.

2 Excludes those portions covered by Note 1 above.

3 *(2015)* Above Area c for class **D0** and **E0** excluding the bow area.

4 *(2010)* Single side strakes for ships exceeding 150 m (492 ft) without inner continuous longitudinal bulkheads between bottom and the single strength deck are not to be less than grade B/AH within cargo region in ships.

5 *(2010)* Not to be less than grade B/AH within 0.4*L* amidships in ships with length exceeding 150 m (492 ft) and single strength deck.

33.7 Criteria for ABS Grade Steels

For those rolled steel products in 2-1-2/Table 5 or 2-1-3/Table 5, the appropriate grade to be used for respective material class and thickness is shown in 6-1-5/Table 17a through 6-1-5/Table 17c. Where 3-1-2/3 results in a higher grade, such higher grade is to be used.

TABLE 17a
Material Grades – Class I

Thickness in mm (in.)	*Design Service Temperature* 0°C (32°F)	-10°C (14°F)	-20°C (-4°F)	-30°C (-22°F)	-40°C (-40°F)
$t < 12.5$ ($t < 0.50$)	A,AH	A,AH	A,AH	A,AH	B (2),AH
$12.5 < t \le 20$ ($0.50 < t \le 0.79$)	A,AH	A,AH	A,AH	B,AH	D,DH
$20 < t \le 25$ ($0.79 < t \le 0.98$)	A,AH	A,AH	B,AH	D,DH	D (1),DH (1)
$25 < t \le 30$ ($0.98 < t \le 1.18$)	A,AH	A,AH	D,DH	D,DH	E,EH
$30 < t \le 35$ ($1.18 < t \le 1.38$)	A,AH	B,AH	D,DH	D,DH	E,EH
$35 < t \le 40$ ($1.38 < t \le 1.57$)	A,AH	D,DH	D,DH	D,DH	E,EH
$40 < t \le 51$ ($1.57 < t \le 2.00$)	B,AH	D,DH	D,DH	D,DH	E,EH

Notes:

1 To be normalized.

2 May be "A" if fully killed.

TABLE 17b
Material Grades – Class II

	Design Service Temperature				
Thickness in mm (in.)	*0°C (32°F)*	*-10°C (14°F)*	*-20°C (-4°F)*	*-30°C (-22°F)*	*-40°C (-40°F)*
$t \le 12.5$ ($t \le 0.50$)	A,AH	A,AH	A,AH	B (2),AH	D,DH
$12.5 < t \le 20$ ($0.50 < t \le 0.79$)	A,AH	A,AH	B,AH	D,DH	D (1),DH (1)
$20 < t \le 25$ ($0.79 < t \le 0.98$)	A,AH	B,AH	D,DH	D (1),DH (1)	E,EH
$25 < t \le 30$ ($0.98 < t \le 1.18$)	A,AH	B,AH	D,DH	E,EH	E,EH
$30 < t \le 35$ ($1.18 < t \le 1.38$)	B,AH	D,DH	D,DH	E,EH	E,EH
$35 < t \le 40$ ($1.38 < t \le 1.57$)	B,AH	D,DH	D,DH	E,EH	E,EH
$40 < t \le 51$ ($1.57 < t \le 2.00$)	D,DH	D,DH	D,DH	E,EH	E,EH

Notes:
1 To be normalized.
2 May be "A" if fully killed.

TABLE 17c
Material Grade – Class III

	Design Service Temperature				
Thickness in mm (in.)	*0°C (32°F)*	*-10°C (14°F)*	*-20°C (-4°F)*	*-30°C (-22°F)*	*-40°C (-40°F)*
$t < 12.5$ ($t < 0.50$)	A,AH	A,AH	B (2),AH	D,DH	D (1),DH (1)
$12.5 < t \le 20$ ($0.50 < t \le 0.79$)	A,AH	B,AH	D,DH (1)	D (1),DH (1)	E,EH
$20 < t \le 25$ ($0.79 < t \le 0.98$)	B,AH	D,DH	D (1),DH (1)	E,EH	E,EH
$25 < t \le 30$ ($0.98 < t \le 1.18$)	B,AH	D,DH	E,EH	E.EH	E,EH
$30 < t \le 35$ ($1.18 < t \le 1.38$)	D,DH	D,DH	E,EH	E,EH	—
$35 < t \le 40$ ($1.38 < t \le 1.57$)	D,DH	D,DH	E,EH	E,EH	
$40 < t \le 51$ ($1.57 < t \le 2.00$)	D,DH	D,DH	E,EH	E,EH	

Notes:
1 To be normalized.
2 May be "A" if fully killed.

33.9 Criteria for Other Steels

33.9.1 Yield Strength Below 410 N/mm^2, (42 kgf/mm^2, 60 ksi)

Where steels other than those in 2-1-2/Table 5 or 2-1-3/Table 5 are intended, their specifications are to be submitted for approval. These steels are to comply with the following impact test requirements:

Yield Strength		
N/mm^2	*(kgf/mm^2)*	*(ksi)*
235-305	(24-31)	(34-44)
315-400	(32-41)	(45.5-58)

CVN (Longitudinal)		
J	*(kgf-m)*	*(ft-lbf)*
27	(2.8)	(20)
34	(3.5)	(25)

At the following temperatures:

Class I – design service temperature

Class II – 10°C (18°F) below design service temperature

Class III – 20°C (36°F) below design service temperature

33.9.2 Yield Strength 410-690 N/mm^2 (42-70 kgf/mm^2, 60-100 ksi)

Where steels of this strength level are intended, their specifications are to be submitted for approval. These steels are to comply with the impact test requirements of 34 J (3.5 kgf-m, 25 ft-1bf) at the following temperatures:

Design Service Temperature	*Test Temperature*
0°C (32°F)	-30°C (-22°F)
-10°C (14°F)	-40°C (-40°F)
-20°C (-4°F)	-40°C (-40°F)
-30°C (-22°F)	-50°C (-58°F)
-40°C (-40°F)	-60°C (-76°F)

33.9.3 Alternative Requirements

As an alternative to the requirements in 6-1-5/33.9.1 and 6-1-5/33.9.2, higher strength steels may comply with the following:

i) For transverse specimens, $^2/_3$ of energy values shown in 6-1-5/33.9.1 and 6-1-5/33.9.2.

ii) For longitudinal specimens, lateral expansion is not to be less than 0.5 mm (0.02 in.). For transverse specimens, lateral expansion is not to be less than 0.38 mm (0.015 in.).

iii) Nil-ductility temperature (NDT), as determined by drop weight tests, is to be 5°C (9°F) below the temperature specified in 6-1-5/33.9.1 and 6-1-5/33.9.2.

33.11 Weld Metal

33.11.1 ABS Hull Steels

When the ABS ordinary and higher strength hull steels of 2-1-2/Table 5 or 2-1-3/Table 5 are applied in accordance with 6-1-5/Table 17a through 6-1-5/Table 17c, approved filler metals appropriate to the grades shown in Part 2, Appendix 3 may be used.

33.11.2 Criteria for Other Steels

For the welding of hull steels other than the ABS grades in 6-1-5/Table 17a through 6-1-5/Table 17c, weld metal is to exhibit a Charpy V-Notch toughness value at least equivalent to the transverse base metal requirements ($^2/_3$ of longitudinal base metal requirements).

35 Weld Design *(1997)*

Weld design of hull construction is to comply with Section 3-2-19. Special attention is to be paid to welds in structures attached to side shell, such as transverse bulkheads, decks, frames, web frames and side shell stringers, within the ice belt, which are to be of double continuous weld.

37 Towing Arrangements *(2012)*

37.1 Bow *(2015)*

Every ice class vessel intended to be escorted by a higher ice class leading vessel, as given in 6-1-5/Table 1, is to be fitted with a tow chock pipe and a tow bitt on the bow. The chock and the bitt are to be properly connected to the stem frame. The portions of the decks at which the chock and the bitt are attached are to meet requirements of 6-1-5/23. The shell plating and framing below and 1.5 m (5 ft) around the chock are to be as required by 6-1-5/11 and 6-1-5/13 for the bow area of the ice belt for ice classes **A0**, **B0**, **C0**, **D0** and **E0.** The stem frame below the connections with the chock is to be as required by 6-1-5/27.3 for the portion of the stem within the ice belt.

Where a bulbous bow is fitted, the bulb is not to extend beyond the fore end of the lower ice waterline specified by 6-1-5/5.5.

39 Propeller Nozzles

39.1 General *(2015)*

This Subsection applies to fixed nozzles. Special consideration will be given to steering nozzles for ice class **A0**. For ice class **A0**, the nozzles are to be supported at least at the upper and lower ends. For ice classes **B0**, **C0**, **D0** and **E0**, the nozzles supported only at the upper ends are to be attached to the hull for a width of not less than $^1/_6$ of the outer circumference of the nozzle. The strength, rigidity and resistance to buckling of the nozzle are to be adequate for the design ice forces given in 6-1-5/39.3. All of the critical loading cases are to be considered. In no case under the design ice forces are the normal and axial displacements of the inside ring to exceed 10% of the clearance between the inside plating of the nozzle and the propeller blade tips, or 0.5% of the inside ring diameter, whichever is less. Nozzles are to be protected by stern structures as much as possible against direct impacts with large ice features.

39.3 Design Ice Forces

The design ice forces are to be not less than those obtained from the following equations:

$$F_n = K_1 K_2 (D\, d_1)^{1/2} \quad \text{kN (tf, Ltf)}$$

$$F_f = K_3 K_4 [D\,(d_1 - d_2)]^{1/2} \quad \text{kN (tf, Ltf)}$$

where

F_n	=	design ice force applied normal to the outside surface of the nozzle in the most critical location	
K_2	=	1	for the external sides of a single nozzle of a single screw vessel
	=	1.1	for the outboard external sides of the outermost nozzles of vessels with two or more screws
	=	0.25	for the external sides of nozzles situated between the outermost ones and for the internal sides of any nozzles
	=	0.8	for bottoms of the nozzles
D	=	ship displacement, in tonnes (long tons), as specified in 6-1-5/5.7	
d_1	=	maximum outer diameter of the nozzle, in m (ft)	
d_2	=	minimum internal diameter of the nozzle, in m (ft)	
F_f	=	design ice force applied to the ends of the nozzle, parallel to the propeller axis, in the most critical locations	
K_4	=	1	for aft end face of the nozzle having no rudder behind
	=	0.7	for the aft end face of the nozzle with a rudder behind
	=	0.6	for the fore end face of the nozzle

K_1 and K_3 are as given in 6-1-5/Table 18.

Values of K_2 and K_4 less than above will be approved, provided the stern and bottom hull structures effectively protect the nozzle against large ice fragments.

TABLE 18
Design Ice Force Coefficient *(2015)*

Ice Class	K_1	K_3
	SI units (MKS, US)	*SI units (MKS, US)*
A0	20 (2.1, 1.1)	59 (6.0, 3.3)
B0	13 (1.3, 0.7)	35 (3.6, 2.0)
C0	9 (0.9, 0.5)	22 (2.2, 1.2)
D0	7 (0.7, 0.4)	18 (1.8, 1.0)
E0	6 (0.6, 0.3)	16 (1.6, 0.9)

39.5 Plate Thickness

The plate thickness of both inner and outer surfaces of the nozzle is to be not less than required by 6-1-5/11.1 for the stern ice belt area with coefficient $C = 0.3$. A value of $C = 0$ will be considered for a high abrasion-resistant coating of the nozzle. In this case, the results of operational experience information, required in the note to 6-1-5/Table 7, are to be submitted.

41 Rudder and Steering Arrangements

41.1 General *(1993)*

41.1.1 All Ice Classes, Multiple Rudders

Where two or more rudders are provided, they are to be mechanically independent.

41.1.2 Ice Class **A0** *(2012)*

41.1.2(a) Pintles. Rudders are to have at least two pintles.

41.1.2(b) Locking. Rudders are to be protected by strong and effective external rudder stops and provided with mechanical means of locking the rudder parallel to the centerline for use in the astern condition.

41.1.3 Ice Classes **A0** through **B0** *(2012)*

41.1.3(a) Ice Knife. Rudders are to be protected by ice knives or other similar structures located abaft the rudder. Clearance between the ice knife and the rudder is not to exceed 100 mm (4 in.)

41.3 Rudder Stocks, Couplings and Pintles *(2012)*

41.3.1 Ice Classes **A0** through **E0** *(2015)*

For ice classes **A0** through **E0**, rudder stocks, pintles, gudgeons and other bolting arrangements to the stern frames are to meet the requirements in Section 3-2-14 in association with V_i as defined below, in lieu of V.

V_i = the greater of V, as defined in Section 3-2-14, or the minimum design speed in 6-1-5/Table 19

TABLE 19
Design Speed for Rudders, Couplings and Pintles *(2015)*

Ice Class	*Minimum Design Speed, knots*
A0	18
B0	16
C0	14
D0	12
E0	12

41.3.2 Ice Class **A0**

The stresses in these members with the load F applied as follows are not to exceed the shear yielding strength which may be taken as 0.577 times the specified yield point of the material.

$$F = 2K_3(Dt)^{1/2} \quad \text{kN (tf, Ltf)}$$

where

K_3 = as given in 6-1-5/Table 18

D = ship displacement, in tonnes (long tons), as specified in 6-1-5/5.7

t = thickness of the rudder, in m (ft), measured at the level of F and at 10% of the rudder length from the trailing edge.

F is to be applied to the after edge of the rudder in a direction parallel to the centerline of the vessel at all locations below the ice waterline within the middle 40% of the rudder height to determine the most severe requirements. Alternatively, F may be spread over any 60% of the rudder height as a uniform load. No other force need be considered simultaneously with F.

41.5 Double Plate Rudder

For double plate rudders, the minimum thickness of plates is to be not less than required by 6-1-5/39.5.

43 Bossings

The bossings are to be designed to withstand the design ice forces F_n, as specified by 6-1-5/39.3, where d_1 is the diameter of the bossing. The bossing plating thickness is to be not less than required by 6-1-5/11.1 for the stern ice belt area, where s is the distance between stiffeners.

45 Machinery Arrangements *(2012)*

45.1 General

All machinery is to be suitable for operation under the environmental conditions to which it will be exposed in service and is to include all necessary special provisions for that purpose.

45.3 Governmental Authority

Attention is directed to the appropriate governmental authorities in the intended regions of operation for additional requirements in consideration of operation in ice such as fuel capacity, refueling capability, water capacity, radio communications requirements, etc.

45.5 Propulsion Arrangements

In addition to the regular governor, all propulsion engines and turbines are to be fitted with a separate overspeed device so adjusted that the speed cannot exceed the maximum rated speed by more than 20%.

45.7 Electric Propulsion

Propulsion motors are to be fitted with automatic protection against excessive torque, overloading and temperature. This protection is to automatically limit these parameters, but is not to cause loss of propulsion power.

45.9 Boilers

Vessels propelled by steam machinery are to be fitted with at least two boilers of equal capacity.

45.11 Protection Against Excessive Torques

For vessels of all classes, if torsionally flexible couplings or torque-limiting devices are fitted in the propulsion system, positive means are to be provided for transmitting full torque to the propeller in the event of failure of the flexible element. Ratings for flexible couplings are to be in accordance with 6-1-5/57.

45.13 Sea Chests *(2015)*

For vessels of Ice Class **A0**, **B0**, **C0**, **D0** and **E0**, at least one sea chest for supplying water for cooling and fire-fighting purposes is to be connected to the cooling-water discharge by a branch pipe having the same cross sectional area as the main pipe-line, in order to stay free from ice and slush ice. As far as practicable, the sea inlet chest is to be situated well aft, adjacent to the keel.

47 Materials for Propellers and Propulsion Shafting *(2018)*

Propeller materials are to be in accordance with the applicable requirements of 4-3-3/3.

In addition to the applicable requirements of 4-3-2/3, the material, for propeller shafts and other shafting that are exposed to sea water, is to have a Charpy V-notch impact value of not less than 20.5 J (2.1 kgf-m, 15 ft-lbf) at a temperature of -10°C (14°F) for all ice classes, except ice class **D0** and **E0**. The propulsion shafts and couplings are to be made of steel.

49 Determination of Ice Torque for Propulsion Systems *(2012)*

The Ice Torque M for determining the dimensions of propellers and gears is to be in accordance with 6-1-5/Table 20 and associated notes.

51 Propellers

51.1 Propeller Arrangements *(2012)*

Propeller arrangements, the shape of the stern and the propeller protecting structures are to be adequate for the intended service. Special consideration is to be given to the propeller protection when moving astern.

TABLE 20
Value of Ice Torque M *(2015)*

Location of Propeller	*Centerline*	*Off Centerline*
Propellers protected by nozzle		
Nozzle protected (see Note 1)		
class **A0**-**B0**	$0.75M_1$ (see Note 2)	$0.85M_1$ (see Note 2)
class **C0**-**E0**	$0.85M_1$	$0.9M_1$
Nozzle unprotected		
class **A0**-**C0**	$0.9M_1$ (see Note 3)	$0.9M_1$ (see Note 3)
class **D0** and **E0**	$0.9M_1$	$0.9M_1$
Open propellers		
class **A0**-**E0**	M_1	M_1

M_1 = mD^2, in kN-m (tf-m, Ltf-ft)

m = value from 6-1-5/Table 21

D = propeller diameter, in m (ft)

Notes:

1 These requirements apply where the nozzle is well protected by ice knives, fins or other adequate stern arrangement from large ice fragments entering into nozzle from forward or backward motion of the vessel. These reductions are subject to special consideration.

2 To be not less than required for the second lower ice class.

3 To be not less than required for the next lower ice class.

4 Need not be greater than required for next higher ice class.

TABLE 21
Values of *m* (2015)

Ice Class	*SI units*	*MKS units*	*US units*
A0	15.7	1.60	0.48
B0	13.0	1.33	0.40
C0	12.1	1.23	0.37
D0	11.1	1.13	0.34
E0	8.8	0.90	0.27

51.3 Propeller Section

51.3.1 Width and Thickness *(2018)*

The thickness T and width W of propeller blade sections are to be determined so that the WT^2 calculated by the actual designed W and T is not less than that required by the following equations:

- *At the 0.25 radius for solid propellers*

  $WT^2 = [a_1/U(0.65 + 0.7P_{0.25})]\ [(a_2CN/nR) + a_3M]$ cm^3 (in^3)

- *At the 0.35 radius for solid propellers with hubs larger than 0.25 propeller diameter*

  $WT^2 = [a_4/U(0.65 + 0.7P_{0.35})]\ [(a_2CN/nR) + a_5M]$ cm^3 (in^3)

- *At the 0.35 radius for controllable-pitch propellers*

  $WT^2 = [a_4/U(0.65 + 0.49P_{nominal})]\ [(a_2CN/nR)+a_5M]$ cm^3 (in^3)

- *At the 0.6 radius for solid propellers*

  $WT^2 = [a_6/U(0.65 + 0.7P_{0.6})]\ [(a_2CN/nR) + a_7M]$ cm^3 (in^3)

- *At the 0.6 radius for controllable-pitch propellers*

  $WT^2 = [a_6/U(0.65 + 0.49P_{nominal})]\ [(a_2CN/nR) + a_7M]$ cm^3 (in^3)

where

a_1 = 2650 (270, 27000)

a_2 = 272 (200, 176)

a_3 = 22.4 (220, 59.134)

a_4 = 2108 (215, 21500)

a_5 = 23.5 (230, 61.822)

a_6 = 932 (95, 9500)

a_7 = 28.6 (280, 75.261)

W = expanded width of a cylindrical section at the appropriate radius, cm (in.)

T = maximum thickness at the appropriate radius from propeller drawing, cm (in.)

U = tensile strength of propeller material, N/mm^2 (kgf/mm^2, psi)

P = pitch at the appropriate radius divided by the propeller diameter (for controllable-pitch propellers, the nominal value of pitch is to be used)

C	=	1	for $N \le 7{,}460$ kW (10,140 mhp, 10,000 hp)
	=	$0.667 + \dfrac{N}{22380}$	for 7,460 kW $< N <$ 29,840 kW
	=	$0.667 + \dfrac{N}{30420}$	for 10,140 mhp $< N <$ 40,560 mhp
	=	$0.667 + \dfrac{N}{30000}$	for 10,000 hp $< N <$ 40,000 hp
	=	2	for $N \ge 29{,}840$ kW (40,560 mhp, 40,000 hp)
N	=	as defined in 6-1-5/9.1, per propeller	
n	=	number of blades	
R	=	rpm at the maximum continuous rating	
M	=	ice torque, as defined in 6-1-5/49	

51.3.2 Blade Tip Thickness *(2015)*

The minimum blade thickness t_a, in mm (in.), at the tip of the blade (D/2) is to be determined from the following equations:

- *For Classes* ***A0****,* ***B0****,* ***C0****,* ***D0*** *and* ***E0***

$$t_a = (a_4 + a_2 D)\sqrt{a_3 / U} \quad \text{mm (in.)}$$

where

a_2	=	2 (2, 0.024)
a_3	=	490 (50, 71000)
a_4	=	15 (15, 0.591)
D	=	propeller diameter, m (ft)
U	=	tensile strength of the propeller material, N/mm^2 (kgf/mm^2, psi)

51.3.3 Blade Bolts

For built-up or controllable-pitch propellers, the cross sectional area of the bolts at the root of the thread is to be determined by the following equation:

$$\alpha = 0.082UWT^2/U_b nr$$

where

α	=	area of each bolt at root of thread, in mm^2 (in^2)
U	=	tensile strength of the propeller material, N/mm^2 (kgf/mm^2, psi)
U_b	=	tensile strength of the bolt material, N/mm^2 (kgf/mm^2, psi)
n	=	number of bolts on one side of blade (if n is not the same on both sides of the blade, the smaller number is to be used.)
r	=	radius of bolt pitch circle, in mm (in.)

W and T are as defined in 6-1-5/53, in mm (in.).

51.5 Additional Requirements

51.5.1 Rule Required Thickness

Where the blade thickness derived from the equations in 6-1-5/51.1 is less than the required thickness detailed in 4-3-3/5.1 through 4-3-3/5.7, the latter is to be used.

51.5.2 Other Sections

The thicknesses of propeller sections at radii intermediate to those specified are to be determined from fair curves connecting the required section thicknesses.

51.5.3 Blade Edges *(1999)*

The thickness of blade edges is not to be less than 50% of the required tip thickness t_a, measured at a point $1.25t_a$ from the leading edge for controllable-pitch propellers, and from each edge for solid propellers.

51.5.4 Controllable-pitch Propellers

The strength of the internal mechanisms of controllable-pitch propellers is to be at least 1.5 times that of the blade in the weakest direction of the blade for a load applied on the blade at the 0.9 radius and at an offset from the blade spindle axis equal to two-thirds the distance from the spindle axis to the leading or trailing edge (whichever is greater, as measured at the 0.9 radius).

51.5.5 Highly Skewed Propellers

Where highly skewed propellers are utilized, stress calculations considering both the ahead and astern operating conditions as well as the above ice loads are to be submitted for review.

51.7 Friction Fitting of Propeller Hubs and Shaft Couplings

Friction fitting of propeller hubs, shaft couplings or other torque transmitting components in those portions of the shaft line subject to shock loading from the propeller, is to have a factor of safety against slip considering both propulsion torque and ice torque of at least 2.4. Detailed stress and fitting calculations for all friction-fitted components are to be submitted for review. See 4-3-3/5.15.2(c).

53 Propulsion Shafting Diameters *(2018)*

The diameters of the propulsion shafts are to be not less than that obtained from the following equation:

$$d = k_o k_1 (W_a T_a^2 U/Y)^{1/3} \quad \text{cm (in.)}$$

where

d = diameter of the shaft being considered, measured at its aft bearing, cm (in.)

k_o = 1.00

k_1 = as given in 6-1-5/Table 22

W_a, T_a = actual values of the propeller blade expanded width and maximum thickness measured at the blade section at the 0.25 radius for solid propellers with the propeller hub not larger than $0.25D$ and at the 0.35 radius otherwise; in cm (in.)

U = tensile strength of the propeller material, N/mm^2 (kgf/mm^2, psi)

Y = yield strength of the shaft steel, N/mm^2 (kgf/mm^2, psi)

TABLE 22
Propulsion Shaft Diameter Factor k_1

	Solid Propellers with Hubs	
	Not Larger than 0.25D	*Larger than 0.25D and CPP's*
Tail shaft	1.08	1.15
Tube shaft	1.03	1.10
Intermediate shaft(s)	0.87	0.95
Thrust shaft	0.95	1.01

55 Reduction Gears *(2006)*

Pinions, gears and gear shafts are to be designed to withstand an increase in torque over that normally required for ice-free service. The following corrected ice torque (T_i) is to be utilized in Section 4-3-1.

$$T_i = T + C[MI_H R^2/(I_L + I_H R^2)]$$

where

T_i = ice corrected torque, N-m (kgf-cm, lbf-in)

T = torque corresponding to maximum continuous power, N-m (kgf-cm, lbf-in)

M = ice torque, as defined in 6-1-5/49, kN-m (tf-m, Ltf-ft)

I_H = sum of mass moment of inertia of machinery components rotating at higher rpm (drive side)

R = gear ratio (pinion rpm/gear wheel rpm)

I_L = sum of mass moment of inertia of machinery components rotating at lower rpm (driven side) including propeller with an addition of 30% for water

C = 1000 (100,000, 26800)

I_H and I_L are to be expressed in the same units.

For calculations in Appendix 4-3-1A1, for diesel engine propulsion, $K_{Aice} = T_i/T$. If $K_{Aice} > K_A$ per 4-3-1A1/11, apply K_{Aice}. If $K_{Aice} < K_A$ per 4-3-1A1/11, apply K_A.

57 Flexible Couplings

Torsionally flexible couplings are to be selected so that the ice-corrected torque, as determined in 6-1-5/55, does not exceed the coupling manufacturer's recommended rating for continuous operation. When the rotating speed of the coupling differs from that of the propeller, the ice-corrected torque is to be suitably adjusted for the gear ratio. If a torque-limiting device is installed between the propeller and the flexible coupling, the maximum input torque to the torque-limiting device may be taken as the basis for selecting the coupling, in lieu of the ice-corrected torque. Flexible couplings which may be subject to damage from overheating are to be provided with temperature-monitoring devices or equivalent means of overload protection with alarms at each engine control station.

59 Tunnel Thrusters *(2018)*

Where **APS**, **PAS**, or Dynamic Positioning Systems Notations are assigned, the mechanical components of a tunnel thruster (i.e., propellers, gears, shafts, couplings, etc.) are to meet the applicable requirements of propulsion systems in this Section.

Alternatively, Section 4-3-5 may be applied to the mechanical components of a tunnel thruster when a comprehensive study to determine the effect of ice is submitted for consideration.

PART
6

CHAPTER **1 Strengthening for Navigation in Ice**

SECTION **6 Baltic Ice Classes**

1 General

1.1 Application

Vessels to be distinguished in the *Record* by **Ice Class** followed by ice class **I AA** through **I C**, as specified in 6-1-6/3.1 are to meet the applicable requirements of this Section.

All vessels so designated are to be self-propelled and equipped with a radio telephone (VHF).

1.3 Area of Operation *(1 July 2019)*

The ice strengthening requirements in this Section are in agreement with the *Finnish-Swedish Ice Class Rules 2017*, developed for vessels sailing in the Baltic Sea area in winter or in other sea areas in similar ice conditions.

1.5 Additional Guidance *(1 July 2019)*

For additional guidance, see the latest revision of the *Guidelines for the Application of the Finnish-Swedish Ice Class Rules*.

3 Assignment of Ice Class

3.1 Ice Class *(2012)*

The requirements in this Section are intended primarily for vessels sailing in the Baltic Sea area in winter or in other sea areas in similar ice conditions and are assigned to ice classes as follows:

- **Ice Class I AA**; vessels with such structure, engine output and other properties that they are normally capable of navigating in difficult ice conditions without the assistance of ice breakers
- **Ice Class I A**; vessels with such structure, engine output and other properties that they are capable of navigating in difficult ice conditions, with the assistance of ice breakers when necessary
- **Ice Class I B**; vessels with such structure, engine output and other properties that they are capable of navigating in moderate ice conditions, with the assistance of ice breakers when necessary
- **Ice Class I C**; vessels with such structure, engine output and other properties that they are capable of navigating in light ice conditions, with the assistance of ice breakers when necessary

The administrations of Sweden and Finland (hereafter called the Administrations) provide ice breaker assistance to vessels bound for their ports in winter. Depending on the ice conditions, restrictions by the administrations may apply to the size and ice class of the vessel.

3.3 General Suitability for Operating in Ice

Where no specific requirements are given, vessels are assumed to be normal seagoing cargo vessels of conventional proportions, hull form and propulsion arrangement. A vessel having very unconventional proportions, hull form or propulsion arrangement, or any other characteristics, may have a lower ice class assigned by the Administrations.

3.5 General Suitability for Winter Conditions *(1 July 2019)*

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

The following temperatures are given for reference in the Baltic Sea area:

- Ambient temperature: –30°C
- Sea water temperature: –2°C

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

5 Definitions

5.1 Ice Belt

The *Ice Belt* is the area over which the shell plating is required to be reinforced for navigation in ice, see 6-1-6/13.1 and 6-1-6/Figure 4.

5.3 Upper and lower Ice Waterlines *(2012)*

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

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

5.5 Main Frame

Main Frames are real, or in the case of longitudinal framing, imaginary transverse frames, whose spacing corresponds to that of the vessel clear of the ice strengthening area, or of the vessel if it were not ice-strengthened.

5.7 Propulsion Machinery Output *(1 July 2019)*

The *Propulsion Machinery Output*, P, is the maximum output in kW that the machinery can continuously deliver to the propeller(s). If the output is restricted by technical means or by any regulations applicable to the vessel, P is to be taken as the restricted output. If additional power sources are available for propulsion power (e.g., shaft motors), in addition to the power of the main engine(s), they shall also be included in the total propulsion machinery output. The Propulsion Machinery Output used for the calculation of the hull scantlings shall be clearly stated on the Shell Expansion drawing.

7 Maximum and Minimum Draft Fore and Aft *(1 July 2019)*

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

Restrictions on drafts when operating in ice shall be documented and kept onboard readily available to the master. The maximum and minimum ice class drafts fore, amidships and aft are to be indicated in the classification certificate. For vessels built on or after 1 July 2007, if the summer load line in fresh water is anywhere located at a higher level than the UIWL, the vessel's sides are to be provided with a warning triangle and with an ice class draft mark at the maximum permissible ice class draft amidships (see Appendix 6-1-6A1).

Vessels built before 1 July 2007 are to be provided with such a marking, if the UIWL is below the summer load line, not later than the first scheduled dry docking after 1 July 2007. The draft and trim, limited by the UIWL, must not be exceeded when the vessel is navigating in ice. The salinity of the sea water along the intended route shall be taken into account when loading the vessel.

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

$$d_f = (2 + 0.00025\Delta)h_o \quad \text{m}$$

$$d_f = (2 + 0.000254\Delta)h_o \quad \text{ft}$$

but need not exceed $4h_o$

where

Δ = displacement of the vessel, in metric tons (long tons), at the upper ice waterline (UIWL) amidships, as defined in 6-1-6/5.3

h_o = level ice thickness, in m (ft), as defined in 6-1-6/11.5

9 Power of Propulsion Machinery *(1 September 2003)*

The minimum required engine output power P is to be determined in accordance with 6-1-6/9.1.2 and stated in the Classification certificate.

9.1 Propulsion Machinery Output, Ice Classes I AA, I A, I B and I C* *(1 September 2003)*

*(*NOTE: For reference purposes, the propulsion machinery output requirements for* ***I AA, I A, I B*** *and* ***I C*** *in the1985 Finnish-Swedish Ice Class Rules were amended as follows for vessels with the keel laid or which are at a similar stage of construction on or after 1 September 2003.)*

9.1.1 Definitions *(1 July 2019)*

The dimensions of the vessel are defined below in 6-1-6/Figure 1.

L = length of the vessel between perpendiculars at the UIWL, m (m, ft)

L_{BOW} = length of the bow, m (m, ft)

L_{PAR} = length of the parallel midship body, m (m, ft)

B = maximum breadth of the vessel at the UIWL, m (m, ft)

T = actual ice class drafts of the vessel in accordance with 6-1-6/9.1.2. Drafts to be used are the maximum draft amidships corresponding to UIWL and the minimum draft corresponding to LIWL, m (m, ft)

A_{Wf} = area of waterline of the bow, m^2 (m^2, ft^2)

H_F = thickness of the brash ice layer displaced by the bow, m (m, ft)

H_M = thickness of the brash ice in mid channel, m (m, ft)

α = the angle of the waterline at $B/4$, deg

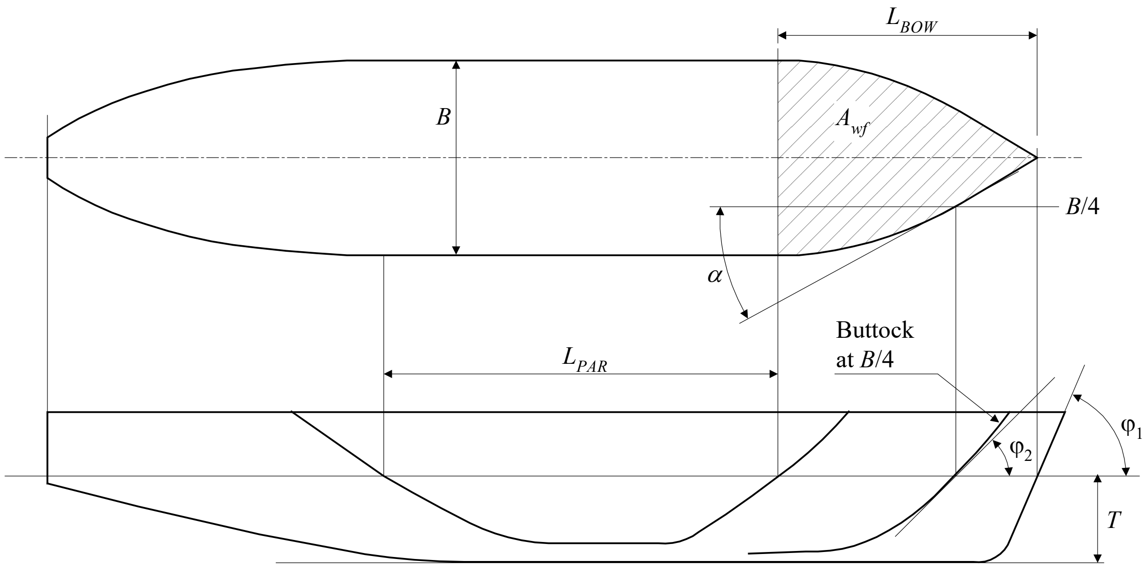
φ_1 = the rake of the stem at the centerline, deg

φ_2 = the rake at the bow, at $B/4$, deg

ψ = flare angle calculated as $\psi = \arctan(\tan\varphi/\sin\alpha)$ using angles α and φ at each location. For 6-1-6/9, flare angle is calculated using $\varphi = \varphi_2$

D_P = diameter of the propeller, m (m, ft)

FIGURE 1
Vessels' Dimensions



For a vessel with a bulbous bow, φ_1 is to be taken as 90°.

9.1.2 Power Calculation *(1 July 2019)*

To be entitled to ice class **I AA**, **I A**, **I B** or **I C**, a vessel the keel of which is laid or which is at a similar stage of construction on or after 1 September 2003 is to comply with the following requirements regarding its engine output. The engine output requirement shall be calculated for two drafts. Drafts to be used are the maximum draft amidships referred to as UIWL and the minimum draft referred to as LIWL, as defined in 6-1-6/7. In the calculations, the vessel's parameters which depend on the draft are to be determined at the appropriate draft, but L and B are to be determined only at the UIWL.

The engine output shall not be less than the greater of these two outputs. The engine output is to be not less than determined by the formula below and in no case less than 1000 kW (1360 mhp; 1341 hp) for Ice Class **I A, I B** and **I C**, and not less than 2800 kW (3807 mhp; 3754 hp) for Ice Class **I AA**.

$$P = K_C \frac{(R_{CH}/1000)^{3/2}}{D_p} \quad \text{kW (mhp, hp)}$$

where K_C is to be taken as follows:

Propeller Type or Propulsion Machinery	*Controllable Pitch Propeller or Electric or Hydraulic Propulsion Machinery*			*Fixed Pitch Propeller*		
	SI Units	*MKS Units*	*US Units*	*SI Units*	*MKS Units*	*US Units*
1 propeller	2.03	84.76	83.79	2.26	94.37	93.29
2 propellers	1.44	60.13	59.44	1.6	66.81	66.04
3 propellers	1.18	49.27	48.71	1.31	54.70	54.07

These K_C values apply for conventional propulsion systems. Other methods may be used for determining the required power for advanced propulsion systems (see 6-1-6/9.1.3).

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

$$R_{CH} = C_1 + C_2 + C_3\, C_\mu (H_F + H_M)^2 (B + C_\varnothing H_F) + C_4 L_{PAR} H_F^2 + C_5 \left[\frac{LT}{B^2}\right]^3 \frac{A_{Wf}}{L} \quad \text{N (kgf, lbf)}$$

where

C_μ	=	$0.15\cos\varphi_2 + \sin\psi\sin\alpha$,	C_μ is to be taken equal or larger than 0.45
$C_\varnothing$	=	$0.047\psi - 2.115$,	and $C_\varnothing = 0$ if $\psi \le 45°$
H_F	=	$0.26 + (H_M B)^{0.5}$ m	
H_F	=	$0.85 + (H_M B)^{0.5}$ ft	
H_M	=	1.0 m (3.28 ft)	for Ice Class **I A** and **I AA**
	=	0.8 m (2.62 ft)	for Ice Class **I B**
	=	0.6 m (1.97 ft)	for Ice Class **I C**

The coefficients C_1 and C_2 take into account a consolidated upper layer of the brash ice and can be taken as zero for Ice Class **I A, I B and I C**.

For Ice Class **I AA**:

$$C_1 = f_1 \frac{BL_{PAR}}{(2T/B)+1} + (1 + 0.021\varphi_1)(f_2 B + f_3 L_{BOW} + f_4 BL_{BOW}) \quad \text{N (kgf, lbf)}$$

$$C_2 = (1 + 0.063\varphi_1)(g_1 + g_2 B) + g_3(1 + 1.2T/B)\frac{B^2}{\sqrt{L}} \quad \text{N (kgf, lbf)}$$

	SI units	*MKS units*	*US units*
f_1	23 N/m^2	2.35 kgf/m^2	0.48 lbf/ft^2
f_2	45.8 N/m	4.67 kgf/m	3.138 lbf/ft
f_3	14.7 N/m	1.50 kgf/m	1.007 lbf/ft
f_4	29 N/m^2	2.96 kgf/m^2	0.61 lbf/ft^2
g_1	1530 N	156.02 kgf	343.96 lbf
g_2	170 N/m	17.34 kgf/m	11.649 lbf/ft
g_3	400 N/m$^{1.5}$	40.79 kgf/m$^{1.5}$	15.132 lbf/ft$^{1.5}$
C_3	845 N/m^3	86.2 kgf/m^3	5.38 lbf/ft^3
C_4	42 N/m^3	4.28 kgf/m^3	0.267 lbf/ft^3
C_5	825 N/m	84.1 kgf/m	56.5 lbf/ft

ψ = $\arctan[\tan\varphi_2/\sin\alpha]$ deg.

If the value of the term $(LT/B^2)^3$ is less than 5, the value 5 shall be used and if the value of the term is more than 20, the value 20 shall be used.

9.1.3 Other Methods of Determining K_C and R_{CH}

The Administration may for an individual vessel, in lieu of the K_C or R_{CH} values defined in 6-1-6/9.1 above, approve the use of K_C and R_{CH} values based on more exact calculations or values based on model test. Such an approval will be given on the understanding that it can be revoked if experience with the vessel's performance in practice motivates this.

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

I AA H_M = 1.0 m (3.28 ft) and a 0.1 m (0.328 ft) thick consolidated layer of ice

I A H_M = 1.0 m (3.28 ft)

I B H_M = 0.8 m (2.62 ft)

I C H_M = 0.6 m (1.97 ft)

11 Hull Structural Design *(2012)*

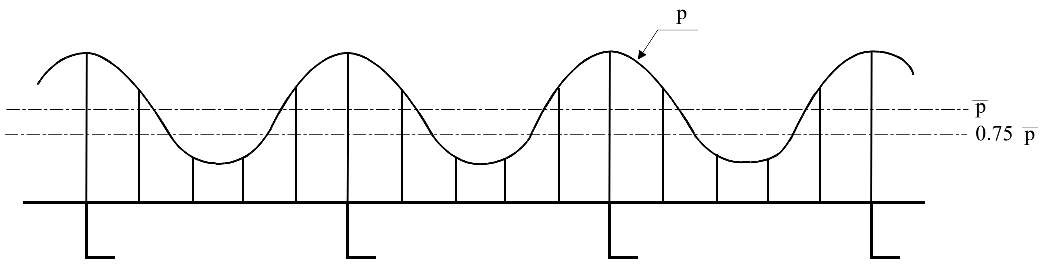
11.1 Application

The requirements for the hull scantlings are based on certain assumptions concerning the nature of the ice load on the structure. These assumptions are from full scale observations made in the Northern Baltic.

The local ice pressure on small areas can reach high values. This pressure may be well in excess of the normal uniaxial crushing strength of sea ice since the stress field is multi-axial.

It has also been observed that the ice pressure on a frame can be greater than on the shell plating at mid-spacing between frames. This is due to the different flexural stiffness of the frames and shell plating. The load distribution on the side structure is assumed to be as shown in 6-1-6/Figure 2.

FIGURE 2
Ice Load Distribution on Ship's Side



The formulae and values given in this section may be substituted by direct analysis if they are deemed by the Administration or ABS to be invalid or inapplicable for a given structural arrangement or detail. Otherwise, direct analysis is not to be utilized as an alternative to the analytical procedures prescribed by explicit requirements in 6-1-6/13 through 6-1-6/17.

Direct analyses are to be carried out using the load patch defined in 6-1-6/11.5 and 6-1-6/11.7 (p, h and ℓ_a). The pressure to be used is 1.8p where p is determined according to 6-1-6/11.7. The load patch is to be applied at locations where the capacity of the structure under the combined effects of bending and shear are minimized. In particular, the structure is to be checked with load centered at UIWL, 0.5h_o below the LIWL, and positioned several vertical locations in between. Several horizontal locations should also be checked, especially the locations centered at the mid-span – or spacing. Further, if the load length ℓ_a cannot be determined directly from the arrangement of the structure, several values of ℓ_a should be checked using corresponding values for c_a.

Acceptance criterion for designs is that the combined stresses from bending and shear, using the von Mises yield criterion, are lower than the yield point σ_y. When the direct calculation is using beam theory, the allowable shear stress is not to be larger than 0.9τ_y, where $\tau_y = \sigma_y / \sqrt{3}$.

Where the scantlings given by these requirements are less than those required by the Rules for a not ice-strengthened vessel, the greater requirements are to apply.

11.1.1

The frame spacings and spans defined in the following text are normally (in accordance with the Rules) assumed to be measured along the plate and perpendicular to the axis of the stiffener for plates, along the flange for members with a flange, and along the free edge for flat bar stiffeners. For curved members the span (or spacing) is defined as the chord length between span (or spacing) points. The span points are defined by the intersection between the flange or upper edge of the member and the supporting structural element (stringer, web frame, deck or bulkhead). 6-1-6/Figure 3 illustrates the determination of span and spacing for curved members.

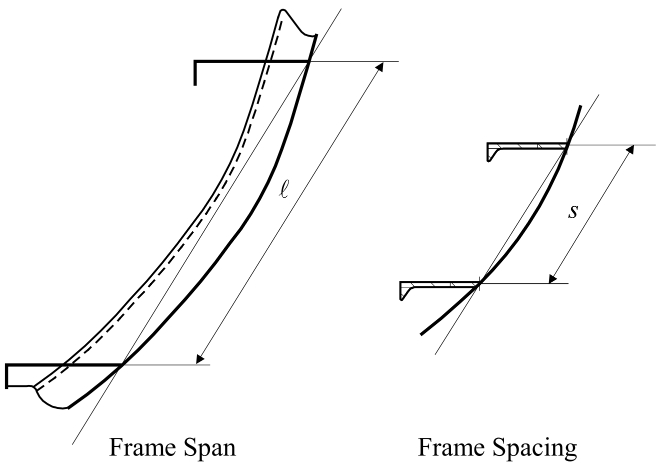
11.1.2

The effective breadth of the attached plate to be used for calculating the combined section modulus of the stiffener, stringer and web frame and attached plate is to be taken as the Rules require. The effective breadth is in no case to be more than what is stated in 3-1-2/13.3.

11.1.3

The requirements for the section modulus and shear area of the frames, stringers and web frames in 6-1-6/15, 6-1-6/17, and 6-1-6/19 are with respect to effective member cross section. For such cases where the member is not normal to the plating, the section properties are to be adjusted in accordance with the Rules.

FIGURE 3
Definition of the Frame Span and Frame Spacing for Curved Members *(2012)*



11.3 Hull Regions

For the application of this Section the vessel's ice belt is divided forward and aft into the following regions, see also 6-1-6/Figure 4.

11.3.1 Bow Region *(1 July 2019)*

From the stem to a line through the ice belt parallel to and 0.04L aft of the forward borderline of the part of the hull where the waterlines run parallel to the centerline. For ice classes **I AA** and **I A**, the overlap over the borderline need not exceed 6 m (19.7 ft); for ice classes **I B** and **I C**, this overlap need not exceed 5 m (16.4 ft).

11.3.2 Midbody Region *(1 July 2019)*

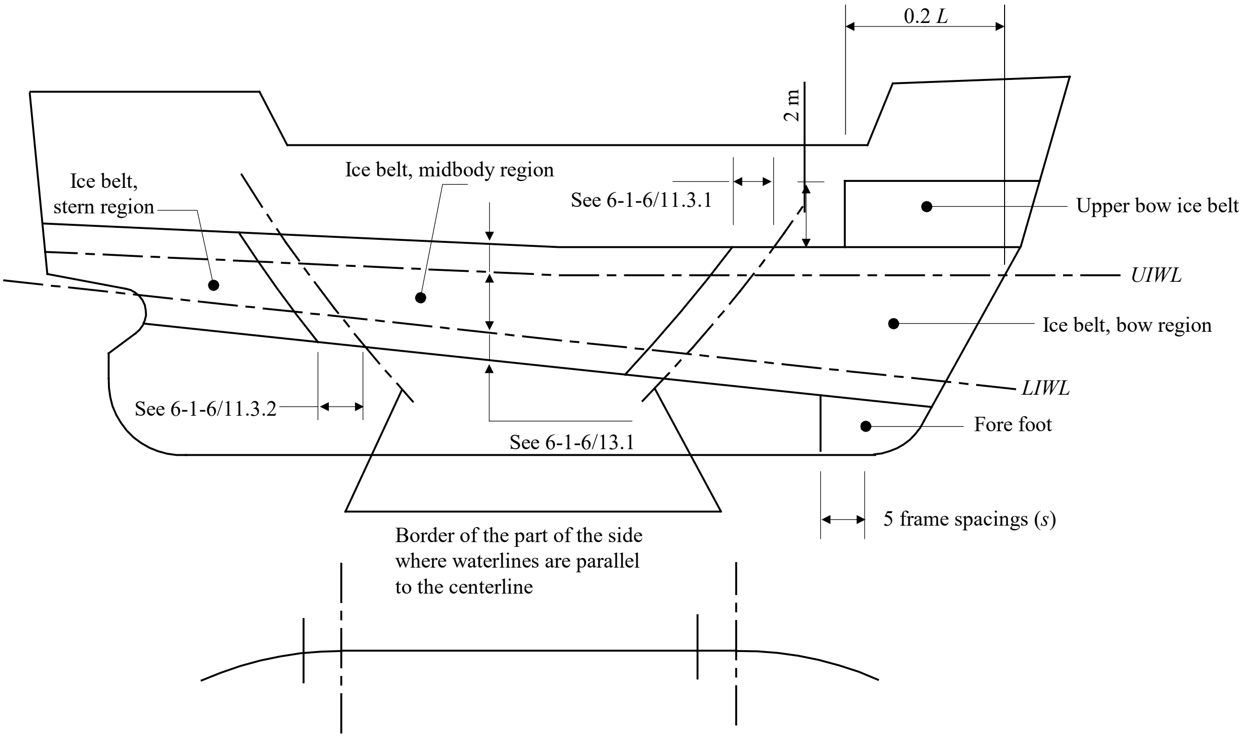
From the aft boundary of the Bow region to a line parallel to and 0.04L aft of the aft borderline of the part of the hull where the waterlines run parallel to the centerline. For ice classes **I AA** and **I A**, the overlap over the borderline need not exceed 6 m (19.7 ft); for ice classes **I B** and **I C**, this overlap need not exceed 5 m (16.4 ft).

11.3.3 Stern Region

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

L is to be taken as the vessel's rule length, as defined in 3-1-1/3.1.

FIGURE 4
Ice Strengthened Regions of the Hull *(2012)*



11.5 Vertical Extent of Design Ice Pressure

An ice strengthened vessel is assumed to operate in open sea conditions with level ice thickness not exceeding h_o. The design load height, h, of the area actually under ice pressure at any particular time is, however, assumed to be only a fraction of the ice thickness. The values for h_o and h are given in the following table:

Ice Class	*h_o m (ft)*	*h m (ft)*
I AA	1.0 (3.28)	0.35 (1.15)
I A	0.8 (2.62)	0.30 (0.98)
I B	0.6 (1.97)	0.25 (0.82)
I C	0.4 (1.31)	0.22 (0.72)

11.7 Design Ice Pressure

The design ice pressure is to be not less than given by the following equation:

$p = c_d \cdot c_1 \cdot c_a \cdot p_o$ N/mm^2 (kgf/mm^2, psi)

where

c_d = a factor which takes into account the influence of the size and propulsion machinery output of the vessel. This factor is taken as maximum $c_d = 1$.

= $(ak + b)/1000$

k = $\sqrt{n\Delta P}/1000$

a and b are given in the following table:

	Region			
	Bow		Midbody and Stern	
	$k \le 12$	$k > 12$	$k \le 12$	$k > 12$
a	30	6	8	2
b	230	518	214	286

n = 1.0 (1.0, 1.016)

Δ = displacement of the vessel, in metric tons (long tons), at the upper ice waterline (UIWL) amidships, as defined in 6-1-6/5.3

P = the actual continuous propulsion machinery output, in kW, as defined in 6-1-6/5.7

c_1 = factor which takes into account the probability that the design ice pressure occurs in a certain region of the hull for the particular ice class

The value of c_1 is given in the following table:

	Region		
Ice Class	Bow	Midbody	Stern
I AA	1.0	1.0	0.75
I A	1.0	0.85	0.65
I B	1.0	0.70	0.45
I C	1.0	0.50	0.25

c_a = a factor which takes into account the probability that the full length of the area under consideration will be under pressure at the same time

= $\sqrt{\dfrac{\ell_0}{\ell_a}}$, maximum 1.0, minimum 0.35

ℓ_0 = 0.6 m (2 ft)

ℓ_a is as given in the following table:

Structure	Type of framing	ℓ_a m (ft)
Shell	Transverse	Frame spacing
	Longitudinal	1.7 times spacing of frame
Frames	Transverse	Frame spacing
	Longitudinal	Span of frame
Ice stringer		Span of stringer
Web frame		2 times spacing of web frames

p_o = the nominal ice pressure; the value 5.6 N/mm^2 (0.571 kgf/mm^2, 812 psi) is to be used

13 Shell Plating

13.1 Vertical Extent of Ice Strengthening for Plating (Ice Belt) *(1 July 2019)*

The vertical extension of the ice belt is given in the following table (see 6-1-6/Figure 4):

<table>
<tr><th>Ice Class</th><th>Hull Region</th><th>Above UIWL
m (ft)</th><th>Below LIWL
m (ft)</th></tr>
<tr><td rowspan="3">I AA</td><td>Bow</td><td rowspan="3">0.60 (1.97)</td><td rowspan="2">1.20 (3.94 ft)</td></tr>
<tr><td>Midbody</td></tr>
<tr><td>Stern</td><td>1.0 (3.28 ft)</td></tr>
<tr><td rowspan="3">I A</td><td>Bow</td><td rowspan="3">0.50 (1.64)</td><td>0.90 (2.95)</td></tr>
<tr><td>Midbody</td><td rowspan="2">0.75 (2.46)</td></tr>
<tr><td>Stern</td></tr>
<tr><td rowspan="3">I B and I C</td><td>Bow</td><td rowspan="3">0.40 (1.31)</td><td>0.70 (2.30)</td></tr>
<tr><td>Midbody</td><td rowspan="2">0.6 (1.97)</td></tr>
<tr><td>Stern</td></tr>
</table>

In addition, the following areas are to be strengthened:

13.1.1 Fore Foot *(1 July 2019)*

For ice class **I AA**, the shell plating below the ice belt from the stem to a position five main frame spaces abaft the point where the bow profile departs from the keel line shall be ice-strengthened in the same way as the bow region.

13.1.2 Upper Bow Ice Belt *(1 July 2019)*

For ice class **I AA** and **I A**, on vessels with an open water service speed equal to or exceeding 18 knots, the shell plating from the upper limit of the ice belt to 2 m (6.56 ft) above it and from the stem to a position at least $0.2L$ abaft the forward perpendicular is to be at least the thickness required for the ice belt in the Midbody region. A similar strengthening of the bow region is also advisable for a ship with a lower service speed when, on the basis of the model tests, for example, it is evident that the ship will have a high bow wave.

Side lights, side scuttles etc., are not to be situated in the ice belt. If the weather deck in any part of the vessel is situated below the upper limit of the ice belt (e.g., in way of the well of a raised quarter decker), the bulwark is to be given at least the same strength as is required for the shell in the ice belt. The strength of the construction of the freeing ports is to meet the requirements for the bulwark.

13.3 Ice Belt Plating Thickness *(2010)*

With transverse framing, the thickness of the shell plating is to be not less than given by the following equation:

$$t = a\,s\sqrt{f_1 P_{PL} / \sigma_y} + t_c \quad \text{mm (in.)}$$

With longitudinal framing, the thickness of the shell plating is to be not less than given by the following equation:

$$t = a\,s\sqrt{p / f_2 \sigma_y} + t_c \quad \text{mm (in.)}$$

where

s = frame spacing, in m (ft)

P_{PL} = 0.75 p, in N/mm^2 (kgf/mm^2, psi)

p = as given in 6-1-6/11.7

f_1 = $1.3 - 4.2/[(h/s) + 1.8]^2$; maximum 1.0

f_2 = $0.6 + 0.4/(h/s)$; when $h/s \le 1$

= $1.4 - 0.4\,(h/s)$; when $1 \le h/s < 1.8$

h = as given in 6-1-6/11.5, in m (ft)

σ_y = yield strength of the material, in N/mm² (kgf/mm², psi)

a = 667 (8)

Use of steels with yield strengths greater than 390 N/mm² (40 kgf/mm², 56565 psi) are subject to special consideration.

t_c = increment for abrasion and corrosion, in mm (in.); normally, t_c is to be 2 mm (0.08 in.); however, if a special surface coating by experience is shown capable to withstand the abrasion of ice and is applied and maintained effective, lower values may be approved.

15 Framing *(2012)*

15.1 General

15.1.1 End Attachments

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

15.1.2 Frames *(1 July 2019)*

15.1.2(a) Welding. Frames are to be attached to the shell by double continuous welding. Scallops are to be avoided, except where frames cross shell plate butts.

15.1.2(b) Web Thickness (1 July 2019). The web thickness of the frames is to be at least the maximum of the following:

- $$\frac{h_w\sqrt{\sigma_y}}{C}$$

  where

  h_w = web height

  C = 805 for profiles

  = 282 for flat bars

- Half of the net thickness of the shell plating, $t - t_c$. For the purpose of calculating the web thickness of frames, the required thickness of the shell plating is to be calculated according to 6-1-6/13.3 using the yield strength σ_y of the frames

- 9 mm (0.35 in.)

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

15.1.2(c) Slanted frames. Frames that are not normal to the plating or the profile is unsymmetrical, and the span exceeds 4.0 m (13.1 ft) are to be supported against tripping by brackets, intercostals, stringers or similar at a distance preferably not exceeding 1.3 m (4.25 ft). If the span is less than 4.0 m (13.1 ft), the supports against tripping are required for unsymmetrical profiles and stiffeners the web of which is not normal to plating in the following regions:

- **I AA** All hull regions
- **I A** Bow and Midbody regions
- **I B** and **I C** Bow region

15.3 Vertical Extent of Ice Strengthening for Framing

The vertical extent of the ice strengthening of framing is to be at least as given in the following table:

Ice Class	*Hull Region*	*Above UIWL*	*Below LIWL*
I AA	Bow	1.2 (3.94)	Down to double bottom or below top of floors
	Midbody		2.0 (6.56)
	Stern		1.6 (5.25)
I A, I B, I C	Bow	1.0 (3.28)	1.6 (5.25)
	Midbody		1.3 (4.27)
	Stern		1.0 (3.28)

Where an upper Bow ice belt is required, see 6-1-6/13.1, the ice strengthening of the framing is to be extended at least to the top of this ice belt.

Where the ice strengthening would go beyond a deck or a tanktop by not more than 250 mm (9.8 in.), it may be terminated at that deck or tanktop.

15.5 Transverse Framing

15.5.1 Section Modulus and Shear Area

The section modulus, *SM,* of a main or intermediate frame is to be not less than that obtained from the equation:

$$SM = n\left(\frac{p \cdot h \cdot s \cdot \ell}{m_t \cdot \sigma_y}\right) \quad \text{cm}^3\ (\text{in}^3)$$

and the effective shear area is calculated from

$$A = k\left(\frac{\sqrt{3} \cdot f_3 \cdot p \cdot h \cdot s}{2\sigma_y}\right) \quad \text{cm}^2\ (\text{in}^2)$$

where

n = 10^6 (1728)

k = 10^4 (144)

p = ice pressure, as given in 6-1-6/11.7, in N/mm^2 (kgf/mm^2, psi)

s = frame spacing, in m (ft)

h = height of load area, as given in 6-1-6/11.5, in m (ft)

ℓ = span of the frame, in m (ft)

m_t = $7m_o/[7 - 5\,(h/\ell)]$

f_3 = is a factor which takes into account the maximum shear force versus the load location and the shear stress distribution

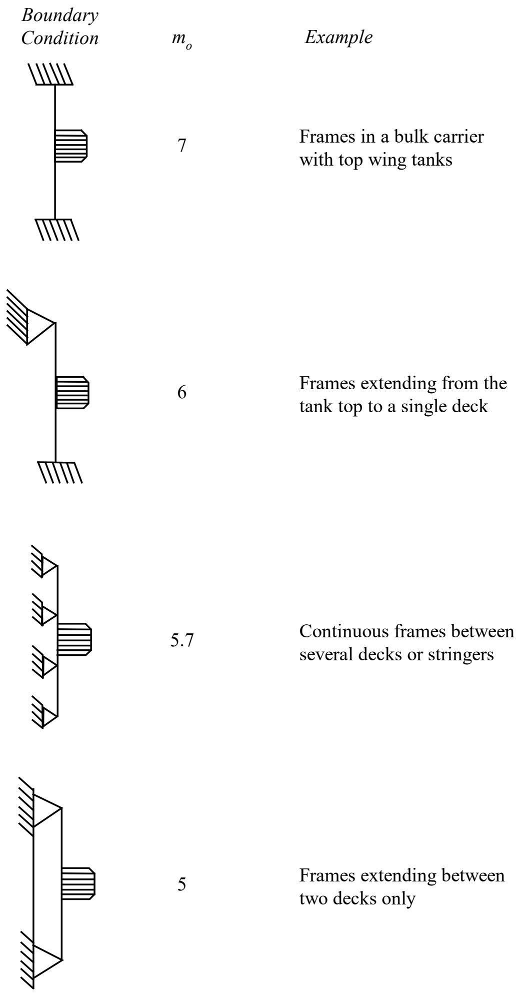
= 1.2

σ_y = yield strength, as defined in 6-1-6/13.3, in N/mm^2 (kgf/mm^2, psi)

m_o values are given in 6-1-6/Figure 5.

The boundary conditions shown are for the main and intermediate frames. Possible different conditions for the main frames are assumed to have been taken care of by interaction between the frames and are reflected in the m_o values. The load is considered applied at mid span. Where less than 15% of the span, ℓ, of the frame is situated within the ice-strengthening zone for frames as defined in 6-1-6/15.3, ordinary frame scantlings may be used.

FIGURE 5
Web Frame Model



15.5.2 Upper End of Transverse Frames

The upper end of an ice-strengthened part of a main frame and of an intermediate ice frame is to be attached to a deck or ice stringer, see 6-1-6/17.

Where an intermediate ice frame terminates above a deck or ice stringer that is situated at or above the upper limit of the ice belt, see 6-1-6/13.1, the part above the deck or stringer may have scantlings as required for a non-ice-strengthened vessel and the upper end of the intermediate frame may be connected to the adjacent main frames by a header of the same scantlings as the main frame.

15.5.3 Lower End of Transverse Framing

The lower end of an ice-strengthened part of a main frame and of an intermediate ice frame is to be attached to a deck, tanktop or ice stringer, see 6-1-6/17.

Where an intermediate ice frame terminates below a deck, tanktop or ice stringer which is situated at or below the lower limit of the ice belt, see 6-1-6/13.1, the lower end of the frame may be connected to the adjacent main frames by a header of the same scantlings as the main frame. Note that the main frames below the lower edge of the ice belt must be ice strengthened, see 6-1-6/15.3.

15.7 Longitudinal Framing

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

15.7.1 Frames with and without Brackets

The section modulus, *SM,* of a longitudinal frame is to be not less than that obtained from the equation:

$$SM = n(f_4\, p\, h\, \ell^2/m_1\, \sigma_y) \quad \text{cm}^3\ (\text{in}^3)$$

The effective shear area, *A*, is to be not less than that obtained from the equation:

$$A = k\,(\sqrt{3}\, f_4 f_5\, p\, h\, \ell/\sigma_y) \quad \text{cm}^2\ (\text{in}^2)$$

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

f_4	=	factor which takes into account the load distribution to adjacent frames
	=	$(1 - 0.2h/s)$
f_5	=	factor which takes into account the pressure definition and maximum shear force versus load location and also the shear stress distribution
	=	2.16
p	=	ice pressure, as given in 6-1-6/11.7, in N/mm^2 (kgf/mm^2, psi)
h	=	height of load area, as given in 6-1-6/11.5, in m (ft)
s	=	frame spacing, in m (ft)
n	=	10^6 (1728)
k	=	5×10^3 (72)
ℓ	=	total span of frame, in m (ft)
m_1	=	boundary condition factor; $m_1 = 13.3$ for a continuous beam. Where the boundary conditions deviate significantly from those of a continuous beam (e.g., in an end field), a smaller boundary factor may be required. For frames without brackets a value $m_1 = 11.0$ is to be used.
σ_y	=	yield strength, as defined in 6-1-6/13.3, in N/mm^2 (kgf/mm^2, psi)

17 Ice Stringers *(2012)*

17.1 Stringers within the Ice Belt

The section modulus, *SM*, of a stringer within the ice belt (see 6-1-6/13.1) is to be not less than that obtained from the equation:

$$SM = n\left(\frac{f_6 \cdot f_7 \cdot p \cdot h \cdot \ell^2}{m \cdot \sigma_y}\right) \quad \text{cm}^3\ (\text{in}^3)$$

The effective shear area, *A,* is to be not less than that obtained from the equation:

$$A = k\left(\frac{\sqrt{3} \cdot f_6 \cdot f_7 \cdot f_8 \cdot p \cdot h \cdot \ell}{2\sigma_y}\right) \quad \text{cm}^2\ (\text{in}^2)$$

where

p = ice pressure, as given in 6-1-6/11.7, in N/mm^2 (kgf/mm^2, psi)

h = height of load area, as given in 6-1-6/11.5, in m (ft)

The product ($p \times h$) is not to be taken as less than 0.15 SI units (0.0153 MKS units, 71.4 US units)

n = 10^6 (1728)

k = 10^4 (144)

ℓ = span of stringer, in m (ft)

m_s = boundary condition factor; as defined in 6-1-6/15.7

f_6 = factor which takes into account the distribution of the load on the transverse frames

= 0.9

f_7 = factor that takes into account the design point of stringers

= 1.8

f_8 = factor that takes into account the maximum shear force versus load location and the shear stress distribution

= 1.2

σ_y = yield strength, as defined in 6-1-6/13.3, in N/mm^2 (kgf/mm^2, psi)

17.3 Stringers Outside the Ice Belt

The section modulus, *SM,* of a stringer outside the ice belt that supports ice strengthened frames is to be not less than that obtained from the equation:

$$SM = n\left(\frac{f_9 \cdot f_{10} \cdot p \cdot h \cdot \ell^2}{m_s \cdot \sigma_y}\right)\left(1 - \frac{h_s}{\ell_s}\right) \quad \text{cm}^3\ (\text{in}^3)$$

The effective shear area, *A,* is to be not less than that obtained from the equation:

$$A = k\left(\frac{\sqrt{3} f_9 \cdot f_{10} \cdot f_{11} \cdot p \cdot h \cdot \ell}{2\sigma_y}\right)\left(1 - \frac{h_s}{\ell_s}\right) \quad \text{cm}^2\ (\text{in}^2)$$

where

p = ice pressure, as given in 6-1-6/11.7, in N/mm^2 (kgf/mm^2, psi)

h = height of load area, as given in 6-1-6/11.5, in m (ft)

The product ($p \times h$) is to be not taken as less than 0.15 SI units (0.0153 MKS units, 71.4 US units).

n = 10^6 (1728)

k = 10^4 (144)

ℓ = span of stringer, in m (ft)

m_s = boundary condition factor; m_s = 13.3 for a continuous beam

ℓ_s = the distance to the adjacent ice stringer, in m (ft)

h_s = the distance to the ice belt, in m (ft)

f_9 = factor which takes into account the distribution of load on transverse frames.

= 0.80

f_{10} = factor that takes into account the design point of stringers

= 1.8

f_{11} = factor that takes into account the maximum shear force versus load location and the shear stress distribution

= 1.2

σ_y = yield strength, as defined in 6-1-6/13.3, in N/mm^2 (kgf/mm^2, psi)

17.5 Deck Strips

The deck strips abreast of hatches serving as ice stringers are to comply with the section modulus and shear area requirements in 6-1-6/17.1 and 6-1-6/17.3, respectively. In the case of very long hatches, the product ($p \times h$) may be taken as less than 0.15 SI units (0.0153 MKS units, 71.4 US units), but in no case less than 0.10 SI units (0.0102 MKS units, 47.6 US units).

In designing weather deck hatch covers and their fittings, special attention is to be paid to the deflection of the vessel's sides due to ice pressure in way of very long (more than $B/2$) hatch openings.

19 Web Frames *(2012)*

19.1 Design Ice Load

The design load, F, on a web frame from an ice stringer or from longitudinal framing may be obtained from the following equation:

$$F = n f_{12} phS \quad \text{kN (tf, Ltf)}$$

where

n = 10^3 (0.0643)

f_{12} = a factor that takes into account the design point of web frames

= 1.8

p = ice pressure, as given in 6-1-6/11.7, in N/mm^2 (kgf/mm^2, psi); in calculating c_a however, ℓ_a is to be taken as $2S$

h = height of ice load area, as given in 6-1-6/11.5, in m (ft)

The product ($p \times h$) is not to be taken as less than 0.15 SI units (0.0153 MKS units, 71.4 US units).

S = distance between web frames, in m (ft)

In case the supported stringer is outside the ice belt, the force F shall be multiplied by $(1 - h_s/\ell_s)$, where h_s and ℓ_s shall be taken as defined in 6-1-6/17.3.

19.3 Section Modulus and Shear Area *(1 July 2013)*

The section modulus and shear area may be obtained from the following equations:

- *Effective Shear Area*

$$A = k\left(\frac{\sqrt{3} \cdot f_{13} \cdot \alpha \cdot Q}{\sigma_y}\right) \quad \text{cm}^2 \text{ (in}^2\text{)}$$

where

Q = maximum calculated shear force under the load F, as given in 6-1-6/19.1

k = 10 (2240)

f_{13} = factor that takes into account the shear force distribution

= 1.1

α = as given in the Table below

σ_y = yield strength, as defined in 6-1-6/13.3, in N/mm² (kgf/mm², psi)

F = as in 6-1-6/19.1

- *Section Modulus*

$$SM = n\left(\frac{M}{\sigma_y}\right)\sqrt{\frac{1}{1-(\gamma A/A_a)^2}} \quad \text{cm}^3 \text{ (in}^3\text{)}$$

where

n = 1000 (26880)

M = maximum calculated bending moment under the load F; this is to be taken as

= $0.193F\ell$

γ = as given in the Table below

A = required shear area

A_a = actual cross sectional area of the web frame, in cm² (in²)

= $A_f + A_w$

- *Factors α and γ*

A_f = actual cross section area of free flange, in cm² (in²)

A_w = actual effective cross section area of web plate, in cm² (in²)

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

21 Bow *(2012)*

21.1 Stem

The stem may be made of rolled, cast or forged steel or of shaped steel plates as shown in 6-1-6/Figure 6.

The thickness of a shaped plate stem and, in the case of a blunt bow, any part of the shell where $\alpha \geq 30°$ and $\psi \geq 75°$ (see 6-1-6/9.1.1 for angle definitions), is to be obtained from the equation in 6-1-6/13.3 where:

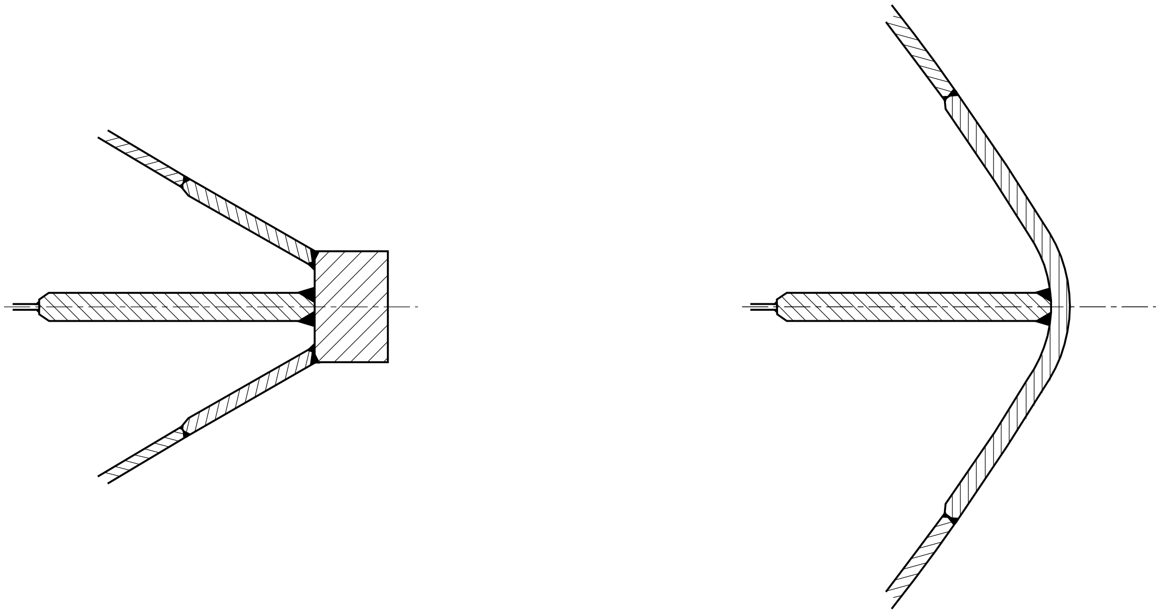
s = spacing of elements supporting the plate, in m (ft)

P_{PL} = p, in N/mm² (kgf/mm², psi), see 6-1-6/11.7

ℓ_a = spacing of vertical supporting elements, in m (ft)

The stem and that part of a blunt bow defined above is to be supported by floors, breasthooks or brackets spaced not more than 0.6 m (1.97 ft) apart and of a thickness at least half the shell plate thickness. This reinforcement of the stem is to extend from the keel to a point 0.75 m (2.46 ft) above *UIWL*, or where an upper Bow ice belt is required, see 6-1-6/13.1, to the upper limit of this upper Bow ice belt.

FIGURE 6
Examples of Suitable Ice Stems *(2012)*



23 Stern *(1 July 2019)*

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

In order to avoid very high loads on propeller blade tips, the minimum distance between propeller(s) and hull (including stern frame) should not be less than h_0 (see 6-1-6/11.5).

On twin and triple screw vessels, the ice strengthening of the shell and framing is to extend to the double bottom for 1.5 meters (4.92 ft) forward and aft of the side propellers.

Shafting and stern tubes of side propellers are to be normally enclosed within plated bossing. If detached struts are used, their design, strength and attachment to the hull is to be duly considered for ice loading.

25 Rudder and Steering Arrangements *(2012)*

25.1 Minimum Design Speed

The scantlings of rudder post, rudder stock, pintles, steering gear etc., as well as the capacity of the steering gear are to comply with Section 3-2-14 of the Rules. Where the design ahead speed of the vessel, as defined in 3-2-14/3.1, is less than the minimum speed indicated in the table below, the latter speed is to be used in lieu of V in Section 3-2-14.

Class	*Minimum Speed*
I AA	20 knots
I A	18 knots
I B	16 knots
I C	14 knots

For use with the minimum ahead speeds in the above table, k_c may be taken as 80% of that specified in Section 3-2-14. Also, k_1 for rudders situated behind nozzles need not be taken as greater than 1.0.

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

25.3 Double Plated Rudders

For double plated rudders, the minimum thickness of plates and horizontal and vertical webs in the ice-belt region is to be determined as for shell plating in the Stern region in accordance with 6-1-6/13.

25.5 Rudder and Rudder Stock Protection

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

25.7 Overload Design *(1 July 2019)*

For ice classes **I AA** and **I A**, due regard is to be given to the excessive loads caused by the rudder being forced out of the midship position when going astern in ice or backing into an ice ridge. Suitable arrangements such as rudder stops are to be installed to absorb these loads.

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

27 Propulsion Machinery *(2010)*

27.1 Scope *(1 July 2019)*

Requirements 6-1-6/27 apply to propulsion machinery covering open- and ducted-type propellers with controllable pitch or fixed pitch design for the ice classes **I AA**, **I A**, **I B** and **I C**. The given propeller loads are the expected ice loads for the whole ship's service life under normal operational conditions, including loads resulting from the changing rotational direction of FP propellers. However, these loads do not cover off-design operational conditions, for example when a stopped propeller is dragged through ice. The requirements also apply to azimuthing and fixed thrusters for main propulsion, considering loads resulting from propeller-ice interaction and loads on the thruster body-ice interaction. However, the load models do not include propeller/ice interaction loads when ice enters the propeller of a turned azimuthing thruster from the side (radially).

The given azimuthing thruster body loads are the expected ice loads for the ship's service life under normal operational conditions. The local strength of the thruster body shall be sufficient to withstand local ice pressure when the thruster body is designed for extreme loads.

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

27.3 Symbols *(1 July 2019)*

c = chord length of blade section, m (ft)

$c_{0.7}$ = chord length of blade section at 0.7R propeller radius, m (ft)

CP = controllable pitch

D = propeller diameter, m (ft)

d = external diameter of propeller hub (at propeller plane), m (ft)

D_{limit} = limit value for propeller diameter, m (ft)

EAR = expanded blade area ratio

F_b = maximum backward blade force for the ship's service life, kN (kgf, lbf)

F_{ex} = ultimate blade load resulting from blade loss through plastic bending, kN (kgf, lbf)

F_f = maximum forward blade force for the ship's service life, kN (kgf, lbf)

F_{ice} = ice load, kN (kgf, lbf)

$(F_{ice})_{max}$ = maximum ice load for the ship's service life, kN (kgf, lbf)

FP = fixed pitch

h_0 = depth of the propeller centerline from the lower ice waterline, m (ft)

H_{ice} = thickness of maximum design ice block entering to propeller, m (ft)

I_e = equivalent mass moment of inertia of all parts on engine side of component under consideration, kg-m^2 (lb-ft^2)

I_t = equivalent mass moment of inertia of the whole propulsion system, kg-m^2 (lb-ft^2)

k = shape parameter for Weibull distribution

$LIWL$ = lower ice waterline, m (ft)

m = slope for SN curve in log/log scale

M_{BL} = blade bending moment, kN-m (kgf-m, lbf-ft)

MCR = maximum continuous rating

n = propeller rotational speed, rev/s

n_n = nominal propeller rotational speed at MCR in free running condition, rev/s

N_{class} = reference number of impacts per propeller rotational speed per ice class

N_{ice} = total number of ice loads on propeller blade for the ship's service life

N_R = reference number of load for equivalent fatigue stress (10^8 cycles)

N_Q = number of propeller revolutions during a milling sequence

$P_{0.7}$ = propeller pitch at 0.7R radius, m (ft)

$P_{0.7n}$ = propeller pitch at 0.7R radius at MCR in free running condition, m (ft)

$P_{0.7b}$ = propeller pitch at 0.7R radius at MCR in bollard condition, m (ft)

Q = torque, kN-m (kgf-m, lbf-ft)

Q_{emax} = maximum engine torque, kN-m (kgf-m, lbf-ft)

Q_{max} = maximum torque on the propeller resulting from propeller-ice interaction, kN-m (kgf-m, lbf-ft)

Q^n_{max} = maximum torque on the propeller resulting from propeller-ice interaction reduced to the rotational speed in question, kN-m (kgf-m, lbf-ft)

Q_{motor} = electric motor peak torque, kN-m (kgf-m, lbf-ft)

Q_n = nominal torque at MCR in free running condition, kN-m (kgf-m, lbf-ft)

Q_r = maximum response torque along the propeller shaft line, kN-m (kgf-m, lbf-ft)

Q_{peak} = maximum of response torque Q_r, kN-m (kgf-m, lbf-ft)

Q_{smax} = maximum spindle torque of the blade for the ship's service life, kN-m (kgf-m, lbf-ft)

Q_{sex} = maximum spindle torque due to blade failure caused by plastic bending, kN-m (kgf-m, lbf-ft)

Q_{vib} = vibratory torque at considered component, taken from frequency domain open water torque vibration calculation (TVC), kN-m (kgf-m, lbf-ft)

R = propeller radius, m (ft)

r = blade section radius, m (ft)

T = propeller thrust, kN (kgf, lbf)

T_b = maximum backward propeller ice thrust for the ship's service life, kN (kgf, lbf)

T_f = maximum forward propeller ice thrust for the ship's service life, kN (kgf, lbf)

T_n = propeller thrust at MCR in free running condition, kN (kgf, lbf)

T_r = maximum response thrust along the shaft line, kN (kgf, lbf)

t = maximum blade section thickness, m (ft)

Z = number of propeller blades

α_i = duration of propeller blade/ice interaction expressed in rotation angle, deg

α_1 = phase angle of propeller ice torque for blade order excitation component, deg

α_2 = phase angle of propeller ice torque for twice the blade order excitation component, deg

$\gamma_{\varepsilon 1}$ = reduction factor for fatigue; scatter effect

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

γ_v = reduction factor for fatigue; variable amplitude loading effect

γ_m = reduction factor for fatigue; mean stress effect

ρ = reduction factor for fatigue correlating the maximum stress amplitude to the equivalent fatigue stress for 10^8 stress cycles

$\sigma_{0.2}$ = proof yield strength (at 0.2% offset) of blade material, MPa (kgf/cm^2, psi)

σ_{exp} = mean fatigue strength of blade material at 10^8 cycles to failure in sea water, MPa (kgf/cm^2, psi)

σ_{fat} = equivalent fatigue ice load stress amplitude for 10^8 stress cycles, MPa (kgf/cm^2, psi)

σ_{fl} = characteristic fatigue strength for blade material, MPa (kgf/cm^2, psi)

σ_{ref} = reference stress $\sigma_{ref} = 0.6\sigma_{0.2} + 0.4\sigma_u$, MPa (kgf/cm^2, psi)

σ_{ref2} = reference stress $\sigma_{ref2} = 0.7\sigma_u$ or $\sigma_{ref2} = 0.6\sigma_{0.2} + 0.4\sigma_u$, whichever is less, MPa (kgf/cm^2, psi)

σ_{st} = maximum stress resulting from F_b or F_f, MPa (kgf/cm^2, psi)

σ_u = ultimate tensile strength of blade material, MPa (kgf/cm^2, psi)

$(\sigma_{ice})_{b\max}$ = principal stress caused by the maximum backward propeller ice load, MPa (kgf/cm^2, psi)

$(\sigma_{ice})_{f\max}$ = principal stress caused by the maximum forward propeller ice load, MPa (kgf/cm^2, psi)

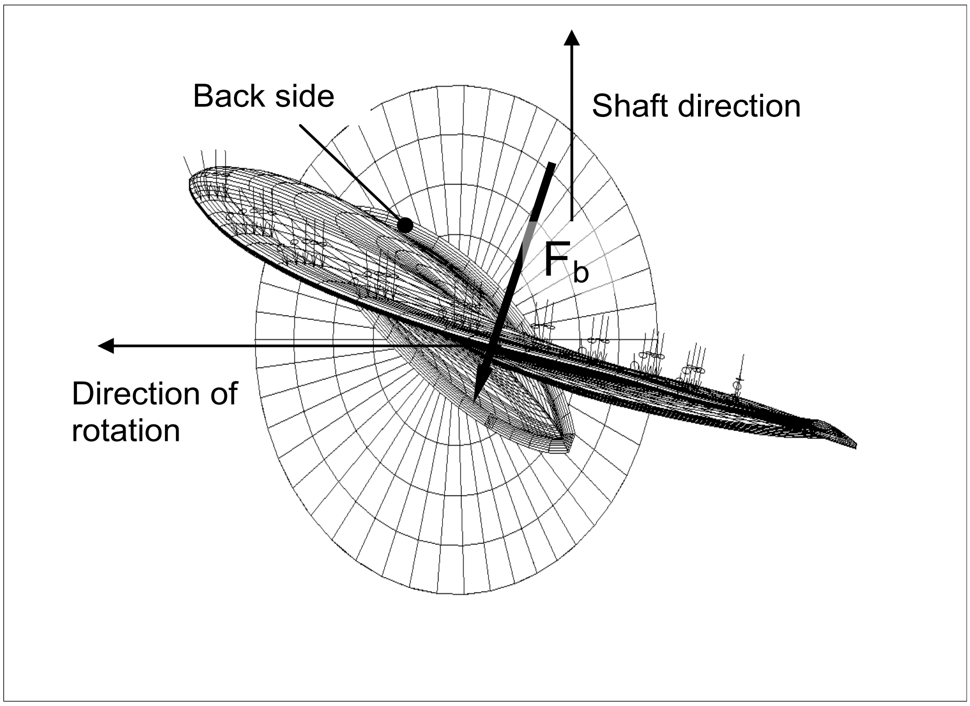
$(\sigma_{ice})_{\max}$ = maximum ice load stress amplitude, MPa (kgf/cm^2, psi)

TABLE 1
Definition of Loads *(1 July 2019)*

	Definition	*Use of the load in design process*
F_b	The maximum lifetime backward force on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to 0.7 *r*/*R* chord line. See 6-1-6/Figure 7	Design force for strength calculation of the propeller blade.
F_f	The maximum lifetime forward force on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to 0.7 *r*/*R* chord line.	Design force for calculation of strength of the propeller blade.
Q_{smax}	The maximum lifetime spindle torque on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade.	In designing the propeller strength, the spindle torque is automatically taken into account because the propeller load is acting on the blade as distributed pressure on the leading edge or tip area.
T_b	The maximum lifetime thrust on a propeller (all blades) resulting from propeller/ice interaction. The direction of the thrust is the propeller shaft direction and the force is opposite to the hydrodynamic thrust.	Is used for 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 a propeller (all blades) resulting from propeller/ice interaction. The direction of the thrust is the propeller shaft direction acting in the direction of hydrodynamic thrust.	Is used for estimation of the response thrust T_r. T_f can be used as an estimate of excitation for axial vibration calculations. However, axial vibration calculations are not required in the Rules.
Q_{max}	The maximum ice-induced torque resulting from propeller/ice interaction on one propeller blade, including hydrodynamic loads on that blade.	Is used for estimation of the response torque (Q_r) along the propulsion shaft line and as excitation for torsional vibration calculations.
F_{ex}	Ultimate blade load resulting from blade loss through plastic bending. The force that is needed to cause total failure of the blade so that plastic hinge appears in the root area. The force is acting on 0.8 *r*/*R*. Spindle arm is to be taken as $^2/_3$ of the distance between the axis of blade rotation and leading/trailing edge (whichever is the greater) at the 0.8*R* radius.	Blade failure load is used to dimension the blade bolts, pitch control mechanism, propeller shaft, propeller shaft bearing and trust bearing. The objective is to guarantee that total propeller blade failure does not lead to damage to other components.
Q_r	Maximum response torque along the propeller shaft line, taking into account the dynamic behavior of the shaft line for ice excitation (torsional vibration) and the hydrodynamic mean torque on the propeller.	Design torque for propeller shaft line components.
T_r	Maximum response thrust along shaft line, taking into account the dynamic behavior of the shaft line for ice excitation (axial vibration) and the hydrodynamic mean thrust on the propeller.	Design thrust for propeller shaft line components.
F_{ti}	Maximum response force caused by ice block impacts on the thruster body or the propeller hub.	Design load for thruster body and slewing bearings.
F_{tr}	Maximum response force on the thruster body caused by ice ridge-thruster body interaction.	Design load for thruster body and slewing bearings.

FIGURE 7
Direction of the Backward Blade Force Resultant Taken Perpendicular to Chord Line at Radius 0.7*R* *(1 July 2019)*

Ice contact pressure at leading edge is shown with small arrows.



27.5 Design Ice Conditions

In estimating the ice loads of the propeller for ice classes, different types of operation as given in 6-1-6/Table 2 were taken into account. For the estimation of design ice loads, a maximum ice block size is determined. The maximum design ice block entering the propeller is a rectangular ice block with the dimensions $H_{ice} \times 2H_{ice} \times 3H_{ice}$. The thickness of the ice block (H_{ice}) is given in 6-1-6/Table 3.

TABLE 2
Types of Ice Operation *(2010)*

Ice Class	*Operation of the Ship*
I AA	Operation in ice channels and in level ice The ship may proceed by ramming
I A, I B, I C	Operation in ice channels

TABLE 3
Thickness of the Ice Block (H_{ice}) *(2010)*

	I AA	**I A**	**I B**	**I C**
Thickness of the design maximum ice block entering the propeller (H_{ice})	1.75 m (5.74 ft)	1.5 m (4.92 ft)	1.2 m (3.94 ft)	1.0 m (3.28 ft)

27.7 Materials

27.7.1 Materials Exposed to Sea Water *(1 July 2019)*

Materials of components exposed to sea water, such as propeller blades, propeller hubs, and thruster body, are to have an elongation of not less than 15% on a test specimen, the gauge length of which is five times the diameter. A Charpy V impact test is to be carried out for materials other than bronze and austenitic steel. An average impact energy value of 20 J (2.04 kgf-m, 14.75 lbf-ft) taken from three tests is to be obtained at minus 10°C (14°F). For nodular cast iron, average impact energy of 10 J at minus 10°C (14°F) is required accordingly.

27.7.2 Materials Exposed to Sea Water Temperature *(1 July 2019)*

Materials exposed to sea water temperature are to be of ductile material. An average impact energy value of 20 J (2.04 kgf-m, 14.75 lbf-ft) taken from three tests is to be obtained at minus 10°C (14°F). This requirement applies to the propeller shaft, blade bolts, CP mechanisms, shaft bolts, strut-pod connecting bolts etc. This does not apply to stoppers and surface hardened components, such as bearings and gear teeth. The nodular cast iron of a ferrite structure type may be used for relevant parts other than bolts. The average impact energy for nodular cast iron shall be a minimum of 10 J at minus 10°C (14°F).

27.9 Design Loads *(1 July 2019)*

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

The values of the parameters in the formulae in this section are to be given in the units shown in the symbol list in 6-1-6/27.3.

If the highest point of the propeller is not at a depth of at least h_o below the water surface when the ship is in ballast condition, the propulsion system shall be designed according to ice class **I A** for ice classes **I B** and **I C**.

27.9.1 Design Loads on Propeller Blades

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. These forces originate from different propeller/ice interaction phenomena, not acting simultaneously. Hence, they are to be applied to one blade separately.

27.9.1(a) Maximum backward blade force, F_b, for open propellers (2012):

$$F_b = k \cdot [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot D^2 \quad \text{kN (kgf, lbf)} \qquad \text{when } D \le D_{limit}$$

k = 27 (2753.23, 245.48)

$$F_b = k \cdot [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot D \cdot H_{ice}^{1.4} \quad \text{kN (kgf, lbf)} \qquad \text{when } D > D_{limit}$$

k = 23 (2345.35, 130.01)

where

D_{limit} = $0.85 \cdot H_{ice}^{1.4}$ m

D_{limit} = $0.622 \times 0.85 \cdot H_{ice}^{1.4}$ (ft)

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

27.9.1(b) Maximum forward blade force, F_f, for open propellers (2012):

$$F_f = k \cdot \left[\frac{EAR}{Z}\right] \cdot D^2 \quad \text{kN (kgf, lbf)} \qquad \text{when } D \le D_{limit}$$

k = 250 (25492.9, 5221.36)

$$F_f = k \cdot \left[\frac{EAR}{Z}\right] \cdot D \cdot \frac{1}{\left(1 - \frac{d}{D}\right)} \cdot H_{ice} \quad \text{kN (kgf, lbf)} \qquad \text{when } D > D_{limit}$$

k = 500 (50985.81, 10442.72)

where

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

27.9.1(c) Loaded area on the blade for open propellers. Load cases 1-4 are to be covered, as given in 6-1-6/Table 4 below, for CP and FP propellers. In order to obtain blade ice loads for a reversing propeller, load case 5 also is to be covered for FP propellers.

27.9.1(d) Maximum backward blade ice force, F_b, for ducted propellers:

$$F_b = k \cdot [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot D^2 \quad \text{kN (kgf, lbf)} \qquad \text{when } D \le D_{limit}$$

k = 9.5 (968.73, 86.37)

$$F_b = k \cdot [n \cdot D]^{0.7} \cdot \left[\frac{EAR}{Z}\right]^{0.3} \cdot D^{0.6} \cdot H_{ice}^{1.4} \quad \text{kN (kgf, lbf)} \qquad \text{when } D > D_{limit}$$

k = 66 (6730.13, 600.06)

where

D_{limit} = $4 \cdot H_{ice}$ m (ft)

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

27.9.1(e) Maximum forward blade ice force, F_f, for ducted propellers:

$$F_f = k \cdot \left[\frac{EAR}{Z}\right] \cdot D^2 \quad \text{kN (kgf, lbf)} \qquad \text{when } D \le D_{limit}$$

k = 250 (25492.91, 5221.35)

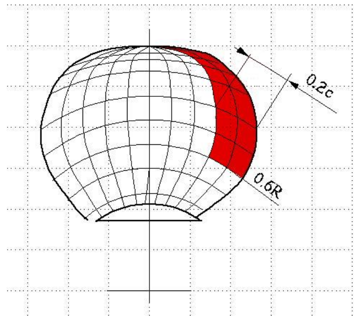
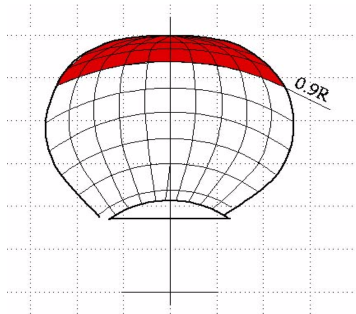
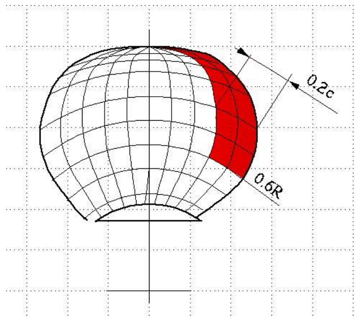
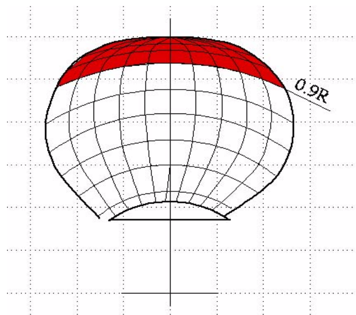
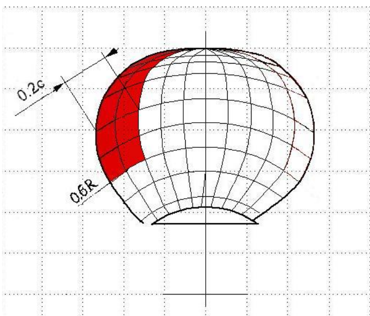
$$F_f = k \cdot \left[\frac{EAR}{Z}\right] \cdot D \cdot \frac{1}{\left(1 - \frac{d}{D}\right)} \cdot H_{ice} \quad \text{kN (kgf, lbf)} \qquad \text{when } D > D_{limit}$$

k = 500 (50985.91, 10442.72)

where

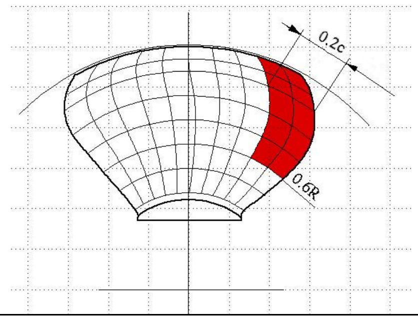
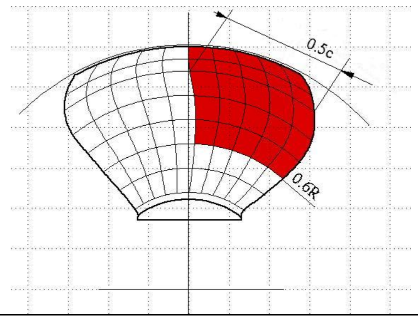
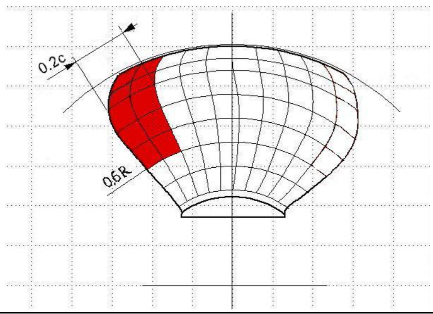
$$D_{limit} = \frac{2}{\left(1 - \frac{d}{D}\right)} \cdot H_{ice} \quad \text{m (ft)}$$

TABLE 4
Load Cases for Open Propellers *(2010)*

	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 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	

27.9.1(f) Loaded area on the blade for ducted propellers. Load cases 1 and 3 are to be covered as given in 6-1-6/Table 5 for all propellers, and an additional load case (load case 5) for an FP propeller, to cover ice loads when the propeller is reversed.

TABLE 5
Load Cases for Ducted Propellers *(2010)*

	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 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.	

27.9.1(g) Maximum blade spindle torque, Q_{smax}, for open and ducted propellers (1 July 2019). The spindle torque, Q_{smax}, around the axis of the blade fitting is to be determined both for the maximum backward blade force, F_b, and forward blade force, F_f, which are applied as in 6-1-6/Table 4 and 6-1-6/Table 5. The larger of the obtained torques is used as the dimensioning torque. If the above method gives a value which is less than the default value given by the formula below, the default value is to be used.

$$Q_{smax} = 0.25 \cdot F \cdot c_{0.7} \quad \text{kN-m (kgf-m, lbf-ft)}$$

where

$c_{0.7}$ = length of the blade section at 0.7*R* radius

F = either F_b or F_f, whichever has the greater absolute value

27.9.1(h) Load distributions for blade loads (1 July 2019). The Weibull-type distribution (probability that F_{ice} exceeds $(F_{ice})_{max}$), as given in 6-1-6/Figure 8, is used for the fatigue design of the blade.

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

where

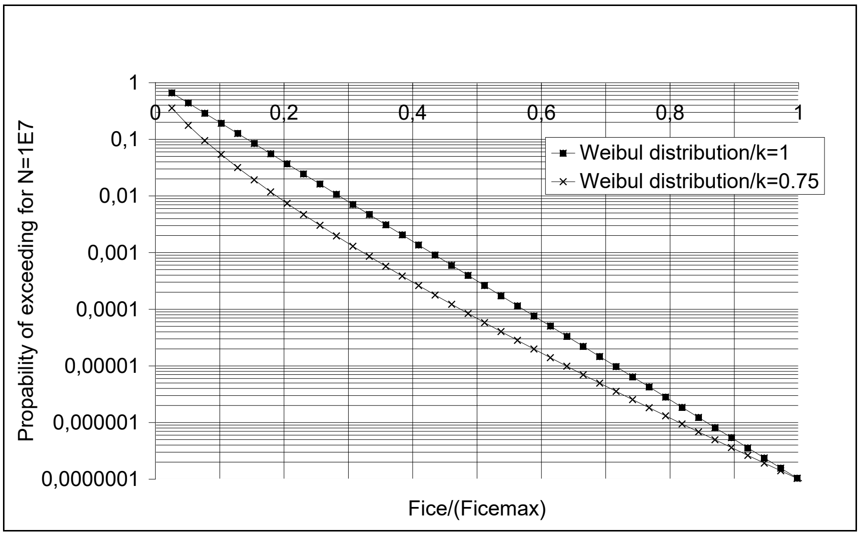
k = shape parameter of the spectrum

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

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

The shape parameter k = 0.75 is to 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 8
The Weibull-type Distribution (Probability that F_{ice} Exceeds $(F_{ice})_{max}$) That is Used for Fatigue Design *(1 July 2019)*



27.9.1(i) Number of ice loads (1 July 2019). The number of load cycles per propeller blade in the load spectrum is to be determined according to the formula:

$$N_{ice} = k_1 k_2 k_3 N_{class} n_n$$

where

Reference number of loads for ice classes N_{class}:

Class	**I AA**	**I A**	**I B**	**I C**
Impacts in life/n_n	9×10^6	6×10^6	3.4×10^6	2.1×10^6

Propeller location factor k_1:

Centerline Propeller Bow First Operation	*Wing Propeller Bow First Operation*	*Pulling Propeller (Wing and Centerline) Bow Propeller or Stern First operation*
1	2	3

The submersion factor, k_2, is determined from the equation:

$$K_2 = 0.8 - f \qquad \text{when } f < 0$$
$$= 0.8 - 0.4f \qquad \text{when } 0 \le f \le 1$$
$$= 0.6 - 0.2f \qquad \text{when } 1 \le f \le 2.5$$
$$= 0.1 \qquad \text{when } f > 2.5$$

Where the immersion function f is:

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

where h_o is the depth of the propeller centerline at the lower ice waterline (LIWL) of the vessel.

Propulsion type factor k_3:

Type	*Fixed*	*Azimuthing*
k_3	1	1.2

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).

27.9.2 Axial Design Loads for Open and Ducted Propellers

27.9.2(a) Maximum ice thrust on propeller (1 July 2019). The maximum forward and backward ice thrusts are:

$$T_f = 1.1 \cdot F_f \quad \text{kN (kgf, lbf)}$$

$$T_b = 1.1 \cdot F_b \quad \text{kN (kgf, lbf)}$$

27.9.2(b) 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 direction loads is to be taken as the design load for both directions. The factors 2.2 and 1.5 take into account the dynamic magnification resulting from axial vibration.

In a forward direction:

$$T_r = T + 2.2 \cdot T_f \quad \text{kN (kgf, lbf)}$$

In a backward direction:

$$T_r = 1.5 \cdot T_b \quad \text{kN (kgf, lbf)}$$

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

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

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

27.9.3 Torsional Design Loads

27.9.3(a) Design ice torque on propeller Q_{max} for open propellers (1 July 2019). Q_{max} is the maximum torque on a propeller resulting from ice/propeller interaction during the service life of the vessel.

$$Q_{max} = k \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot (nD)^{0.17} \cdot D^3 \quad \text{kN-m (kgf-m, lbf-ft)}$$

when $D \le D_{limit}$

$k = 10.9\ (1111.49,\ 186.02)$

$$Q_{max} = k \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot (nD)^{0.17} \cdot D^{1.9} \cdot H_{ice}^{1.1} \quad \text{kN-m (kgf-m, lbf-ft)}$$

when $D > D_{limit}$

$k = 20.7\ (2110.81,\ 353.26)$

where

D_{limit} = $1.8 \cdot H_{ice}$ m (ft)

n is the rotational propeller speed at MCR in bollard condition. If unknown, n is to be attributed a value in accordance with the following table.

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	$0.85n_n$

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

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

27.9.3(b) Design ice torque on propeller Q_{max} for ducted propellers (1 July 2019). Q_{max} is the maximum torque on a propeller during the service life of the ship resulting from ice/propeller interaction.

$$Q_{max} = k \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot (nD)^{0.17} \cdot D^3 \quad \text{kN-m (kgf-m, lbf-ft)}$$

when $D \le D_{limit}$

$k = 7.7\ (785.18,\ 131.41)$

$$Q_{max} = k \cdot \left[1 - \frac{d}{D}\right] \cdot \left[\frac{P_{0.7}}{D}\right]^{0.16} \cdot (nD)^{0.17} \cdot D^{1.9} \cdot H_{ice}^{1.1} \quad \text{kN-m (kgf-m, lbf-ft)}$$

when $D > D_{limit}$

$k = 14.6\ (1488.78,\ 249.16)$

where

D_{limit} = $1.8 \cdot H_{ice}$ m (ft)

n = rotational propeller speed at MCR in bollard condition. If not known, n is to have a value according to the table in 6-1-6/27.9.3(a)

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 $0.7P_{0.7n}$, where $P_{0.7n}$ is the propeller pitch at MCR in free running condition.

27.9.3(c) Design torque for non-resonant shaft lines (1 July 2019). If there is no relevant first blade order torsional resonance in the operational speed range or in the range 20% above and 20% below the maximum operating speed (bollard condition), the following estimation of the maximum torque can be used.

Directly coupled two stroke diesel engines without flexible coupling

$$Q_{peak} = Q_{e\max} + Q_{vib} + Q_{\max} \cdot \frac{I_e}{I_t} \quad \text{kN-m (kgf-m, lbf-ft)}$$

and other plants

$$Q_{peak} = Q_{e\max} + Q_{\max} \cdot \frac{I_e}{I_t} \quad \text{kN-m (kgf-m, lbf-ft)}$$

where

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

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, $Q_{e\max}$ is unknown, it is to be taken as follows:

TABLE 6
Default Values for Prime Mover Maximum Torque $Q_{e\max}$ *(1 July 2019)*

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	$0.75Q_n$

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

27.9.3(d) Design torque for shaft lines having resonances (1 July 2019). If there is a first blade order torsional resonance in the operational speed range or in the range 20% above and 20% below the maximum operating speed (bollard condition), the design torque (Q_{peak}) of the shaft component is to be determined by means of torsional vibration analysis of the propulsion line.

There are two alternative ways of performing the dynamic analysis.

i) Time domain calculation for estimated milling sequence excitation

ii) Frequency domain calculation for blade orders sinusoidal excitation

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

27.9.3(e) Time domain calculation of torsional response (1 July 2019). Time domain calculations are to be performed for the MCR condition, MCR bollard conditions and for blade order resonant rotational speeds so that the resonant vibration responses can be obtained.

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

The following load cases are intended to reflect the operational loads on the propulsion system, when the propeller interacts with ice, and the respective reaction of the complete system. The ice impact and system response causes loads in the individual shaft line components. The ice torque Q_{max} may be taken as a constant value in the complete speed range. When considerations at specific shaft speeds are performed, a relevant Q_{max} may be calculated using the relevant speed according to section 6-1-6/27.9.3(a) or 6-1-6/27.9.3(b).

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

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

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

$$Q(\varphi) = C_q \cdot Q_{max} \cdot \sin[\varphi(180/\alpha_i)] \qquad \text{when } \varphi = 0 \ldots \alpha_i$$

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

where

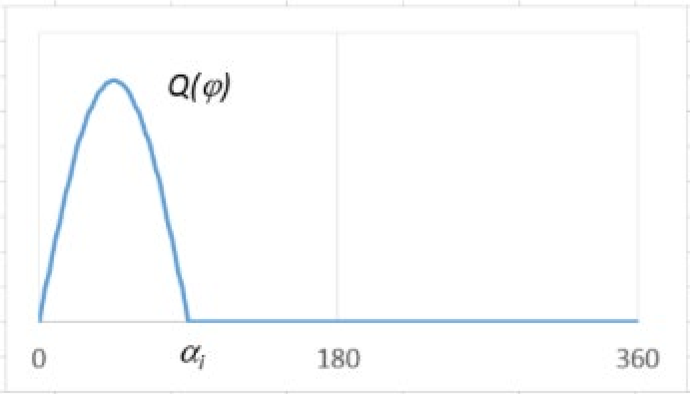
φ = the rotation angle from when the first impact occurs

C_q and α_i are given in the table below.

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

α_i is the duration of propeller blade/ice interaction expressed in terms of the propeller rotation angle (see 6-1-6/Figure 9).

FIGURE 9
Schematic Ice Torque due to a Single Blade Ice Impact as a Function of the Propeller Rotation Angle *(1 July 2019)*



The total ice torque is obtained by summing the torque of single blades, while taking account of the phase shift 360 deg/Z, see 6-1-6/Figure 10 or 6-1-6/Figure 11. At the beginning and end of the milling sequence (within the calculated duration) linear ramp functions shall be used to increase C_q to its maximum value within one propeller revolution and vice versa to decrease it to zero (see the examples of different Z numbers in 6-1-6/Figure 10 or 6-1-6/Figure 11).

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

$N_Q = 2 \times H_{ice}$ where H_{ice} in m

$N_Q = 0.3048 \times 2 \cdot H_{ice}$ where H_{ice} in ft

The number of impacts is $Z \cdot N_Q$ for blade order excitation. An illustration of all excitation cases for different numbers of blades is given in Figure 10 or Figure 11.

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

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

FIGURE 10
The Shape of the Propeller Ice Torque Excitation for Propellers with 3 or 4 Blades *(1 July 2019)*

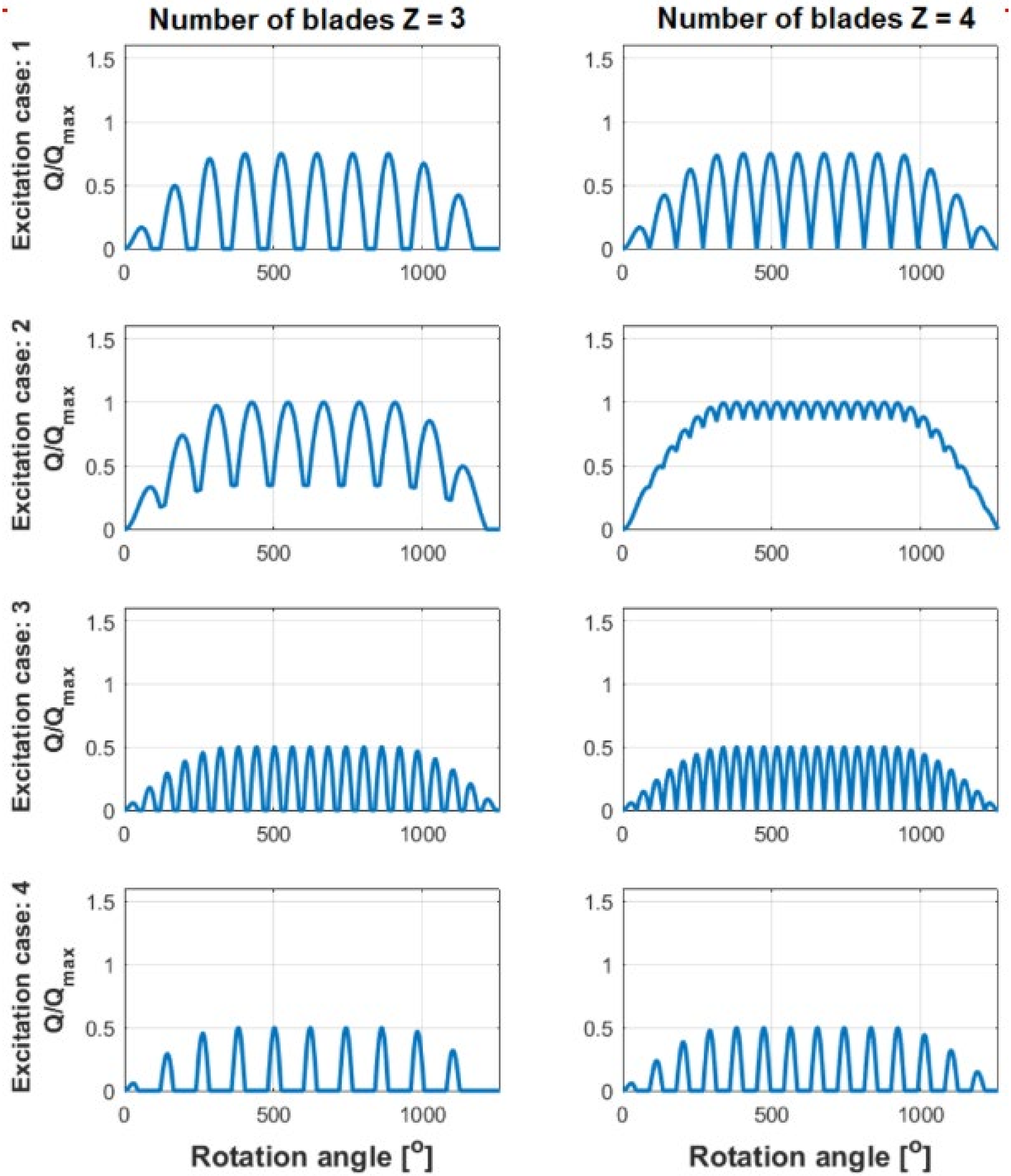
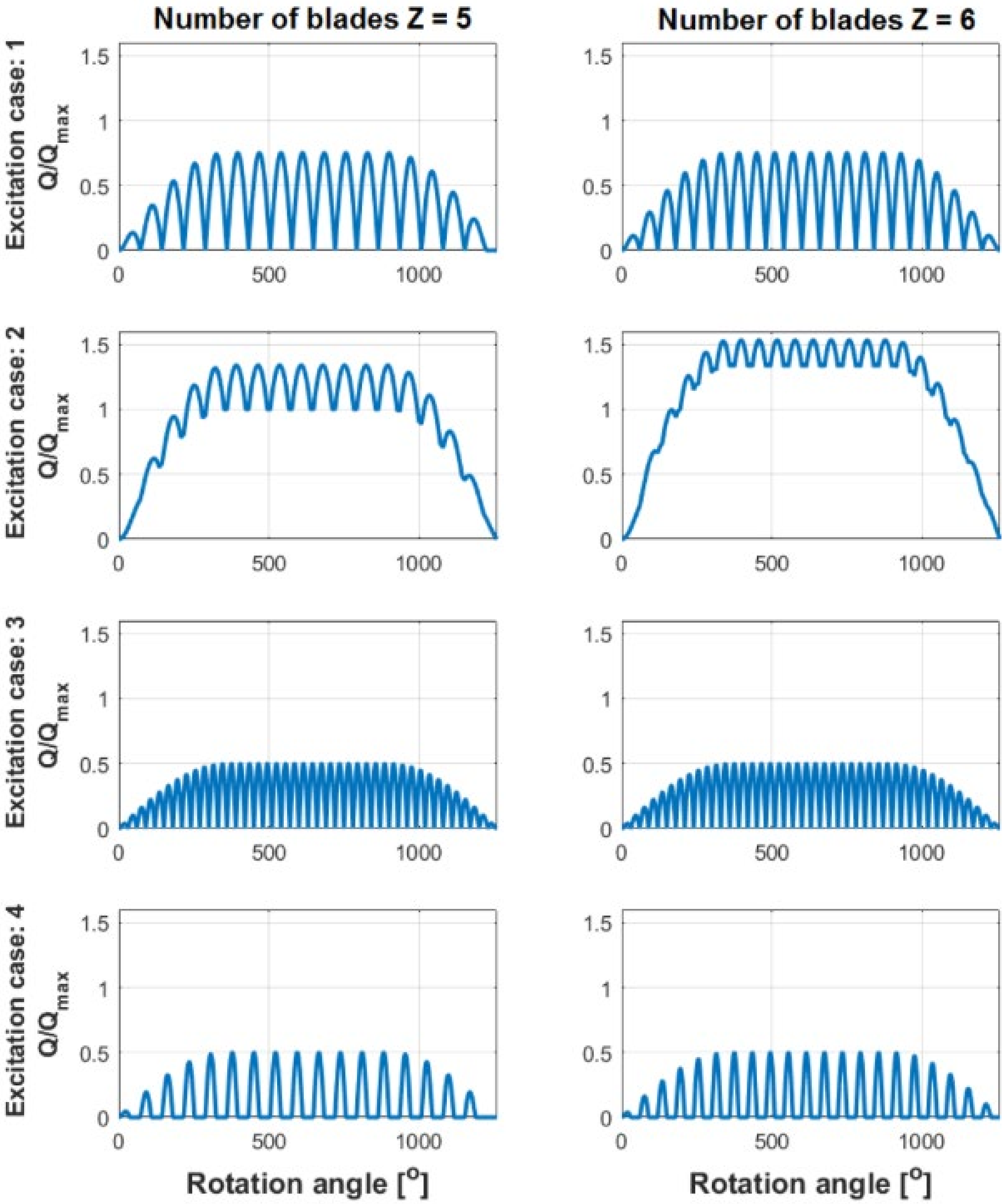


FIGURE 11
The Shape of the Propeller Ice Torque Excitation for Propellers with 5 or 6 Blades *(1 July 2019)*



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

$$Q_{peak} = Q_{emax} + Q_{rtd} \quad \text{kN-m (kgf-m, lbf-ft)}$$

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

27.9.3(f) Frequency domain calculation of torsional response (1 July 2019). For frequency domain calculations, blade order and twice-the-blade-order excitation may be used. The amplitudes for the blade order and twice-the-blade-order sinusoidal excitation have been derived based on the assumption that the time domain half sine impact sequences were continuous, and the Fourier series components for blade order and twice-the-blade-order components have been derived. The propeller ice torque is then:

$$Q_F(\varphi) = Q_{max}\,[C_{q0} + C_{q1}\sin(ZE_0\varphi + \alpha_1) + C_{q0} + C_{q2}\sin(2ZE_0\varphi + \alpha_2) \quad \text{kN-m (kgf-m, lbf-ft)}$$

where

C_{q0} = mean torque parameter

C_{q1} = first blade order excitation parameter

C_{q2} = second blade order excitation parameter

α_1, α_2 = phase angles of the excitation component

φ = angle of rotation

E_0 = number of ice blocks in contact

The values of the parameters are given in the following table:

TABLE 7
Coefficient Values for Frequency Domain Excitation Calculation *(1 July 2019)*

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

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

$$Q_{peak} = Q_{emax} + Q_{vib} + \left(Q^n_{\max} C_{q0}\right)\frac{I_e}{I_t} + Q_{rf1} + Q_{rf2} \quad \text{kN-m (kgf-m, lbf-ft)}$$

where

$Q^n_{\max}$ = maximum propeller ice torque at the operation speed in consideration

C_{q0} = mean static torque parameter from 6-1-6/Table 7

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

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

If the prime mover maximum torque, Q_{emax}, is not known, it is to be taken as given in 6-1-6/Table 6. All the torque values are to be scaled to the shaft revolutions for the component in question.

27.9.3(g) Guidance for torsional vibration calculation (1 July 2019). The aim of time domain torsional vibration simulations is to estimate the extreme torsional load for the ship's lifespan. The simulation model can be taken from the normal lumped mass elastic torsional vibration model, including damping. For a time domain analysis, the model should include the ice excitation at the propeller, other relevant excitations and the mean torques provided by the prime mover and hydrodynamic mean torque in the propeller. The calculations should cover variation of phase between the ice excitation and prime mover excitation. This is most relevant to propulsion lines with directly driven combustion engines. Time domain calculations shall be calculated for the MCR condition, MCR bollard conditions and for resonant speed, so that the resonant vibration responses can be obtained.

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

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

27.9.4 Blade Failure Load *(1 July 2019)*

27.9.4(a) Bending force, F_{ex}. The ultimate load resulting from blade failure as a result of plastic bending around the blade root is to be calculated with the formula below, or alternatively by means of an appropriate stress analysis, reflecting the non-linear plastic material behavior of the actual blade. In such a case, the blade failure area may be outside the root section. The ultimate load is acting on the blade at the 0.8R radius in the weakest direction of the blade.

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

$$F_{ex} = \frac{k \cdot c \cdot t^2 \cdot \sigma_{ref1}}{0.8 \cdot D - 2 \cdot r} \quad \text{kN (kgf, lbf)}$$

k = 300 (300000, 43.20)

where

σ_{ref1} = $0.6 \cdot \sigma_{0.2} + 0.4 \cdot \sigma_u$ MPa (kgf/cm^2, psi)

σ_u = minimum ultimate tensile strength to be specified on the drawing

$\sigma_{0.2}$ = minimum yield or 0.2% proof strength to be specified on the drawing

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 typically located at the point where the fillet terminates at the blade profile.

27.9.4(b) 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 center towards the leading and trailing edges. At a certain distance from the blade center of rotation, the maximum spindle torque will occur. This maximum spindle torque shall be defined by an appropriate stress analysis or using the equation given below.

$$Q_{sex} = \max\,[C_{LE0.8};\ 0.8C_{TE0.8}]C_{spex}F_{ex} \quad \text{kN-m (kgf-m, lbf-ft)}$$

where

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

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

C_{sp} = non-dimensional parameter taking account of the spindle arm

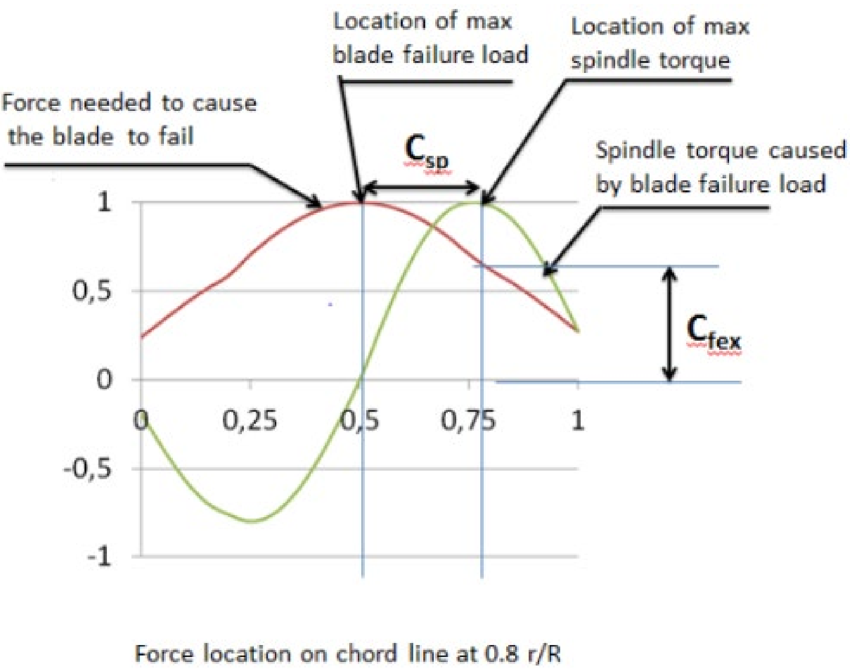
C_{fex} = non-dimensional parameter taking account of the reduction of the blade failure force at the location of the maximum spindle torque

$C_{LE0.8}$ = leading edge portion of the chord length at 0.8R

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

6-1-6/Figure 12 illustrates the spindle torque values due to blade failure loads across the entire chord length.

FIGURE 12
Blade Failure Load and the Related Spindle Torque when the Force Acts at a Different Location on the Chord Line at Radius 0.8*R* *(1 July 2019)*



27.11 Design

27.11.1 Design Principle

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

27.11.2 Propeller Blade

27.11.2(a) Calculation of blade stresses (2012). The blade stresses are to be calculated for the design loads given in 6-1-6/27.9.1. Finite element analyses are to be used for stress analysis for final approval for all propellers.

The following simplified formulae can be used in estimating the blade stresses for all propellers at the root area ($r/R < 0.5$). The root area dimensions will be accepted even if the FEM analysis would show greater stresses at the root area.

$$\sigma_{st} = C_1 \frac{M_{BL}}{k \cdot ct^2} \quad \text{MPa (kgf/cm}^2\text{, psi)}$$

$$k = 10^2 \ (10^3, 14.4)$$

where constant C_1 is the "actual stress"/"stress obtained with beam equation". If the actual value is not available, C_1 should be taken as 1.6.

$$M_{BL} = (0.75 - r/R) \cdot R \cdot F \qquad \text{for relative radius } r/R < 0.5$$

F is the maximum of F_b and F_f, whichever is greater.

27.11.2(b) Acceptability criterion (1 July 2019). The following criterion for calculated blade stresses is to be fulfilled.

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

where

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

σ_{ref2} = reference stress, defined as:

= $0.7 \cdot \sigma_u$ or

= $0.6 \cdot \sigma_{0.2} + 0.4 \cdot \sigma_u$, whichever is less

27.11.2(c) Fatigue design of propeller blade (Note - SI units) (1 July 2019). The fatigue design of the propeller blade is based on an estimated load distribution for the service life of the ship and the S-N curve for the blade material. An equivalent stress that produces the same fatigue damage as the expected load distribution shall be calculated and the acceptability criterion for fatigue should be fulfilled as given in this section. The equivalent stress is normalized for 10^8 (100 million) cycles.

For materials with a two-slope SN curve (6-1-6/Figure 13), if the following criterion is fulfilled, fatigue calculations according to this section are not required.

$$\sigma_{exp} \geq B1 \cdot \sigma_{ref2}^{B2} \cdot \log(N_{ice})^{B3}$$

where $B1$, $B2$, and $B3$ are coefficients for open and nozzle propellers are given in the table below.

	Open Propeller	*Nozzle Propeller*
*B*1	0.00246	0.00167
*B*2	0.947	0.956
*B*3	2.101	2.470

For calculation of equivalent stress, two types of S-N curves are available:

1. Two slope S-N curve (slopes 4.5 and 10), see 6-1-6/Figure 13.
2. One slope S-N curve (the slope can be chosen), see 6-1-6/Figure 14.

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

FIGURE 13
Two-slope S-N Curve *(2010)*

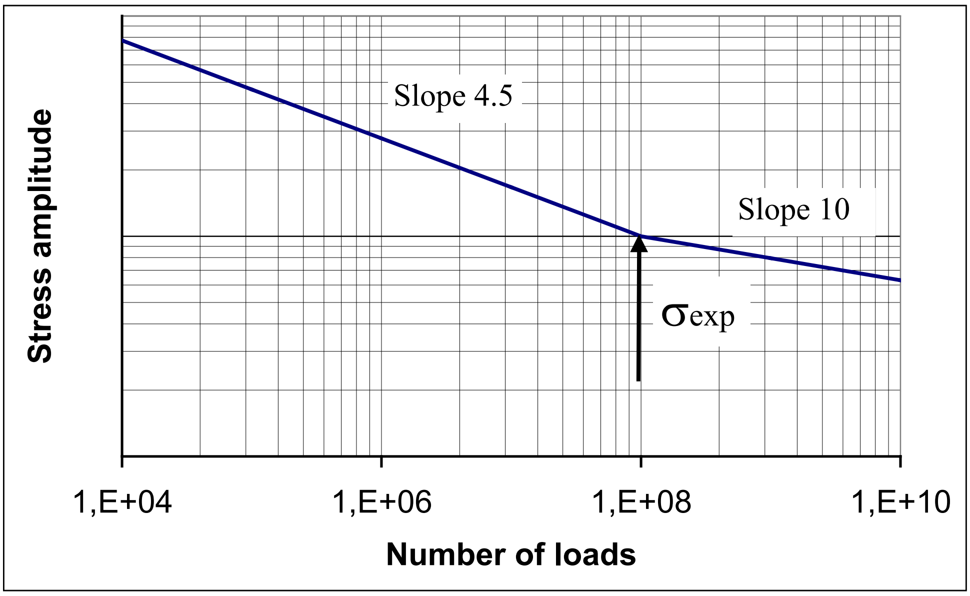
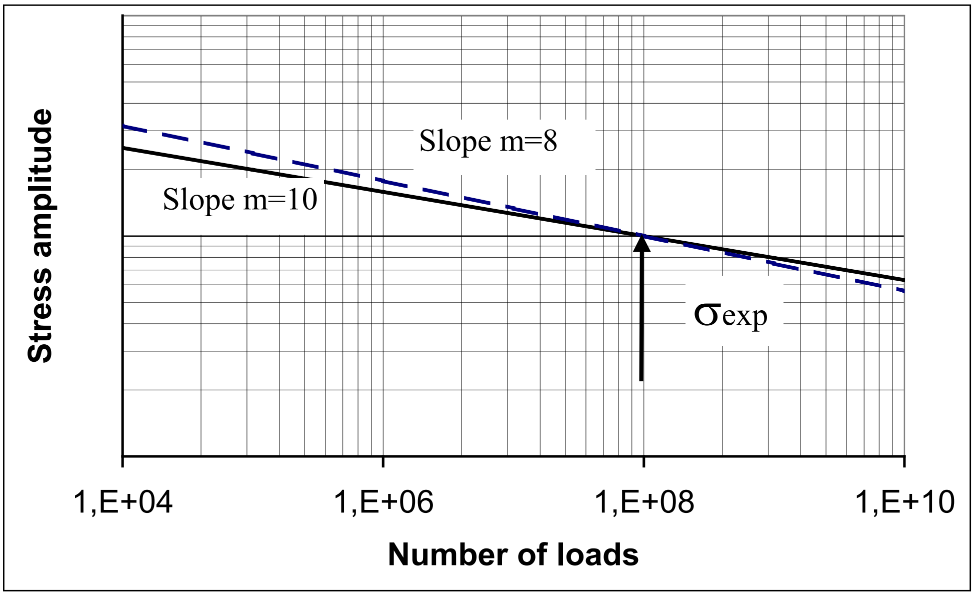


FIGURE 14
Constant-slope S-N Curve *(2010)*



i) *Equivalent fatigue stress.* The equivalent fatigue stress for 10^8 (100 million) stress cycles which produces the same fatigue damage as the load distribution for the service life of the ship is:

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

where

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

= $0.5 \cdot [(\sigma_{ice})_{f\max} - (\sigma_{ice})_{b\max}]$

$(\sigma_{ice})_{f\max}$ = principal stress resulting from forward load

$(\sigma_{ice})_{b\max}$ = principal stress resulting from backward load

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

ii) *Calculation of ρ parameter for two-slope S-N curve.* The parameter ρ relates the maximum ice load to the distribution of ice loads according to the regression formula:

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

where

σ_{fl} = $\gamma_{\varepsilon 1} \cdot \gamma_{\varepsilon 2} \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp}$

$\gamma_{\varepsilon 1}$ = 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

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

The following values are to be used for the reduction factors if actual values are not available: $\gamma_\varepsilon = \gamma_{\varepsilon 1} \cdot \gamma_{\varepsilon 2} = 0.67$, $\gamma_v = 0.75$, and $\gamma_m = 0.75$.

The coefficients C_1, C_2, C_3, and C_4 are given in 6-1-6/Table 8, below. The applicable range of N_{ice} for calculating ρ is $5 \times 10^6 \le N_{ice} \le 10^8$.

TABLE 8
Coefficients *C* (*1 July 2019*)

	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

iii) *Calculation of ρ parameter for constant-slope S-N curve.* For materials with a constant-slope S-N curve, see 6-1-6/Figure 14, the ρ-factor is to be calculated with the following formula:

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

where

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

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

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

Values for the G parameter are given in 6-1-6/Table 7. Linear interpolation may be used to calculate the G value for other m/k ratios than given in the 6-1-6/Table 9.

TABLE 9
Value for the G Parameter for Different m/k Ratios *(1 July 2019)*

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

27.11.2(d) Acceptability criterion for fatigue. The equivalent fatigue stress at all locations on the blade is to fulfill the following acceptability criterion:

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

where

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

$\gamma_{\varepsilon 1}$ = 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

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

The following values are to be used for the reduction factors if actual values are not available: $\gamma_\varepsilon = \gamma_{\varepsilon 1} \cdot \gamma_{\varepsilon 2} = 0.67$, $\gamma_v = 0.75$, and $\gamma_m = 0.75$.

27.11.3 Propeller Bossing and CP Mechanism

The blade bolts, the CP mechanism, the propeller boss, and the fitting of the propeller to the propeller shaft shall be designed to withstand the maximum and fatigue design loads, as defined in 6-1-6/27.9. The safety factor against yielding shall be greater than 1.3 and that against fatigue greater than 1.5. In addition, the safety factor for loads resulting from loss of the propeller blade through plastic bending as defined in 6-1-6/27.9.4 is to be greater than 1.0 against yielding.

27.11.4 Propulsion Shaft Line *(1 July 2019)*

The shafts and shafting components, such as the thrust and stern tube bearings, couplings, flanges and sealings, are to be designed to withstand the propeller/ice interaction loads as given in 6-1-6/27.9. The safety factor is to be at least 1.3 against yielding for extreme operational loads, 1.5 for fatigue loads and 1.0 against yielding for the blade failure load.

27.11.4(a) Shafts and shafting components (1 July 2019). The ultimate load resulting from total blade failure as defined in 6-1-6/27.9.4 is not to cause yielding in shafts and shaft components. The loading shall consist of the combined axial, bending, and torsion loads, wherever this is significant. The minimum safety factor against yielding is to be 1.0 for bending and torsional stresses.

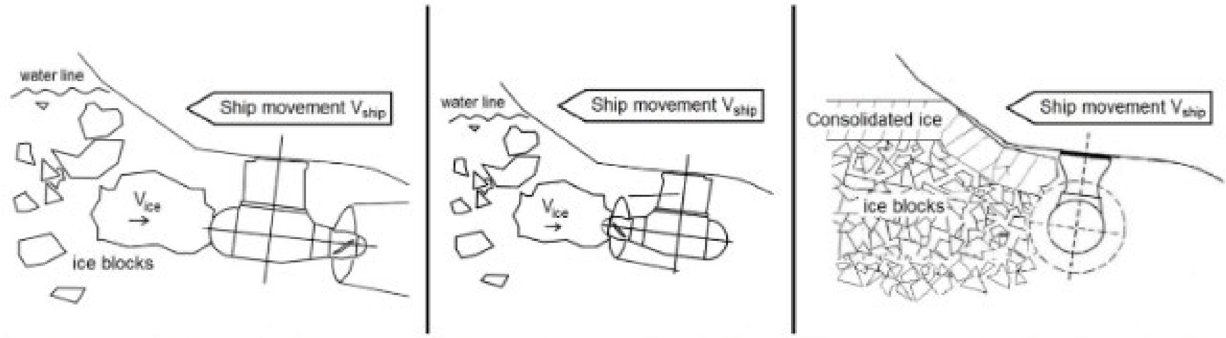
Note: The requirements in this section are complementary to those described in Section 4-3-2. For fatigue evaluation, cumulative fatigue analyses are to be performed (see 6-1-3/11.1.3 for recommended method/practice). The applicable Q_{peak} and the corresponding load spectrum shall be determined for the component or connection in question, as described in 6-1-6/27.9.3.

27.11.5 Azimuthing Main Propulsors

27.11.5(a) Design principle (1 July 2019). In addition to the above requirements for propeller blade dimensioning, azimuthing thrusters must be designed for thruster body/ice interaction loads. Load formulae are given for estimating once-in-a-lifetime extreme loads on the thruster body, based on the estimated ice condition and ship operational parameters. Two main ice load scenarios have been selected for defining the extreme ice loads. Examples of loads are illustrated in 6-1-6/Figure 15. In addition, blade order thruster body vibration responses may be estimated for propeller excitation. The following load scenario types are considered:

1. Ice block impact on the thruster body or propeller hub
2. Thruster penetration into an ice ridge that has a thick consolidated layer
3. Vibratory response of the thruster at blade order frequency

FIGURE 15
Examples of Load Scenario Types *(1 July 2019)*



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

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

27.11.5(b) Extreme ice impact loads (1 July 2019). When the ship is operated in ice conditions, ice blocks formed in channel side walls or from the ridge consolidated layer may impact on the thruster body and the propeller hub. Exposure to ice impact is very much dependent on the ship size and ship hull design, as well as the location of the thruster. The contact force will grow in terms of thruster/ice contact until the ice block reaches the ship speed.

The thruster must withstand the loads occurring when the design ice block defined in 6-1-6/Table 3 impacts on the thruster body when the ship is sailing at a typical ice operating speed. Load cases for impact loads are given in 6-1-6/Table 10. The contact geometry is estimated to be hemispherical in shape. If the actual contact geometry differs from the shape of the hemisphere, a sphere radius must be estimated so that the growth of the contact area as a function of penetration of ice corresponds as closely as possible to the actual geometrical shape penetration.

TABLE 10
Load Cases for Azimuthing Thruster Ice Impact Loads *(1 July 2019)*

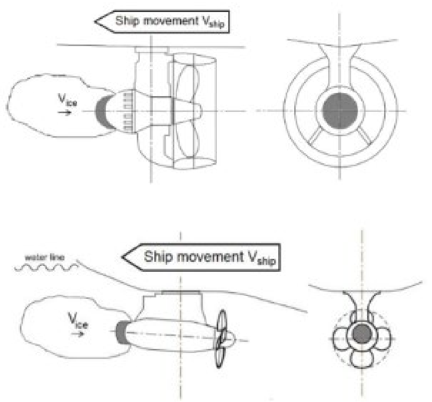
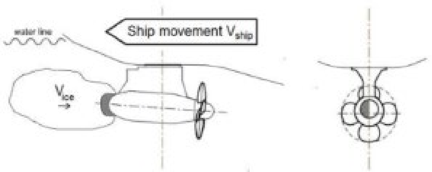
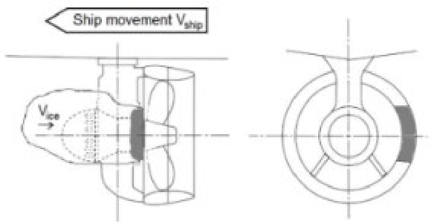
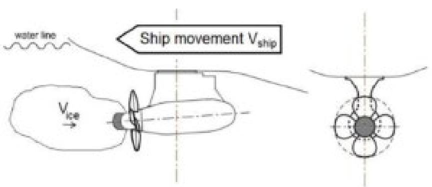
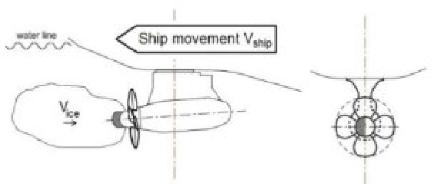
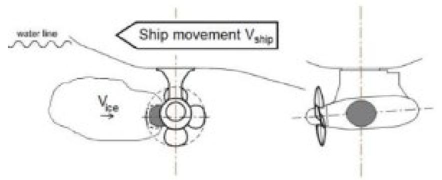
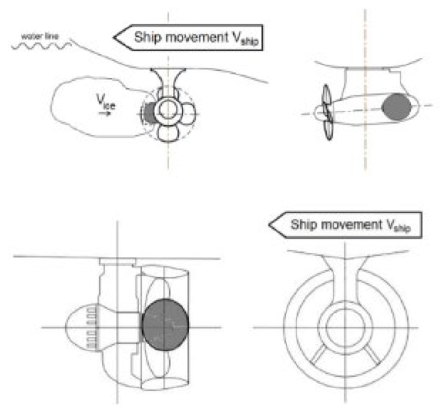
	Force	*Loaded Area*	
Load case T1a Symmetric longitudinal ice impact on thruster	F_{ti}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T1b Non-symmetric longitudinal ice impact on thruster	50% of F_{ti}	Uniform distributed load or uniform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

TABLE 10 (continued)
Load Cases for Azimuthing Thruster Ice Impact Loads *(1 July 2019)*

	Force	*Loaded Area*	
Load case T3a Symmetric lateral ice impact on thruster body	F_{ti}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T3b Non-symmetric lateral ice impact on thruster body or nozzle	F_{ti}	Uniform distributed load or uniform pressure, which are applied on the impact area. Nozzle contact radius *R* to be taken from the nozzle length (L_{nz}).	

The ice impact contact load must be calculated using the formula below. The related parameter values are given in 6-1-6/Table 11. The design operation speed in ice can be derived from 6-1-6/Table 12 and 6-1-6/Table 13, or the ship in question's actual design operation speed in ice can be used. The longitudinal impact speed in 6-1-6/Table 12 and 6-1-6/Table 13 refers to the impact in the thruster's main operational direction. For the pulling propeller configuration, the longitudinal impact speed is used for load case T2, impact on hub; and for the pushing propeller unit, the longitudinal impact speed is used for load case T1, impact on thruster end cap. For the opposite direction, the impact speed for transversal impact is applied.

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

$$F_{ti} = C_{DMI}\, 2113.85\, R_c^{0.5}\, (m_{ice} v_s^2)^{0.333} \quad \text{lbf}$$

where

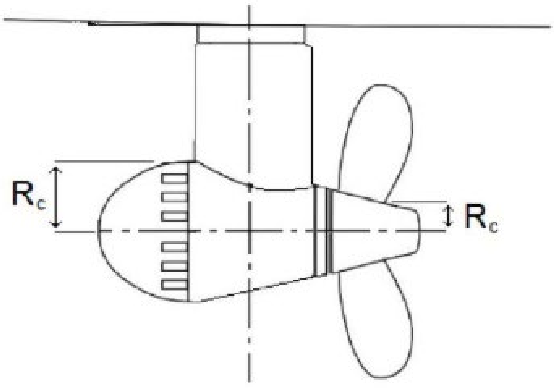
R_c = impacting part sphere radius, in m (ft), see 6-1-6/Figure 16

m_{ice} = ice block mass, in kg (lbs)

v_s = ship speed at the time of contact, in m/s (knots)

C_{DMI} = dynamic magnification factor for impact loads. If unknown, it shall be taken from 6-1-6/Table 11

FIGURE 16
Dimensions used for R_c (1 July 2019)



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

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

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

TABLE 11
Parameter Values for Ice Dimensions and Dynamic Magnification *(1 July 2019)*

	I AA	I A	I B	I C
Thickness of the design ice block impacting thruster (2/3 of H_{ice})	1.17 m 3.84 ft	1.0 m 3.28 ft	0.8 m 2.62 ft	0.67 m 2.2 ft
Extreme ice block mass (m_{ice})	8670 kg 19114 lb	5460 kg 12037 lb	2800 kg 6173 lb	1600 kg 3527 lb
C_{DMI} (if not known)	1.3	1.2	1.1	1

TABLE 12
Impact Speeds for Aft Centerline Thruster *(1 July 2019)*

Aft Centerline Thruster	I AA	I A	I B	I C
Longitudinal impact in main operational direction	6 m/s 11.67 knot	5 m/s 9.72 knot	5 m/s 9.72 knot	5 m/s 9.72 knot
Longitudinal impact in reversing direction (pushing unit propeller hub or pulling unit cover end cap impact)	4 m/s 7.78 knot	3 m/s 5.83 knot	3 m/s 5.83 knot	3 m/s 5.83 knot
Transversal impact in bow first operation	3 m/s 5.83 knot	2 m/s 3.89 knot	2 m/s 3.89 knot	2 m/s 3.89 knot
Transversal impact in stern first operation (double acting ship)	4 m/s 7.78 knot	3 m/s 5.83 knot	3 m/s 5.83 knot	3 m/s 5.83 knot

TABLE 13
Impact Speeds for Aft Wing, Bow Centerline and Bow Wing Thrusters *(1 July 2019)*

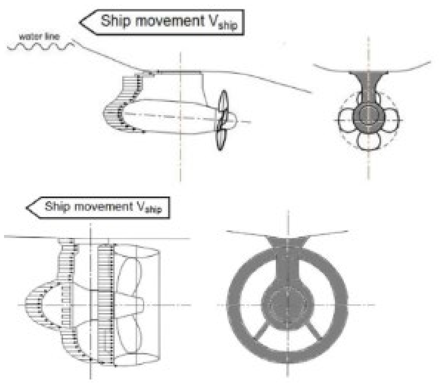
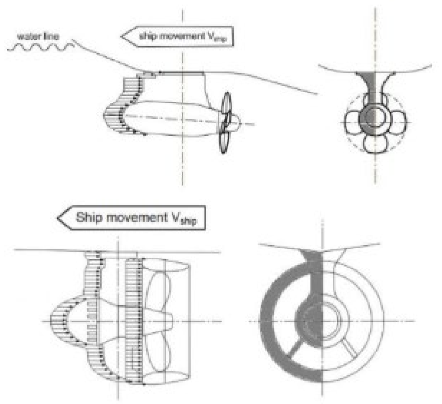
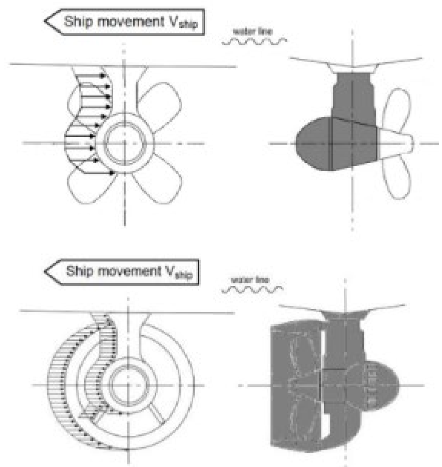
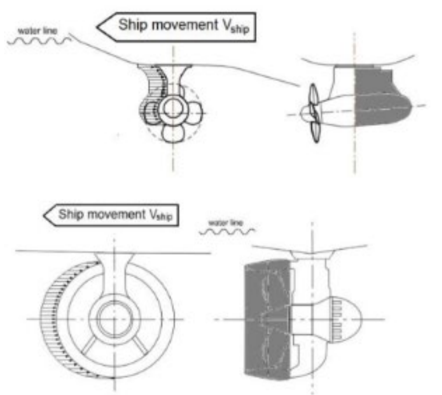
Aft Wing, Bow Centerline and Bow Wing Thruster	**I AA**	**I A**	**I B**	**I C**
Longitudinal impact in main operational direction	6 m/s 11.67 knot	5 m/s 9.72 knot	5 m/s 9.72 knot	5 m/s 9.72 knot
Longitudinal impact in reversing direction (pushing unit propeller hub or pulling unit cover end cap impact)	4 m/s 7.78 knot	3 m/s 5.83 knot	3 m/s 5.83 knot	3 m/s 5.83 knot
Transversal impact	4 m/s 7.78 knot	3 m/s 5.83 knot	3 m/s 5.83 knot	3 m/s 5.83 knot

27.11.5(c) Extreme ice loads on thruster hull when penetrating an ice ridge (1 July 2019). In icy conditions, ships typically operate in ice channels. When passing other ships, ships may be subject to loads caused by their thrusters penetrating ice channel walls. There is usually a consolidated layer at the ice surface, below which the ice blocks are loose. In addition, the thruster may penetrate ice ridges when backing. Such a situation is likely in the case of **IAA** ships in particular, because they may operate independently in difficult ice conditions. However, the thrusters in ships with lower ice classes may also have to withstand such a situation, but at a remarkably lower ship speed.

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

The load magnitude must be estimated for the load cases shown in 6-1-6/Table 14, using the equation after 6-1-6/Table 14. The parameter values for calculations are given in 6-1-6/Table 15 and 6-1-6/Table 16. The loads must be applied as uniform distributed load or uniform pressure over the thruster surface. The design operation speed in ice can be derived from 6-1-6/Table 15 or 6-1-6/Table 16. Alternatively, the actual design operation speed in ice of the ship in question can be used.

TABLE 14
Load Cases for Azimuthing Thruster Ice Impact Loads *(1 July 2019)*

	Force	*Loaded Area*	
Load case T4a Symmetric longitudinal ridge penetration loads	F_{tr}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T4b Non-symmetric longitudinal ridge penetration loads	50% of F_{tr}	Uniform distributed load or uniform pressure, which are applied on the other half of the contact area.	
Load case T5a Symmetric lateral ridge penetration loads for ducted azimuthing unit and pushing open propeller unit	F_{tr}	Uniform distributed load or uniform pressure, which are applied symmetrically on the contact area.	
Load case T5b Non-symmetric lateral ridge penetration loads for all azimuthing units	50% of F_{tr}	Uniform distributed load or uniform pressure, which are applied on the other half of the contact area.	

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

$$F_{tr} = 274.432\, v_s^{0.66}\, H_r^{0.9}\, A_t^{0.74} \quad \text{lbf}$$

where

v_s = ship speed, in m/s (knots)

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

A_t = projected area of the thruster, in m^2 (ft^2)

When calculating the contact area for thruster-ridge interaction, the loaded area in the vertical direction is limited to the ice ridge thickness, as shown in 6-1-6/Figure 17.

FIGURE 17
Schematic Figure Showing the Reduction of the Contact Area by the Maximum Ridge Thickness *(1 July 2019)*

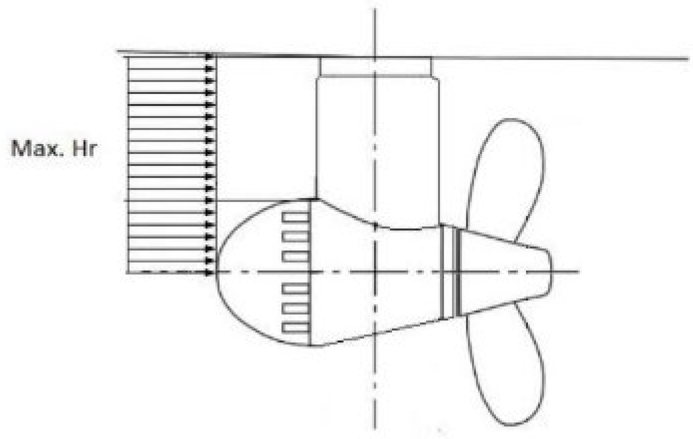


TABLE 15
Parameters for Calculating Maximum Loads when the Thruster Penetrates an Ice Ridge – Aft Thrusters, Bow First Operation *(1 July 2019)*

	I AA	I A	I B	I C
Thickness of the design ridge consolidated layer	1.5 m 4.92 ft	1.5 m 4.92 ft	1.2 m 3.94 ft	1.0 m 3.28 ft
Total thickness of the design ridge, H_r	8 m 26.25 ft	8 m 26.25 ft	6.5 m 21.33 ft	5 m 16.40 ft
Initial ridge penetration speed (longitudinal loads)	4 m/s 7.78 knot	2 m/s 3.89 knot	2 m/s 3 3.89 knot	2 m/s 3.89 knot
Initial ridge penetration speed (transversal loads)	2 m/s 3.89 knot	1 m/s 1.94 knot	1 m/s 1.94 knot	1 m/s 1.94 knot

TABLE 16
Parameters for Calculating Maximum Loads when the Thruster Penetrates an Ice Ridge – Thruster First Mode such as Double Acting Ships *(1 July 2019)*

	I AA	I A	I B	I C
Thickness of the design ridge consolidated layer	1.5 m 4.92 ft	1.5 m 4.92 ft	1.2 m 3.94 ft	1.0 m 3.28 ft
Total thickness of the design ridge, H_r	8 m 26.25 ft	8 m 26.25 ft	6.5 m 21.33 ft	5 m 16.40 ft
Initial ridge penetration speed (longitudinal loads)	6 m/s 11.66 knot	4 m/s 7.78 knot	4 m/s 7.78 knot	4 m/s 7.78 knot
Initial ridge penetration speed (transversal loads)	3 m/s 5.83 knot	2 m/s 3.89 knot	2 m/s 3.89 knot	2 m/s 3.89 knot

27.11.5(d) Acceptability criterion for static loads (1 July 2019). The stresses on the thruster must be calculated for the extreme once-in-a-lifetime loads described in 6-1-6/27.11.5. The nominal von Mises stresses on the thruster body must have a safety margin of 1.3 against the yielding strength of the material. At areas of local stress concentrations, stresses must have a safety margin of 1.0 against yielding. The slewing bearing, bolt connections and other components must be able to maintain operability without incurring damage that requires repair when subject to the loads given in 6-1-6/27.11.5(b) and 6-1-6/27.11.5(c) multiplied by a safety factor of 1.3.

27.11.5(e) Thruster body global vibration (1 July 2019). Evaluating the global vibratory behavior of the thruster body is important, if the first blade order excitations are in the same frequency range with the thruster global modes of vibration, which occur when the propeller rotational speeds are in the high power range of the propulsion line. This evaluation is mandatory and it must be shown that there is either no global first blade order resonance at high operational propeller speeds (above 50% of maximum power) or that the structure is designed to withstand vibratory loads during resonance above 50% of maximum power.

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

27.13 Alternative Design Procedure

27.13.1 Scope

As an alternative to 6-1-6/27.9 and 6-1-6/27.11, a comprehensive design study may be carried out to the satisfaction of the Administration. The study is to be based on ice conditions given for different ice classes in 6-1-6/27.5. It is to include both fatigue and maximum load design calculations and fulfill the pyramid strength principle, as given in 6-1-6/27.11.1.

27.13.2 Loading

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

27.13.3 Design Levels

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

Cumulative fatigue damage calculations are to indicate a reasonable safety factor. Due account is to be taken of material properties, stress raisers, and fatigue enhancements.

Vibration analysis is to be carried out and is to demonstrate that the overall dynamic system is free from harmful torsional resonances resulting from propeller/ice interaction.

28 Tunnel Thrusters *(2018)*

Where **APS**, **PAS** or Dynamic Positioning Systems Notations are assigned, the mechanical components of a tunnel thruster (i.e., propellers, gears, shafts, couplings, etc.) are to meet the applicable requirements of propulsion systems in this Section.

Alternatively, Section 4-3-5 may be applied to the mechanical components of a tunnel thruster when a comprehensive study to determine the effect of ice is submitted for consideration.

29 Additional Ice Strengthening Requirements

29.1 Starting Arrangements

The capacity of the air receivers required for reversible propulsion engines is to be sufficient for at least twelve consecutive starts and that for non-reversible propulsion engines is to be sufficient for six consecutive starts of each engine.

If the air receivers supply systems other than starting the propulsion engines, the additional capacity of the receivers is to be sufficient for continued operations of these systems after the capacity for the required number of consecutive engine starts has been used.

The capacity of the air compressors is to be sufficient for charging the air receivers from atmospheric to full pressure in one hour. For a vessel with ice class **I AA** that requires its propulsion engines to be reversed for astern operations, the compressors are to be able to charge the air receivers in half an hour.

29.3 Sea Inlet, Cooling Water Systems and Fire Main *(1 July 2019)*

The sea water system is to be designed to ensure a supply of water for the cooling water system and for at least one of the fire pumps when navigating in ice. For this purpose, at least one sea water inlet chest is to be arranged as follows.

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

ii) Guidance for designing the volume of the chest shall be around one cubic meter (35.3 cubic foot) for every 750 kW (1033 mhp; 1019 hp) in engine output of the ship, including the output of auxiliary engines necessary for the operation of the ship.

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

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

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

Where it is impractical to meet the requirements of 6-1-6/29.3ii) and 6-1-6/29.3iii) above, two smaller sea chests may be arranged for alternating the intake and discharge of the cooling water, provided 6-1-6/29.3i), 6-1-6/29.3iv) and 6-1-6/29.3v) above are complied with.

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

The use of ballast water for cooling purposes while in the ballast condition may be acceptable as an additional means but is not to be considered a permanent substitute for the above required sea inlet chest or chests.

PART
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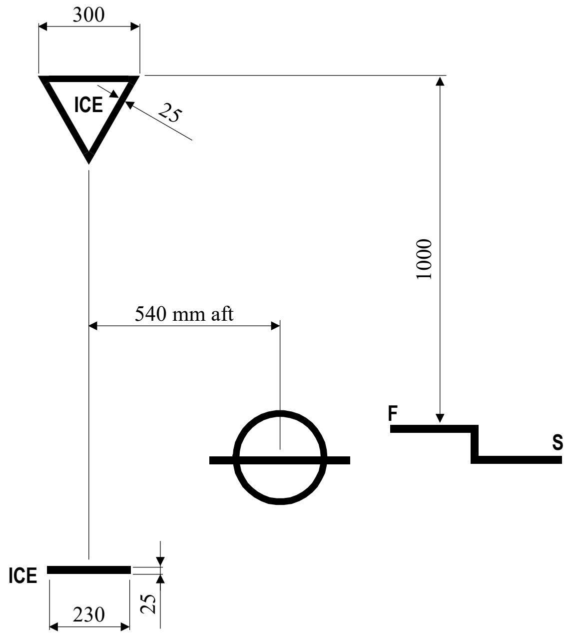
CHAPTER 1 Strengthening for Navigation in Ice

SECTION 6 Appendix 1 – Ice Class Draft Marking *(1 July 2009)*

Subject to 6-1-6/7, the vessel's sides are to be provided with a warning triangle and with a draft mark at the maximum permissible ice class draft amidships (see 6-1-6A1/Figure 1). The purpose of the warning triangle is to provide information on the draft limitation of the vessel when it is sailing in ice for masters of ice breakers and for inspection personnel in ports.

FIGURE 1
Ice Class Draft Marking *(1 July 2009)*



Notes:

1 The upper edge of the warning triangle is to be located vertically above the "ICE" mark, 1000 mm higher than the Summer Load Line in fresh water but in no case higher than the deck line. The sides of the triangle are to be 300 mm in length.

2 The ice class draft mark is to be located 540 mm abaft the center of the load line ring or 540 mm abaft the vertical line of the timber load line mark, if applicable.

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

4 The dimensions of all figures are to be the same as those used in the load line mark.

PART

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CHAPTER 1 Strengthening for Navigation in Ice

SECTION 6 Appendix 2 – Parameters and Calculated Minimum Engine Power for Sample Ships

(1 July 2019)

For checking the results of calculated powering requirements, 6-1-6A2/Table 1 presents input data for a number of sample ships.

TABLE 1
Parameters and Calculated Minimum Engine Power of Sample Ships *(1 July 2019)*

	Sample Ship Number								
	1	2	3	4	5	6	7	8	9
Ice Class	**IAA**	**IA**	**IB**	**IC**	**IAA**	**IAA**	**IA**	**IA**	**IB**
α, degrees	24	24	24	24	24	24	36	20	24
φ_1, degrees	90	90	90	90	30	90	30	30	90
φ_2, degrees	30	30	30	30	30	30	30	30	30
L, m (ft)	150 (492)	150 (492)	150 (492)	150 (492)	150 (492)	150 (492)	150 (492)	150 (492)	150 (492)
B, m (ft)	25 (82)	25 (82)	25 (82)	25 (82)	25 (82)	22 (72)	25 (82)	25 (82)	25 (82)
T, m (ft)	9 (29.5)	9 (29.5)	9 (29.5)	9 (29.5)	9 (29.5)	9 (29.5)	9 (29.5)	9 (29.5)	9 (29.5)
$LBOW$, m (ft)	45 (147.6)	45 (147.6)	45 (147.6)	45 (147.6)	45 (147.6)	45 (147.6)	45 (147.6)	45 (147.6)	45 (147.6)
$LPAR$, m (ft)	70 (229.7)	70 (229.7)	70 (229.7)	70 (229.7)	70 (229.7)	70 (229.7)	70 (229.7)	70 (229.7)	70 (229.7)
A_{wf}, m^2 (ft^2)	500 (5382)	500 (5382)	500 (5382)	500 (5382)	500 (5382)	500 (5382)	500 (5382)	500 (5382)	500 (5382)
DP, m (ft)	5 (16.4)	5 (16.4)	5 (16.4)	5 (16.4)	5 (16.4)	5 (16.4)	5 (16.4)	5 (16.4)	5 (16.4)
Prop. No./Type	1/CP	1/CP	1/CP	1/CP	1/CP	1/CP	1/CP	1/CP	1/FP
New Ships kW (HP)	7840 (10514)	4941 (6626)	3478 (4664)	2253 (3021)	6799 (9118)	6406 (8591)	5343 (7165)	5017 (6728)	3872 (5192)
Existing Ships kW (HP)	9192 (12327)	6614 (8870)	-	-	8466 (11353)	7645 (10252)	6614 (8870)	6614 (8870)	-

PART

6

CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

CONTENTS

Note: Text in italics is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

PART
6

CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

SECTION 1 General

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 Classification

For details of the scope and condition of classification, refer to Part 1, Chapter 1. However, for cargo or container vessels carrying refrigerated cargo, when specific notation related to this capability is requested by the Owners or builders, the following requirements will also apply.

3 Cross-references *(2004)*

Where necessary, applicable requirements in the *Rules for Building and Classing Steel Vessels* have been cross-referenced. It should be understood that for vessels under 90 m (295 ft) in length, the corresponding requirements in the *Rules for Building and Classing Steel Vessels Under 90 meters (295 feet) in Length* are to be applied in lieu of those in the *Rules for Building and Classing Steel Vessels*. For integrated cargo and ballast systems for refrigerated edible bulk liquid tankers, see requirements in 5C-1-7/33.

5 Application

5.1

The requirements of this Chapter are applicable to steel vessels intended to carry refrigerated cargoes such as fruits, vegetables, meat, fish, or other perishable goods in the hold spaces, or in the case of edible bulk liquids, in cargo tanks under controlled temperature conditions and, where fitted, also at controlled atmosphere. These vessels, except in the case of the refrigerated edible bulk liquid tankers, may carry cargoes in bulk, break bulk or palletized in the hold spaces or in refrigerated containers of porthole or plug-in types.

5.3

There are a number of requirements in this Chapter which relate to the safety of the vessel and personnel onboard and therefore, regardless of the notations referred to in 6-2-1/7, are considered necessary as conditions of classification (i.e., compulsory requirements). These requirements, which are shown in Arial italics, are to be applied for all vessels intended to carry refrigerated cargoes.

5.5

The requirements of this Chapter are applicable to those features that are permanent in nature and can be verified by plan review, calculations, physical survey or other appropriate means.

5.7 *(2017)*

The requirements of this Chapter are not applicable to the refrigeration system for a liquefied gas carrier, nor to air conditioning systems or refrigeration systems for provision storage.

7 Class Notations

7.1 Vessels Built Under Survey

Vessels intended for the carriage of refrigerated cargoes, which comply with the requirements of the Rules, and which have been constructed, at the request of the Owners, under survey by the Surveyors, will be distinguished in the *Record* by one of the following notations, as appropriate, followed by the date of survey.

✠ **RCC**	Refrigerated Cargo Carrier
✠ **RC(Hold No.)**	Refrigerated Cargo Carrier – Some Holds Only
✠ **RCCC**	Refrigerated Cargo Container Carrier
✠ **IRCC**	Integral Refrigerated Container Carrier
✠ **REBLT**	Refrigerated Edible Bulk Liquid Tankers
✠ **RFC**	Refrigerated Fish Carrier

7.1.1 Vessels Carrying Cargo in Refrigerated Holds, ✠ **RCC**

Where cargo is carried in refrigerated holds, the *Record* will give the number and state the capacity of the insulated cargo spaces which have been examined, the character of the insulation, a description of the refrigeration machinery and the associated system, the minimum design temperature of each zone attainable with the maximum design ambient and sea water temperature.

The conditions specified in the *Record* will be subject to verification by testing in the presence of Surveyors.

7.1.2 Vessels Carrying refrigerated Cargo in Some of the Cargo Hold(s), ✠ **RC(Hold Number(s))**

Where there are facilities provided onboard the vessel for carriage of refrigerated cargo in some of the cargo hold(s), the *Record* will give the refrigerated cargo hold number(s), the capacity and the characteristics of the insulation, description of the refrigeration machinery and the minimum design temperature attainable with the maximum design ambient and sea water temperatures.

The conditions specified in the *Record* are subject to verification by testing in the presence of Surveyors.

7.1.3 Vessels Carrying Cargo in Refrigerated Containers of Porthole Type, ✠ **RCCC**

Where cargo is carried in refrigerated containers, individually cooled by the shipboard refrigerated machinery and the associated systems and, where fitted, the associated temperature monitoring and control system, the Record will give the number and average design thermal characteristics of the containers, description of the refrigeration machinery and the distribution system for refrigerating the individual containers (porthole type only).

The conditions specified in the *Record* are subject to verification by testing in the presence of Surveyors.

7.1.4 Vessels Carrying Cargo in Refrigerated Containers of Integral Type, ✠ **IRCC** *(1 July 2016)*

Where cargo is carried in refrigerated containers of plug-in or integral types which has its own individually mounted refrigeration machinery, hence requiring shipboard electrical power supply and in some cases the cooling water supply for the condensers and, where fitted, the associated temperature monitoring and control system, the *Record* will give the total number of refrigerated containers onboard, the total design load in kW and the type of temperature monitoring and control system installed.

The conditions specified in the *Record* are subject to verification by testing in the presence of Surveyors.

In addition to the requirements of these Rules, compliance with the ABS *Guide for Carriage of Integral Refrigerated Containers On Board Ships* is required to receive the ✠ **IRCC** notation.

7.1.5 Vessels Carrying Edible Liquids in Bulk in Refrigerated Cargo Tanks, ✠ **REBLT**

Where edible products are carried in bulk in refrigerated cargo tanks cooled by their own shipboard refrigeration machinery and the associated system, the *Record* will give the cubic capacity and the maximum design pressure of the cargo tanks, the minimum permissible design temperature of the cargo, a description of the refrigeration machinery, the maximum design ambient and sea water temperatures.

The conditions specified in the *Record* will be subject to validation by testing in the presence of Surveyors prior to issuance of the certificate.

7.1.6 Vessels Carrying Fish in the Refrigerated Cargo Holds, ✠ **RFC**

Where fish processing or fish storage vessels are provided with facilities for chilling, cooling, or freezing and/or storage in the refrigerated cargo holds cooled by their own shipboard refrigeration machinery and the associated system, the *Record* will give the number and state capacity of the insulated cargo spaces which have been examined, the character of the insulation, a description of the refrigeration machinery and the associated system, the minimum design temperature of each space attainable with the maximum design ambient and sea water temperature.

The conditions specified in the *Record* will be subject to verification by testing in the presence of Surveyors.

7.3 Vessels Not Built Under Survey

Vessels intended for the carriage of refrigerated cargoes, which have not been constructed under survey by the Surveyors, but which have been subsequently surveyed at the request of the Owners, satisfactorily reported upon by the Surveyor, and which comply with the requirements of this Chapter, will be distinguished in the *Record* by one of the notations listed in 6-2-1/7.1, as appropriate, but the mark ✠ signifying survey during construction will be omitted.

7.5 RMC Notation for Existing Vessels

Existing vessels intended for the carriage of refrigerated cargoes, which have not been constructed and installed under ABS survey, and which do not fully meet the requirements in this Chapter, but which are submitted for classification, will be subject to special classification survey. The refrigerated cargo holds and refrigeration machinery of such vessels are to comply with Part 4, Section 12 of the *Rules for Building and Classing Steel Vessels* (1997 edition). Where found satisfactory and thereafter approved by the Committee, they will be classed and distinguished in the *Record* by symbol **RMC**.

9 Supplemental Notations

9.1 Controlled Atmosphere, ✠ CA

At the request of the Owner or the builder, refrigerated cargo vessels fitted with equipment and systems including the associated safety features which have been constructed and installed for compliance with the requirements of Section 6-2-12 will be distinguished in the *Record* ✠ **CA** (date of survey).

9.3 Controlled Atmosphere Installation, ✠ CA (INST)

At the request of the Owners or builders, refrigerated cargo vessels fitted with a permanently installed piping system and the associated safety features and which is ready for connection to the portable controlled atmosphere generating equipment which has been constructed and installed for compliance with the requirements of Section 6-2-12 will be distinguished in the *Record* ✠ **CA (INST)**.

9.5 Automatic Pallet Loading/Unloading System, ✠ APLUS

At the request of the Owners or builders, refrigerated cargo vessels fitted with a system whereby the cargo is loaded and unloaded to and from the refrigerated hold spaces through an automatic pallet handling, stacking and securing system together with a monitoring and control system which indicates the status of the pallets during the loading/unloading operation and having been constructed and installed in compliance with the applicable requirements will be distinguished in the *Record* ✠ **APLUS**.

9.7 Automatic or Semi-Automatic Side Loading System ✠ ASLS or ✠ SASLS

At the request of the owner or the builder, refrigerated cargo vessels fitted with a system whereby the cargo is loaded and unloaded to and from the refrigerated hold spaces through an automatic or semi-automatic side loading pallet handling system together with monitoring and control system which indicates the status of pallets during loading/unloading operation and having been constructed and installed in compliance with the applicable requirements will be distinguished in the *Record* ✠ **ASLS** or ✠ **SASLS**.

9.9 Fruit Carrier, (F)

At the request of the Owner or the builder, refrigerated cargo or container vessels intended for the carriage of fruit which have been constructed and installed in compliance with the applicable requirements will be distinguished in the *Record* **(F).**

11 Alternative Designs

11.1

Equipment designed and constructed to alternative national or international standards to those referred to in the Rules will be considered for acceptance based on the requirements of 1-1-4/7.

11.3

Where the design of the installation contains new features which have not been addressed in the Rules, these will be the subject of special consideration upon receipt of the details such as drawings, data, calculations and, where considered necessary, analysis.

11.5

Refrigerants other than those mentioned in the Rules may be used, provided they are considered to be adequate for use in shipboard applications in accordance with national or international standards, international treaties adopted by the government(s) and the flag states or other similar legislation laid down by the flag state.

For the purpose of class, details such as the chemical properties, toxicity, flammability, together with the supporting data are to be submitted for review.

13 Definitions

13.1 Direct Expansion

A refrigeration system, in which the refrigerant expansion occurs through the direct absorption of heat from the primary medium to be cooled.

13.3 Indirect Expansion

A refrigeration system in which a secondary coolant is cooled by the direct expansion of a primary refrigerant and is then circulated to cool the medium which absorbs heat from the space to be cooled.

13.5 Refrigerant

The fluid used for heat transfer in a refrigeration system, which absorbs heat at a low temperature and low pressure of the fluid and rejects heat at a higher temperature and higher pressure of the fluid, usually involving a change of state of the fluid during the process.

13.7 Secondary Coolant

A liquid used for the transmission of heat, without a change of state, and having either no flash point or a flash point above 66°C (150°F).

13.9 Brine

Brine is a term given to secondary coolants which are water solutions of calcium chloride, sodium chloride and magnesium chloride.

13.11 Refrigerating Machinery Spaces

Refrigerating Machinery Spaces are spaces dedicated for housing refrigerating machinery and the associated equipment.

13.13 Refrigeration Unit

A *Refrigeration Unit* is the machinery comprising the compressor, the compressor's driving motor and a condenser, if fitted, independent of any other refrigeration machinery for provision stores or the air conditioning plant. In indirect refrigeration systems, the refrigeration unit also includes a brine or other secondary coolant cooler.

13.15 Refrigeration System

A *Refrigeration System* comprises one or more refrigeration units, together with the piping and ducting system as well as the equipment necessary for cooling the cargo and maintaining it at the required temperature.

13.17 Refrigerated Container

A portable container designed and constructed to a recognized international standard and primarily intended for carrying refrigerated cargo, and which is adequately insulated to reduce heat loss through the boundary walls and made air tight through effective seals.

There are two types of refrigerated containers referred to in this Chapter:

13.17.1 Port Hole Containers

The refrigerated containers where the cargo contained therein is cooled by cold air circulated by the vessel's refrigeration system through flexible connections.

13.17.2 Integrated or Plug-in Containers

The refrigerated containers which are fitted with an individual refrigeration unit either permanently installed or portable and requiring an electrical power supply, and where necessary, a cooling water supply from the vessel.

13.19 Controlled Atmosphere

For purposes of the Rules, a *Controlled Atmosphere* is where the oxygen concentration in the cargo space is reduced and the CO_2 concentration adjusted to the required levels by the introduction of high purity nitrogen or other suitable gas. The oxygen and CO_2 concentrations within the cargo space are then monitored and controlled throughout the loaded voyage.

13.21 Automatic Pallet Loading/Unloading System

An *Automatic Pallet Loading/Unloading System* is one that is intended to load from the quay side, stows within the hold, and unloads pallets. A stacking system is fitted within the holds, consisting of conveyors, transporters or other similar means together with the control equipment and lifting appliances for use to maneuver the pallets automatically.

13.23 Automatic or Semi-Automatic Side Loading System

An *Automatic or Semi-Automatic Side Loading System* is one that is intended to load from the quay side and deliver the pallets to the appropriate refrigerated hold using hoists, cranes, conveyors or other similar means together with the control equipment for use in maneuvering the pallets automatically or semi-automatically.

13.25 Refrigerated Edible Bulk Liquid Tankers

Tankers carrying refrigerated edible bulk liquid which is required to be maintained at a pre-specified temperature by means of the refrigeration system fitted on board the vessel.

13.27 Cargo Containment System

The *Cargo Containment System* for the carriage of edible bulk liquid cargoes referred to in 6-2-14/7 may consist of cargo tanks as below:

13.27.1 Integral Tanks

Integral Tanks mean a cargo containment envelope which forms part of ship's hull structure and which may be stressed in the manner and by the same loads which stress the contiguous hull structure and which is normally essential to the structural completeness of the ship's hull.

13.27.2 Independent Tank

An *Independent Tank* means a cargo containment envelope which is not contiguous with, or part of, the hull structure.

13.27.3 Gravity Tank

Gravity Tank means a tank having a design pressure not greater than 0.7 bar gauge at the top of the tank. A gravity tank may be an independent or integral tank.

13.27.4 Pressure Tank

Pressure Tank means a tank having a design pressure greater than 0.7 bar gauge. A pressure tank is to be an independent tank.

13.29 Refrigerated Fish Carrier

Fish processing vessels, fishing vessels and mother ships of fishing fleet which are provided with facilities for freezing fish and fish products.

PART
6

CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

SECTION 2 Plans and Data to be Submitted

1 Hull Construction Drawings

General Arrangement

Capacity Plan

Midship Section

Framing Plan

Scantling profile and decks

Bottom Construction, floors, girders, etc.

Inner bottom plating

Shell expansion

Deck plans

Pillars and girders

Watertight and deep tank bulkheads

Miscellaneous non-tight bulkheads used as structural supports

Shaft tunnel

Machinery casings, engine and main auxiliary foundations

Fore end construction

Aft end construction

Stern Frame and rudder

Shaft struts

Superstructures and deckhouses and their closing appliances

Hatches and hatch closing arrangements

Side Shell Door – Construction and locking and sealing arrangements

Ventilation systems on weather decks

Anchor handling arrangements

Foundation structure for cranes and other lifting devices

Plan of hull showing steel grades

Cargo securing manual

For stability review:

- Lines and body plan
- Hydrostatic curves
- Cross curves
- Stability information

Additional plans for container ships

- Stowage arrangement of containers including stacking loads
- Location of container supports and their connection to hull

3 Refrigerated Cargo Spaces

Details of insulation installation including density, *K* factor, etc.

Details of the fixing arrangements for the load bearing supports of the insulation and linings, and of all other insulation support fittings embedded by the insulation.

Details of the weld designs for the attachment of the fittings to the vessel's structure

Proposed arrangements for fixing insulation to the vessel's structure

Details of the fasteners used for supporting pipework embedded in insulation.

Cargo space heating arrangements (where fitted)

Corrosion protection of the steel structure

Temperature gradient calculations

5 Refrigeration System and Refrigeration Machinery Spaces

Design pressure and temperature of the refrigeration system

Details of the refrigerant and secondary coolant

Heat-load calculations and refrigeration capacity, including rate of ventilation of the cargo spaces, where applicable.

Details of the compressors, prime-mover drive, condensers, receivers, pumps, thermostatic expansion valves, oil recovery equipment, filters and dryers, evaporators and other pressure vessels and heat exchangers

Piping diagrams of refrigerant, brine and condenser cooling system

Details of the air-coolers, including corrosion protection

General arrangement of refrigeration units, indicating location

Ventilation details of refrigeration machinery spaces, including ventilation rates

Capacity calculations for all of the pressure vessel safety relief valves

Details of the safety relief devices discharge piping, including design calculations

Corrosion protection of the refrigerant and brine pipes

Cargo hold defrosting arrangements

Drainage and bilge pumping arrangements

Location and types of portable fire extinguishers

Additional plans and data for the ammonia refrigeration system:

- Access arrangement to the refrigeration machinery spaces
- Details of the emergency ventilation system
- Details of the emergency drainage system
- Details of the sprinkler system and water screen devices
- Fixed ammonia detection system
- Details of the personnel safety equipment

7 Electrical Systems

Electrical one line wiring diagram for refrigeration machinery

Power supply and distribution

Arrangements of electrical equipment and cable way in refrigerating machinery spaces and refrigerated cargo holds including cable penetrations of insulated bulkheads and decks

Arrangements of thermometers in refrigerated cargo spaces

Heat tracing arrangements, where fitted

9 Instrumentation, Control and Monitoring Systems

Control and monitoring panels for refrigerating machinery including schematic diagrams, function description, construction plans and outline view

Operational description of automatic or remote control and monitoring systems including a list of alarms and displays

Computer-based systems are to include a block diagram showing system configuration including interface, description of hardware specifications, fail safe features and power supply

Control and monitoring

Temperature measuring system

Refrigerant leakage detection and alarm system

O_2 and CO_2 content measuring system

Ammonia vapor detection and alarm system

11 Cargo Handling Equipment

11.1 Cranes

Where certification is requested, then the drawing submittal is to be in accordance with Chapter 2, “Guide for Certification of Cranes” of the ABS *Guide for Certification of Lifting Appliances*.

- For Crane Structure: as per 2-1/3.3.1 of the above mentioned Guide
- For Crane Machinery, Piping and Electric Systems: as per 2-1/3.3.2 of the above mentioned Guide

11.3 Derrick and Booms

The drawing submittal is to be in accordance with Chapter 3, “Guide for Certification of Cargo Gear on Merchant Vessels” of the ABS *Guide for Certification of Lifting Appliances*.

11.5 Cargo Elevators

Where certification is requested, then the drawing submittal is to be in accordance with Chapter 2, "Guide for Certification of Cranes" of the ABS *Guide for Certification of Lifting Appliances*, as applicable.

- For Structure: as per 2-1/3.3.1 of the above mentioned Guide
- For Machinery, Piping and Electric Systems: as per 2-1/3.3.2 of the above mentioned Guide

11.7 Automatic Pallet Loading/Unloading System

11.7.1 Structural Plans

Stowage arrangement for pallets including stacking loads

Location of guide supports

Guide arrangement, scantlings, material grades and details

Details of the structural connections to the hull (including insulation)

Track, foundation and support structure for the lifting devices

Pallet securing arrangement and scantling plan

Deck openings, framing and closing appliance

Deckhouse

Operating manual

11.7.2 Electrical, Automation and Control

Rated load, rated speed and operating condition

Electric power installation including motor, control, wiring and protective devices

Details of controls, interlock, safety devices and brakes

Control and monitoring panels including schematic diagrams, function description, construction plans and outline view

Hydraulic and control piping system details

Arrangements for emergency operations

13 Automatic or Semi-Automatic Side Loading System

13.1 Structural Plans

Location of guide supports

Guide arrangement, scantlings, material grades and details

Details of the structural connections to the hull (including insulation)

Track, conveyors, foundation and support structure for the lifting devices

Deck openings, framing and closing appliance

Deck and Side shell openings, framing and reinforcement details, details of the closing appliances, locking and sealing arrangements

Deckhouse

Operating manual

13.3 Electrical, Automation and Control

Rated load, rated speed and operating condition

Electric power installation including motor, control, wiring and protective devices

Details of controls, interlock, safety devices and brakes

Control and monitoring panels including schematic diagrams, function description, construction plans and outline view

Hydraulic and control piping system details

Arrangements for emergency operations

15 Refrigerated Porthole Cargo Container System

Number and overall heat transfer rates of insulated cargo containers to be individually cooled by shipboard refrigeration system

Space heating arrangements for cargo cells

Details of the air ducting

Air circulation rates

Details of the flexible coupling, together with means of actuation

17 Refrigerated Integral Cargo Container System

Cooling water arrangements

Air freshening (ventilation) arrangements for cargo cells

19 Controlled Atmosphere

Capacity calculation for the nitrogen plant

Arrangements for controlling the CO_2 in cargo hold

Details of CO_2 and ethylene scrubber

Details of compressors and prime-movers

Details of the pressure vessels and heat exchangers

General arrangement of nitrogen generation plant, indicating location and access

Ventilation details of nitrogen generator space

Piping system, arrangement and details

Arrangements to render cargo spaces gas tight; to include details of liquid sealed traps

Arrangements for pressure and vacuum relief in cargo spaces

Ventilation arrangements, for designated controlled atmosphere spaces, and adjacent spaces

Schematic diagram of control and monitoring systems

One line electrical wiring diagram and details of the power supply

Details of the gas analyzing system

A list of alarms and displays

Details of the humidification system

Details of personnel safety equipment

Operations, equipment and procedure manual

21 Refrigerated Edible Bulk Liquid Tankers

Design specific gravity of cargo

Cargo tanks arrangements and details

Cargo tank construction and material details

Cargo tank foundations/supports (non-integral tanks)

Details of cargo tank coatings

Cargo pumping arrangements

Cargo tank refrigeration system

Cargo tank washing system

Nitrogen injection system for cargo tanks (where fitted)

Details of inert gas system, if provided

23 Refrigerated Fish Carriers

Details of the hull reinforcement (where provided)

Details of the cargo spaces, as per 6-2-2/3

Details of the refrigeration system and refrigeration machinery spaces, as per 6-2-2/5

Details of the refrigerated sea water (RSW) tanks

Details of the arrangement for protection of the Ammonia piping in cargo hold (direct expansion systems)

25 Onboard Tests and Trials

Test schedules for the tests and commissioning trials referred to in Section 6-2-16.

PART
6

CHAPTER **2 Vessels Intended to Carry Refrigerated Cargoes**

SECTION **3 Hull Construction**

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 General

1.1 Applicable Rules

1.1.1

Vessels intended to carry refrigerated cargo are to comply with the following Rules, as appropriate, for the purposes of obtaining Class:

1.1.1(a) For vessels 90 m (295 ft) and over in length, the hull construction and the fire safety arrangements are to be in accordance with the requirements of Part 3, Chapters 2 and 4 of these Rules.

1.1.1(b) For vessels under 90 m (295 ft) in length, the hull construction and the fire safety arrangements are to be in accordance with the requirements of Part 3, Chapters 2 and 4 of the Rules for Building and Classing Steel Vessels Under 90 m (295 ft) in Length*.*

1.1.1(c) Where the vessel is designed primarily for the carriage of containers in holds, or on deck, or both, with structures for that purpose, such as cell guides, pedestals, etc., the requirements of Part 5C, Chapter 5 and Part 5C, Chapter 6, are also applicable.

1.1.1(d) Commercial fishing vessels under 61 m (200 ft) in length are to be in accordance with Part 5, Chapter 12 of the Rules for Building and Classing Steel Vessels Under 90 m (295 ft) in Length*.*

1.1.1(e) Vessels intended to operate in areas with low temperatures for long periods are subject to special consideration. ABS offers the notation ***Ice Class****, followed by an ice class designation, for vessels built in accordance with Part 6, Chapter 1, "Strengthening for Navigation in Ice".*

1.1.2

This Section covers the additional items required for hull construction to obtain the classification notations ***✠ RCC****,* ***✠ RC (Hold No.)****,* ***✠ RCCC****,* ***✠ IRCC****,* ***✠ REBLT*** *and* ***✠ RFC****.*

3 Design Considerations

3.1 Design Temperatures – Steel Boundary of Refrigerated Cargo Spaces

3.1.1

Steel grades for plating and associated longitudinals and girders continuously exposed to temperatures below 0°C (32°F) in refrigerated cargo spaces should be based on the steel design service temperature submitted by the shipyard or Owner. When assessing the steel design service temperature, the temperature of the adjacent, internal, non-refrigerated space may be taken as +5°C (+9°F).

3.1.2

When the shipyard or Owner does not submit a temperature gradient calculation to assess the steel design service temperature, this temperature is to be determined as follows:

3.1.2(a) Un-insulated steel within the refrigerated cargo spaces is at the temperature of the space.

3.1.2(b) Steel insulated within the refrigerated cargo space but un-insulated on other side is at the temperature of the un-insulated side.

3.1.2(c) With steel insulated upon both sides, then the following will apply:

Where the temperature difference is less than 30°C (54°F), a mean temperature is to be used and where the temperature difference is greater than 30°C (54°F), the steel temperature is to be specially considered.

3.3 Avoidance of Notches and Hard Spots in Steel Work

Unless permitted elsewhere in the Rules, structural members are to be effectively connected to the adjacent structures so as to avoid hard spots, notches and other harmful stress concentrations. See 3-1-2/15.

3.5 Air Tightness of Refrigerated Cargo Spaces

Arrangements are to be made to prevent odors passing into the refrigerated cargo space from an external source, as follows:

3.5.1

Each independent cargo space is to be airtight and of steel construction.

3.5.2

The hatches, access doors, access hatches, bilge well plugs, tank top manhole plugs, etc. fitted in the insulated surfaces must have air-tight joints.

3.5.3

Ventilators are to be fitted with airtight closing appliances.

5 Materials

5.1 General

The materials used in the construction of the vessel are to be manufactured and tested in accordance with the requirements of Part 2, Chapter 1.

5.3 Steel Grades

Steel materials for hull construction are not to be of lower grades than those required for the material class for the particular location, as given in 3-1-2/3. Furthermore, for steel used for the construction of the refrigerated cargo spaces, the grade of steel is also to comply with 6-2-3/5.5, 6-2-3/5.7 and 6-2-3/5.9.

5.5 Toughness of Steel

The steel grade is to be chosen upon the basis of its toughness, measured by an impact test. For details, refer to Part 2, Chapter 1, in which the impact test requirements and provision are given for three grades (B, D, E) of normal strength steel. The higher strength steel (H32, H36 and H40) are each subdivided into four grades (A, D, E and F). There is no impact test requirement for Grade A steel of normal strength.

5.7 Areas Exposed to Low Temperatures (2007)

The material selection for the following areas of steel work is to be made on the basis of the design service temperature determined in accordance with 6-2-3/3.1.2 and the thickness. The minimum grades of steel to be used for the following are to be in accordance with 6-2-3/5.9:

- *Tween deck plating and longitudinals*
- *Longitudinal and transverse deck girders and deep side shell stringers (i.e., the portion of the tween deck outboard of a centerline hatch)*
- *Shelf plates, including web and face bars (i.e., the hatch covers supports)*
- *The longitudinal bulkhead strakes attached to deck plating and the longitudinal stiffeners on these strakes*
- *Pillars and vertical bulkhead web frames that replace pillars*

See also the requirements of 6-2-3/5.11 for selection of materials for hull structural members other than the above

5.9 Steel Grades for Areas Exposed to Low Temperature

5.9.1

The following minimum grades of steel are to be used for the areas given in 6-2-3/5.7:

0°C > T ≥ -10°C (32°F > T ≥ 14°F)	
t ≤ 12.5 (0.50)	*A*
12.5 (0.50) < t ≤ 19.0 (0.75)	*B/AH*
19.0 (0.75) < t ≤ 51.0 (2.00)	*D/DH*
-10°C > T ≥ -20°C (14°F > T ≥ -4°F)	
t ≤ 12.5 (0.50)	*B/AH*
12.5 (0.50) < t ≤ 27.5 (1.08)	*D/DH*
27.5 (1.08) < t ≤ 51.0 (2.00)	*E/EH*
-20°C > T ≥ -30°C (-4°F > T ≥ -22°F)	
t ≤ 22.5 (0.89)	*D/DH*
22.5 (0.89) < t ≤ 51.0 (2.00)	*E/EH*

T is design service temperature, in °C (°F)

t is steel thickness, in mm (in.)

5.9.2

Temperature lower than -30°C (-22°F) will be the subject of special consideration.

5.9.3

Steel castings or forgings used in the structure are to meet the same impact test requirements as that for steel plate in the same application.

5.11 For Other Areas of Hull Construction

5.11.1

The steel grades for areas other than given in 6-2-3/5.7 are to be as required by other relevant sections of the Rules for Building and Classing Steel Vessels*. These areas include the following:*

- *Exposed main deck plating and stiffening.*
- *Forecastle deck plating and stiffening.*
- *Inner bottom plating and stiffening.*
- *Transverse bulkheads plating and stiffening.*
- *Transverse deck beams, where fitted to every frame.*
- *Shell plating and shell framing.*

5.11.2

Where the design of these areas is of an unusual construction, the material grade will be the subject of special consideration.

7 Hatch Covers

7.1

The scantlings of the hatch covers are to be designed in accordance with the requirements of Section 3-2-15.

7.3

Main hatch covers for insulated compartments are to be provided with double sealing arrangements, as a minimum.

7.5

Exposed hatch covers to an insulated compartment are also to be weathertight in any sea condition and arrangements are to be made to ensure any water ingress is avoided by packing or by efficient drainage leading to the exposed deck, or by an alternative means approved by ABS.

9 Side Shell Doors

9.1 General

Side shell doors are to be designed in accordance with the applicable requirements of Section 3-2-16. In addition, the following requirements are also applicable.

9.1.1

Suitable arrangements are to be made to allow for ship's movement, to ensure that the watertight integrity of the side shell door is maintained in any sea condition.

9.1.2

Adequate structural stiffening is to be fitted at the hull/door sealing interface so that deflections of a local nature are avoided.

9.1.3

The longitudinal strength of the vessel will be subject to special consideration.

The hull girder strength calculations under the combined vertical and horizontal bending moment are to be submitted. The combined longitudinal hull girder stress is to be calculated at the critical points of the continuous longitudinal material e.g., turn of bilge port and starboard, at the intersection of the intact deck edge and sheer strake, and at the inboard corner of the deck opening. Alternatively, a more comprehensive analysis may be submitted for review. On request, this analysis can be carried out by ABS.

9.1.4

Structural continuity is to be maintained for the remaining longitudinal and transverse members inboard of the opening in the shell and deck.

9.1.5

Each section of a multi-section door is to satisfy the requirements of Section 3-2-16 independently of adjacent sections.

9.1.6

It is to be shown, from the ship's stability book or otherwise, that in conditions of loading or unloading, and when the ship is heeled by cargo or crane movement, or by offset tank weights, that the door edge is not immersed. The door is to have a sill above the uppermost load line of a minimum height of 0.06B. Alternative methods of preventing the ingress of water will be specially considered. Details in this regard are to be submitted for approval.

9.3 Side Loading Doors, Forming Part of the Deck and Sheer Strake

Where a side loading door, forming part of the deck and sheer strake, is fitted, there is an asymmetrical transverse structural section, whereby important longitudinal elements are not continuous at one side (i.e., the deck stringer plate and sheer strake). The above requirements in 6-2-3/9.1.1 through 6-2-3/9.1.6 are applicable in addition to the requirements of Section 3-2-16.

11 Insulation Supports and Fixtures within Refrigerated Cargo Spaces

Supports and fixtures for the insulation are to be suitable for their intended purpose, and in accordance with the following requirements:

11.1

All fixing arrangements of brackets, hangers, bolts, studs etc., and of their welded connections are to be capable of withstanding local loads caused by weight and thermal contraction/expansion and vibration.

11.3

The insulation and linings are to be fully supported.

11.5

The linings, side shorings, their supports and fixtures are to be able to withstand the loads imposed by the cargo.

11.7

Studs used for supporting the insulation panels are to be welded to the steel structure.

13 Fixing Arrangements for Cargo Securing Fittings within the Refrigerated Cargo Spaces

13.1

Stools or other permanent methods for securing cargo within a refrigerated cargo space, and which are welded to the hull, are to be arranged with a thermal break.

13.3

Stools are to be flush with the grating top.

15 Sealing of Doors and Access Hatches

Doors and hatches for access to insulated compartments are to be provided with a double sealing arrangement and are to be designed so that they can be opened from both sides.

17 Tests and Inspections

All spaces are to be tested for tightness by either a hose test before insulating the surfaces or a gas or smoke pressure test after insulating the space.

PART
6

CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

SECTION 4 Cargo Handling Equipment

1 Optional Certification

The following equipment used onboard for loading and unloading cargo may be certified by ABS, upon request by the Owner or Builder, for compliance with the requirements as indicated in 6-2-4/3.1, 6-2-4/3.3, 6-2-4/5, and 6-2-4/7:

- Shipboard cranes
- Derrick-and-boom cargo gear
- Automatic pallet loading and unloading system
- Side Loading System

3 Applicable Rules for Cranes, Derrick and Boom Cargo Gear, and Cargo Elevators

3.1 Cranes

A Certification of Lifting Appliances attesting to compliance with Chapter 2, "Guide for Certification of Cranes" of the ABS *Guide for Certification of Lifting Appliances* will be issued at the request of the Owner or Builder upon satisfactory completion of plan review, in-plant survey, installation and testing of the cranes to the satisfaction of the attending Surveyor. Vessels with this Certification will be distinguished in the *Record* by a notation **CRC** (Crane Register Certificate) with the number and capacity of cranes.

3.3 Derrick Post and Boom, and Cargo Elevators

For arrangements of derrick post and boom, and cargo elevators, the Owner or Builder may request ABS Cargo Gear Certification in accordance with Chapter 3, "Guide for Certification of Cargo Gear on Merchant Vessels" of the ABS *Guide for Certification of Lifting Appliances*. Appropriate certificates for attachment to the *Register of Cargo Gear* will be issued following satisfactory compliance with the above requirements.

5 Automatic Pallet Loading/Unloading System (✠ APLUS Notation)

5.1 General

In order to receive the ✠ **APLUS** notation, an automatic pallet loading/unloading system is to comply with this subsection.

5.3 Automatic Pallet Loading/Unloading System

An automatic pallet loading/unloading system is to be capable of loading, stowing and unloading pallets, and may include the following operations to be carried out automatically:

- Load the pallet from the quay to the deck.
- Transport the pallet to the designated location within the hold.
- Stow the pallet.
- Re-stow the pallet, if necessary.
- Secure the pallet for the voyage.
- Unload the pallet from the hold to the quay.

5.5 Structural Requirements for the Hold Pallet Guide Framework

5.5.1 Guide Framework Design

5.5.1(a) The material for the framework is to be suitable for the anticipated service temperature in the hold.

5.5.1(b) The framework for this method of transportation and stowage of cargo is to be designed for the carriage of standardized pallets.

5.5.1(c) The design of the framework is to take into consideration a maximum pallet load, transverse and longitudinal forces from ship motion, and forces from loading and unloading. The framework is to transfer these loads to the hull structure.

5.5.1(d) Expansion and contraction of the framework:

Due to the cold temperature in the cargo hold, the framework is likely to contract and expand. Therefore, sufficient tolerances are to be provided in the framework to ensure satisfactory operation of the system.

5.5.1(e) Prevention of distortion of the framework:

The framework is to be fitted as to be free of hull stresses, and is to be sufficiently flexible to tolerate movement in the hull due to the ship's motion without causing permanent distortion.

5.5.2 Ship Motions and Forces on the Pallets

5.5.2(a) The Owner is to state the maximum pallet weight for the system design.

5.5.2(b) The dynamic forces associated with the worst roll, pitch and heave motions for the particular loading condition of the vessel are to be used for the design of a guide framework and the lashing arrangements for the constraint of the pallets.

5.5.2(c) Where detailed studies of long term ship motion response to irregular seas are not submitted, suitable empirical formulae may be used for calculating the dynamic forces such as those given in 4.3 of the ABS *Guide for Certification of Container Securing Systems*.

5.5.2(d) In using empirical formulae, the transverse metacentric height value used, i.e. the *GM*, is to be that calculated for the worst service condition.

5.5.3 Permissible Stresses in the Guide Framework

5.5.3(a) The permissible stresses, based upon the minimum yield of the material, are given as follows:

Normal stress = $0.80\ Y$

Shear stress = $0.53\ Y$

where Y is the minimum yield strength of the material.

For higher strength steels, Y is not to be taken as greater than 72% of the specified minimum tensile strength.

5.5.3(b) Steel Grades for areas exposed to low temperature are to be in accordance with 6-2-3/5.9.

5.5.3(c) Temperature lower than -30°C (-22°F) will be the subject of special consideration.

5.7 Lifting Gear Requirements

5.7.1 Recognized Standards for Lifting Gear

As an alternative to the requirements of this section, compliance with recognized design standards appropriate to the construction and service will be specially considered. The plans and the accompanying calculations for approval are to be in accordance with the standard used.

Where cargo handling cranes are fitted to the vessel as part of the automatic pallet loading/unloading system, the requirements of Chapter 2, "Guide for Certification of Cranes" of the ABS *Guide for Certification of Lifting Appliances* are applicable in addition to the requirements given in 6-2-4/5.11 to 6-2-4/5.23.

5.7.2 Loading Conditions

Typical loads to be submitted and considered in the analysis are:

- Dead and live loads
- Dynamic loads
- Loads due to wind (pallet movement ship to shore)
- Loads due to list and/or trim.

5.7.3 Allowable Stresses

5.7.3(a) The structural components are to be designed to the allowable stresses resulting from the coefficients given in 2-2/Table 1 of Chapter 2, "Guide for Certification of Cranes" of the ABS *Guide for Certification of Lifting Appliances*.

5.7.3(b) For wire rope, the Factor of Safety, based on the maximum load imposed on the wire by the safe working load of the lifting device and the breaking strength of the wire rope is to be as follows:

$FOS = 5.0$ based on breaking load of the wire.

5.7.4 Materials and Welding

Materials and welding are to be in compliance with the requirements given in Section 3 of Chapter 2, "Guide for Certification of Cranes" of the ABS *Guide for Certification of Lifting Appliances*.

5.7.5 Wire Rope

The construction of the wire rope is to comply with a recognized standard such as API Spec 9A.

5.7.6 Stowing and Securing

Means are to be provided for safely stowing and securing the lifting gear when not in use while the vessel is on route.

5.9 Deck Houses

Strengthening may be required for the foundations of any lifting appliances that are fixed to the deck house.

The protection of the deck openings is to be in accordance with the requirements of Section 3-2-15.

5.11 Controls

5.11.1

All loading and unloading operations are to be controlled and monitored from a single control station.

5.11.2

Controls are to be provided for the safe operation of the pallet loading/unloading system. These controls are to be clearly marked to show their functions. Energizing the power unit at a location other than the cargo control station is not to set the gear in motion.

5.11.3

Fail safe arrangements are to be provided.

5.11.4

A safe emergency control position is to be provided.

5.11.5

The system is to be provided with adequate back-up arrangements to enable operation in the event of a component failure. Where, due to the design of the system, provisions for a standby system is impracticable, necessary spares are to carried onboard which would enable rectification of a fault and the ability to resume operation.

5.11.6

A key operated switch or other suitable device to prevent unauthorized operation is to be fitted to the control panel of each pallet loading system. Where the equipment in the pallet handling system needs to be operated manually, means are to be provided to enable this operation during commissioning, fault finding and other similar work.

5.11.7

Monitoring is to indicate the system operational status (operating or not operating), availability of power, overload alarm, air pressure, hydraulic pressure, electrical power or current, motor running and motor overload and brake mechanism engagement, as necessary.

5.11.8

The maximum safe working load is to be conspicuously posted near the controls and visible to the operator.

5.13 Emergency Stop Equipment

5.13.1

Emergency stopping equipment is to be provided to stop the pallet handling system without creating additional risks of hazard. The means for the emergency stop are to be located at each control position.

5.13.2

Starting of the pallet handling system is to be possible from the control station after the emergency stopping device has been reset at the location where it was actuated.

5.13.3

Remote emergency shutdown of power units is to be provided outside of the space where they are located, such that they may be stopped in the event of fire or other emergency. Means for local emergency shutdown is also to be provided.

5.15 Hoist Units/Elevators

5.15.1 Braking System

All hoist units/elevators are to be equipped with effective brakes or other equivalent devices capable of stopping the movements of the hoist unit/elevator with its proof load safely at its rated speed and maintaining it in its stopped position. Brakes are to be applied automatically when the power supply is interrupted.

5.15.2 Limitation of the Lifting and Lowering Movement

In order that lifting and lowering movements are stopped without undue shocks, upper and lower limit stops are to be used to define the extent of the vertical travel and the following are to be provided:

5.15.2(a) Arrangements for initiating a controlled stop towards the upper and lower limits when variable or multi-step drives are used.

5.15.2(b) Control devices which prohibit incorrect hoist direction at the hoist travel limits.

5.15.2(c) Ultimate limit switches which in an emergency shall disconnect the main current on all poles via a main contactor to the hoist. The actuators of these switches are to be independent of other switches.

5.15.3 Overload Protection

Where the mass of the load is not controlled prior to reaching the pallet handling equipment, it is to be equipped with an overload protection system.

5.15.4 Rope or Chains

Hoist units/elevators using ropes or chains are to be equipped with a device to identify a slack rope or chain condition which when actuated stops all operational movement of the hoist unit(s)/elevator(s). Provision is to be made to prevent the restarting of the hoist unit(s)/elevator(s) until the fault has been cleared by an authorized person.

5.15.5 Suspension Elements

Means are to be provided to equalize the tension of the suspension elements where more than one element is fixed to one point, and their position is to be monitored.

5.15.6 Hydraulic Drives

5.15.6(a) Where a part of the lifting unit enters the racks, the system is to be so designed that unintentional lowering of the lift unit does not occur even in the event of a failure of the hydraulic system. This does not apply for leakage at the cylinder.

5.15.6(b) For auxiliary hoist units operated by cylinders directly connected to the lifting carriage or forks, valves are to be fitted to prevent uncontrolled lowering in case of pipe or hose failure.

5.15.6(c) The switches specified in 6-2-4/5.15.2 are not necessary if a cushion cylinder is used to prevent excessive stress.

5.17 Traveling Units/Conveyors

5.17.1 Braking System

5.17.1(a) The traveling unit/conveyor is to be capable of being decelerated and stopped safely from the rated speed with the rated load without undue shocks during normal operation and in case of emergency (e.g., over speed) by the following means:

- An electrical or mechanical braking system for normal operation.
- A mechanical braking system to operate automatically in the event of the power supply being interrupted in any way and to act as a parking brake.

5.17.1(b) An additional braking system is to be fitted and is to operate automatically in the event of failure of the main braking system

The additional braking system is to be capable of operating even in the event of the failure of a gear in the travel unit/conveyor.

5.17.2 Speed Reduction System

Automatic speed reduction is to be provided in addition to that required in 6-2-4/5.17.1 when a lower speed is required for reasons of safety.

Function of this system is to be automatically monitored. In case of failure, the machine is to be stopped automatically.

5.17.3 Limitation of Travel

The following means are to be provided to stop the pallet handler safely at the limit of travel:

5.17.3(a) Suitable buffers or other equivalent devices.

5.17.3(b) An operational device in the control circuit to interrupt the power supply to the drive unit before the pallet handler contacts the buffers. This device is not required if the buffer is designed for continual use and is automatically monitored for return to its initial position.

5.17.3(c) In addition, an ultimate limit switch which disconnects in an emergency the mains current on all poles via the main contactor. The actuator of this switch is to be independent of other switches.

5.17.3(d) Means to prevent collisions which may result in injury to personnel or damage to the pallet handler if more than one machine is working on the same rail.

5.17.4 Anti-Derailment Devices

5.17.4(a) A device to prevent derailment (e.g., profile plate around the head of the rail) is to be fitted on the pallet handler which is to be effective in the case of failure of travel wheels or guide rollers.

5.17.4(b) Rail junctions are to include suitable interlocking devices to prevent derailment.

5.17.4(c) Rail sweeps are to be provided in front of travel wheels and guide rollers.

5.17.4(d) Means are to be provided to prevent the pallet handler from dropping more 10 mm (0.40 in.) if a travel wheel or axle fails.

5.17.5 Stability

The machine and the rails are to be designed and built in such a way that the machine will not overturn even during operation of the safety devices.

5.19 Load Handling Devices

5.19.1 Load Stability

The load handling device (e.g., forks or platforms) is to be constructed in such a way that every part of the specified load will remain in a stable position during normal operation.

5.19.2 End Stop

All movements are to be limited by mechanical means. If striking the end stops can create undue stress in the drive system, limiting devices are to be provided in the control circuit.

5.19.3 Limitation of Forces

The drive unit for extending the load handling devices is to be fitted with a friction clutch or other device to limit the drive force to minimize the risk of damage to the pallet handler or associated storage equipment and injury to persons. The racking supplier is to be advised of the resulting forces.

5.19.4 Rotating Devices

To restrain the load handling device when stationary, the drive unit for a rotating load handling device is to be fitted with a braking system or a gear which is self-sustaining in all modes of operation (e.g., an appropriate worm gear).

5.19.5 Interlocks

5.19.5(a) Interlocks are to be provided which only allow lateral movement of the load handling device when the pallet handler has stopped. With the load handling devices extended, lifting and traveling movements are only to be possible at the slow speed intended for that purpose.

Interlocks such as position sensors for forks or load are to be provided which prevent accidental contact of the load or load handling device with racks or other objects.

5.19.5(b) Means are to be provided to prevent loads being moved into occupied positions (e.g., by aperture occupied sensors).

5.19.6 Auxiliary Handling Equipment

Auxiliary lifting and pulling devices are to be built in such a way that the load cannot be moved into or over the operator position and in such a way that the operator is protected against falling parts of the load.

5.19.7 Load Position Monitoring

The load is to be checked for correct positioning on the load handling device before lift or travel movements take place.

5.19.8 Satellite Vehicles

5.19.8(a) Satellite vehicles are to comply with the requirements in 6-2-4/5.17.1 to 6-2-4/5.17.3 and 6-2-4/5.19.1 to 6-2-4/5.19.3.

5.19.8(b) The correct position of the satellite vehicle on the lifting carriage is to be monitored.

5.21 Electrical

5.21.1 General

5.21.1(a) Except as noted herein, compliance with applicable subsections of Part 4, Chapter 8 is required.

5.21.1(b) Electrical equipment in cargo holds is to be in accordance with 4-8-3/Table 2.

5.21.1(c) Design and construction of motors is to be generally in accordance with Part 4, Chapter 8, except that specific service such as the low temperature in a refrigerated cargo hold environment is to be taken into consideration. Accordingly, the operating profile and evidence of the suitability of motors is to be submitted for review. However, motors need not be inspected at the plant of the manufacturer, but will be accepted subject to satisfactory performance witnessed by the Surveyor after installation.

5.21.2 Traveling Cables

5.21.2(a) Traveling cables for power supply, control and communication are to have flame retardant and moisture-resistant outer covers and are to be of a flexible type constructed to an acceptable recognized standard or specification for this service. Further, the traveling cables are to be protected against damage.

5.21.2(b) Where power supply is through arrangements other than cables, such as bus and brushes, rail, etc., the material used is to be suitable for the intended locations, and means are to be provided to protect against accidental contact by personnel during service.

5.21.3 Main Isolator or Disconnecting Switch

5.21.3(a) The power supply for all equipment in an individual area is to be provided with a main isolator or disconnecting switch which is easily and safely accessible, clearly marked for its purpose and safeguarded against unauthorized switching-on by means of padlocks or other similar suitable devices.

5.21.3(b) Where a maintenance or repair area is provided for the main area isolator or disconnecting switch, it is to be possible to interrupt the power supply in the same way as required in 6-2-4/5.21.3(a).

5.21.4 Unintended Connection

The making of unintended connection between a live and disconnected supply line (e.g. by double current collectors) by the storage and retrieval pallet handling equipment or transfer device is to be prevented.

5.23 Piping Arrangements

Hydraulic piping/equipment is to be in accordance with Part 4, Chapter 6.

7 Automatic or Semi-Automatic Side Loading System ✠ ASLS or ✠ SASLS

7.1

In order to receive the ✠ **ASLS** or ✠ **SASLS** notation, a side loading/unloading system is to comply with the requirements in this paragraph in addition to the applicable requirements in 6-2-4/5.7 to 6-2-4/5.17, 6-2-4/5.21 and 6-2-4/5.23.

7.3

A side loading/unloading system is to enable access through the vessel's side shell for the transference of cargo during loading and unloading operation. These operations may be automatically controlled such that the cargo is conveyed via a hoist unit (such as elevator) onto a preselected deck or to the quay.

7.5

Each cargo deck is to be provided with flashing light to give warning when the hoist units are moving. The lights are to be located adjacent to the hoist unit areas.

7.7

The control station is to be provided with the means for maintaining a constant surveillance of the hoist units' access or each deck.

7.9

Local control of the traveling units (such as conveyors) will be permitted at each cargo deck level. The control functions are to be limited to loading and unloading operations for the specific deck.

7.11

An interior communication system is to be provided between the control station, each cargo deck level and the loading platform.

7.13

Emergency stop buttons are to be provided at the each cargo deck level and the loading platform.

7.15

A portable hand pump unit is to be provided to enable emergency operation of the securing system.

7.17 Materials and Welding

7.17.1

Structural materials are to be suitable for the intended service conditions. Materials are to be certified by the mill and verified by an ABS Surveyor.

7.17.2

In general, welding may be in accordance with the latest edition of ANSI/AWS D1.1, Structural Welding Code Steel, or other recognized codes. For Nondestructive Testing (NDT) of welds, the inspection is to be in accordance with the ABS *Guide for Nondestructive Inspection of Hull Welds*, or other recognized codes.

7.19 Loading Conditions

Typical loads to be submitted and considered in the analysis are:

- Dead and live loads
- Loads due to list and/or trim

7.21 Strength Criteria for the Platforms of the Cargo Elevator

7.21.1

Free end supports are assumed for beams and girders unless ends are effectively fixed.

7.21.2

For beams and girders:

Maximum Allowable Bending Stress $= 0.55 \times F_y$

Maximum Allowable Shear Stress $= 0.40 \times F_y$

Maximum Allowable Bearing Stress $= 0.80 \times F_y$

Maximum Allowable resisting tearing failure $= 0.48 \times F_y$

where F_y is the specified minimum yield strength of the material.

7.23 Foundations and Supporting Structure

Detailed drawings of the foundation and supporting structure on which the elevator or moving platforms are to be installed are to be submitted. The applicable strength criteria for the foundation structure is as follows:

Maximum Allowable Bending Stress $= 0.55 \times F_y$

Maximum Allowable Shear Stress $= 0.40 \times F_y$

where F_y is the specified minimum yield strength of the material

7.25 Wire Rope

7.25.1

The construction of the wire rope is to comply with a recognized standard such as API Spec 9A.

7.25.2

The Factor of Safety, based on the maximum load imposed on the wire by the safe working load of the elevator, is to be as follows:

$FOS = 5.0$ based on breaking load of the wire.

7.27 Stowing and Securing

Means are to be provided for safely stowing and securing the lifting gear when not in use while the vessel is on route.

9 Testing for ✠ APLUS and ✠ ASLS or ✠ SASLS Notations

9.1

The pallet handling gear is to be surveyed at the manufacturer's plant during construction. In-plant surveys during construction are required to the extent necessary for the Surveyor to determine that the details, material, welding and workmanship are acceptable to ABS and are in accordance with the approved drawings.

9.3

During the initial survey onboard the vessel, the original proof testing and an examination are to be conducted.

9.5

The pallet handling gear is to be tested onboard to the following proof load using movable known weights:

Proof load = 1.25 × SWL (Safe Working Load)

9.7

The proof load test is to include hoisting and lowering of the equipment and testing of fail safe and limiting devices. After being tested, the equipment is to be examined to ensure that no part has been damaged or permanently deformed by the test.

9.9

The operation of all brakes and fail-safe devices are to be demonstrated under simulated loss of power conditions to the satisfaction of the Surveyor.

9.11

Satisfactory operation of the pallet handling system, together with the controls, is to be demonstrated after installation onboard to the satisfaction of the attending Surveyors.

PART
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CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

SECTION 5 Refrigerated Cargo Spaces

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 General

All refrigerated cargo spaces and air cooler rooms are to have access doors, hatches and ladders arranged for easy access and escape.

3 Design Considerations

3.1

Where cargo spaces are intended to carry palletized cargo, the minimum clear height in tween deck cargo spaces is to be consistent throughout to accommodate pallets of a height specified by the Owners/builders and is to include a minimum air gap above the pallets of 100 mm (4 in.) for air circulation.

3.3

For vessels intended to operate in regions where ambient temperatures are expected to be lower than the cargo space temperatures, the owner or builder may install cargo space heating or other means for maintaining cargo space transport temperatures. These systems should have appropriate controls for maintaining the desired temperature.

3.5

When using either fork lift trucks or pallet trucks, the grating, insulation, lining and spar deck planking is to be of sufficient strength to support the weight of a fully loaded truck carrying the heaviest load envisaged during normal loading and unloading. This is to be demonstrated in accordance with tests specified in 6-2-5/17.5.

3.7 Corrosion and Protection

3.7.1 Hull Structure

3.7.1(a) All steel surfaces are to be cleaned of grease and other organic contaminants and are to be abrasive-blasted to near white finish (SSPC-SP-10, NACE No.2, SWEDISH SA 2.5) or to an alternative finish in accordance with the paint manufacturers specification prior to coating. This may be done before erection and welding, in which case special attention is to be given to the preparation of the welded areas.

3.7.1(b) Steel surfaces of refrigerated cargo spaces, behind insulation and including the inside of hatch coamings are to be coated to a minimum dry film thickness of 150 microns (6 mils). Steel work and fittings, which are to be covered with insulation, are to be similarly cleaned and then coated to prevent corrosion. Where polyurethane foam is applied directly to the steel structure and bulkheads, the surfaces are to be prepared to ensure proper adhesion and resistance to corrosion.

3.7.1(c) Openings in the refrigerated cargo spaces such as the bilge limbers and plugs and other openings to these spaces such as the hatch covers and access doors are to be constructed of moisture resistant material or covered with such material.

3.7.1(d) Where the tank top or bulkhead of an oil storage tank forms part of the refrigerated cargo space walls, the surface of the tank plating is to be coated with an oil impervious coating.

3.7.2 Fittings and Fixtures

Steel bolts, nuts, screws, washers, hangers and other similar fixtures which support or secure insulation, pipes, meat rails, etc., are to be protected against corrosion by means of galvanizing or other equally effective methods approved by ABS.

3.7.3 Pipes, Ducts and Drip Trays

3.7.3(a) Refrigerant and brine pipes in the refrigerated cargo spaces are to have corrosion protection in accordance with 6-2-6/23.3.

3.7.3(b) All steel ducts and pipes passing through the refrigerated cargo spaces are to be protected against corrosion prior to the application and installation of the insulation.

3.7.3(c) Steel drip trays provided under air coolers and vertical cooling grids are to be galvanized or epoxy coated. Materials other than steel such as plastic or flake glass may be used for the construction of the drip trays, provided the material used is suitable for the intended application and has been approved by ABS.

5 Insulation

5.1

The insulation arrangement, materials, construction and installation are to be in accordance with the approved plans and to the satisfaction of the Surveyors.

5.3

Where the insulation is provided in the form of prefabricated insulating panels, the panels are to be approved by ABS. Inspections by the Surveyors are required during the manufacture of these panels.

5.5

When requested, the manufacturing of the panels referred to in 6-2-5/5.3 may be accepted under the quality assurance program.

5.7 Types

Rockwool, polyurethane, styrofoam, glass fiber or equivalent material may be used for insulation purposes.

5.9 Properties

5.9.1

All insulation material used in the refrigerated cargo spaces is to be of a type which does not produce or absorb paint.

5.9.2

Organic foam is to be fire retardant as established by a recognized fire test procedure such as DIN 4102.B2. Test certificates in this regard issued by independent testing laboratories are to be submitted for review.

5.9.3

The insulation material is to be resilient and should not distort or deform due to the temperatures which will be encountered in service. It should also be capable of withstanding shipboard vibrations likely to occur during normal operating conditions.

5.11 Temperature Gradient Calculation

5.11.1

The thickness of insulation over all surfaces is to be in accordance with approved specifications and plans.

5.11.2

Thermal bridges associated with fittings for securing the panels and moisture barriers around the open edges are to be accounted for in the calculations.

5.11.3

Where machinery spaces and other such spaces fitted with heating arrangements such as fuel tanks, etc., are situated adjacent to the refrigerated cargo spaces, the heat transfer calculations are to take this into consideration.

5.13 Installation

5.13.1

The insulation is to be efficiently packed and securely fastened.

5.13.2

Insulation slabs or blocks, where used, are to have the joints staggered and butted as close as possible. If several layers of insulation blocks are employed, these are also to be installed in a similar manner. Any unavoidable gaps between the joints and crevices are to be filled with suitable insulating material.

5.13.3

Panels are to be of sufficient mechanical strength to withstand, without damage, loads due to over- or underpressure of the refrigerated cargo spaces resulting from the defrosting of coolers or rapid cooling of the refrigerated cargo space. Alternatively, suitable pressure equalizing devices are to be fitted.

5.13.4

During the installation of the prefabricated insulation panels, it is to be ensured that the panels are butted together such that all joints along the edges and the corners are sealed at the outer and inner sides to form a vapor barrier using an approved sealant. The same method is to be employed at floors, ceiling intersections and the vertical bulkheads.

5.13.5

Provisions are to be made in the design for an effective moisture barrier at the open edges of the panels at the footing, corner intersections, openings for doorways, etc.

5.13.6

Decks, partitions and other structural members which extend into refrigerated cargo spaces from the ship side, machinery spaces or other such non-refrigerated adjacent spaces are to be effectively insulated over a length of at least 1 m (3.3 ft) into the refrigerated cargo space unless temperature gradient calculations prove less carry-over is sufficient.

5.15 Lining

5.15.1

The insulation is to be protected from water and water vapor by suitable lining material such as marine plywood (coated), metallic sheet or other similar material which is impervious to water.

5.15.2

The insulation lining referred to in 6-2-5/5.17.1 is to be installed in such a way as not to allow water to penetrate into the insulation during hosing down of the chambers.

5.15.3

Lining, cooler room screens and structures supporting these are to be of sufficient strength to withstand the loads imposed by either the refrigerated or general cargo in transit

5.15.4

Where plywood is used it is to be treated against fungi, other microorganisms and dampness.

5.15.5

All timber which is embedded in insulation is to be impregnated under pressure with odorless preservative. All sawn ends and bolt holes to be treated in situ.

5.15.6

In order to protect the lining against damage from forklift trucks or pallet jacks, a metallic plate of minimum height 500 mm (1.6 ft) and thickness of 6 mm (0.24 in.) is to be provided at deck level. Alternative heights and thicknesses proposed by the Owner/builder will be specially considered. Other materials such as glass reinforced plastics may be used, provided it is demonstrated to the satisfaction of ABS to be of suitable strength and durability.

5.17 Insulation of Pipes, Ducts and Vent Trunks

5.17.1

To prevent freezing, vent, sounding, overflow and water pipes are to be insulated from cold surfaces such as the bulkheads and decks and installed so that contact with the warmer surfaces such as the vessel side is maintained as much as possible. Where this is impracticable, heat tracing of these pipes is to be fitted.

5.17.2

Ducts and pipes passing through refrigerated cargo spaces are to be efficiently insulated.

5.17.3

Where thermometer tubes are partially inserted into the space being monitored, the portion of the tube external to that space is to be efficiently insulated.

5.19 Penetration of Insulation

5.19.1

Plugs provided for access to manhole covers, bilge suction wells, drains, etc. are to be insulated in accordance with approved plans.

5.19.2

To prevent seepage of water into the tank top insulation, openings for manholes and bilge covers are to be fitted with liquid tight steel coamings. The height of the coaming is not to be less than the insulation. A sealant may be applied at the edges to prevent seepage into the insulation.

5.19.3

Ducts, pipes and cable penetrations are to be made airtight.

5.19.4

Provisions are to be made in the installation of the insulation to enable inspection during the periodical surveys of the bilge suction pipes, vent and sounding pipes and other similar pipes situated behind the insulation. This may be achieved by installing removable insulation panels or other methods approved by ABS.

7 Stowage and Side Shoring

7.1

Provisions are to be made to ensure the circulation of air between the cargo and the insulation lining surfaces.

7.3

Cooling grids located on vertical surfaces are to be protected by dunnage ribs.

7.5

Side shoring is to be of sufficient strength to withstand the dynamic loads imposed by palletized cargo in transit.

9 Air Circulation and Ventilation

9.1

The required air circulation and fresh air ventilation rates are to be based upon the air volume of empty refrigerated cargo spaces.

9.3

The design of the air circulation system in refrigerated cargo spaces intended for the carriage of fruit is to ensure a sufficient flow of chilled air throughout all the stow in the loaded condition.

9.5

For refrigerated cargo spaces fitted with coolers with forced air circulation, the quantity of circulating air for each refrigerated cargo space is to be based on the nature of the cargo and the design temperature, but shall not be less than 30 air changes per hour. Lower air circulation rates for frozen cargoes will be considered.

9.7

For fruit carriers the cooling fans are to have the capability of running at a minimum of two speeds such that the air circulation rates in the refrigerated holds can be maintained at not less than 45 and 90 air changes per hour.

9.9

Refrigerated cargo spaces intended for carriage of fruit must also be provided with a fresh air mechanical ventilation system providing at least 2 air changes per hour.

9.11

Air circulation and fresh air ventilation rates lower than those stated in 6-2-5/9.5, 6-2-5/9.7 and 6-2-5/9.9, will be considered subject to the submission of an assessment of the heat to be removed, nature of cargo, etc.

9.13

Each refrigerated cargo space intended for the carriage of fruit is to be provided with its own separate inlet and exhaust vent. The position of the air inlet is to be selected to minimize the possibility of contaminated air entering into any refrigerated cargo space.

9.15

For details of the ventilation when the vessel is engaged in carriage of cargoes other than refrigerated cargoes, reference is to be made to the requirements contained elsewhere in the Rules.

11 Ducts, Gratings and Spar Decks

11.1

Cooling air from the fan unit is to be evenly distributed at the bottom of the refrigerated cargo spaces.

11.3

The height of the gratings and size and number of ventilation holes are to be appropriate for the air circulation requirements.

11.5

The size and number of ventilation holes in the spar deck planking are to be appropriate for the air circulation requirements.

11.7

Suitable arrangements are to be made to allow for ease of lifting of the gratings to enable cleaning and maintenance of the deck beneath.

11.9

In each refrigerated cargo space, the grating and associated supports directly underneath the hatch opening and 600 mm (2.0 ft) beyond are to be designed to withstand impact during loading. Increased grating thickness and/or reduced spacing of the supports will be considered, provided air circulation is not adversely affected. The protection of insulation in gratingless cargo spaces is to be no less effective.

13 Bilge and Drainage Arrangements

13.1

The bilge system for cargo spaces is to be in accordance with 4-6-4/5.5.

13.3

Cooling grids fitted vertically on the refrigerated cargo space sides and air coolers are to be provided with drip trays and drain pipes arranged as follows:

13.3.1

Drain pipes are to be sized to allow drainage without overflowing of the drip trays during defrosting operations, taking into consideration the vessel's motion.

13.3.2

Drainage openings in the drip trays are to be easily accessible for cleaning.

13.3.3

Drain pipes are to have flanged connections near the outlets to allow cleaning in the event of blockage.

13.3.4

Trace heating of the drain pipes and drip trays is to be provided when carrying frozen cargo.

13.5

All refrigerated cargo spaces are to have ample continuous drainage.

13.7

Provision is to be made to prevent air and water from leaking into adjacent refrigerated cargo spaces.

13.9

To prevent air from leaking into adjacent refrigerated cargo spaces, open ended pipes such as drains from each deck space or the drip trays from these spaces are to be fitted with liquid seal traps or non-return valves. These requirements are also applicable to drains underneath the coolers.

13.11

When drains from separate refrigerated cargo spaces join in a common main, the branch lines are to be provided with liquid seal traps to prevent air from leaking into adjacent refrigerated cargo spaces. In addition, branch lines from lower spaces are to be provided with non-return valves to prevent flow of water from one compartment to another.

13.13

Liquid seal traps located in areas subject to freezing are to be filled with brine and are to be easily accessible for maintenance purposes.

13.15

Drains from other spaces are not to lead to the bilges of refrigerated cargo spaces.

13.17

Bilge wells where drain pipes are led, and connections to the main bilge system are to be separated from refrigerated cargo spaces by air tight moisture resistant divisions.

15 Pipes Passing Through Refrigerated Cargo Spaces

15.1

Air, sounding and tank filling pipes which pass through insulated spaces are to be arranged as close to the shell and bulkhead structure as possible. Flanged joints are to be kept to a minimum, and where additional supports are necessary, brackets are to be fitted.

15.3

Steel pipes penetrating the tank top in refrigerated cargo spaces are to have a wall thickness of a heavier grade in way of the insulation and the tank top.

15.5

All sounding pipes passing through refrigerated spaces where the temperature may be below 0°C (32°F) are to have an inside diameter of not less than 65 mm (2.6 in.).

15.7

Sounding pipes for oil tanks are not to terminate in refrigerated cargo spaces or air cooler rooms.

17 Tests and Inspections

17.1

The shipyard is to submit the results of the corrosion resistance coating thickness measurement to the attending Surveyor.

17.3

The Surveyor is to verify the adequacy of the seals and traps for each refrigerated cargo space.

17.5

The test required by 6-2-5/3.5 on the insulation and lining is to be carried out in the presence of the attending Surveyor as follows:

A 4 × 4 m (13 × 13 ft) sample of the cargo floor construction, including insulation, is to be prepared and tested by a fully loaded fork lift truck with its heaviest load envisaged during normal loading and unloading operations being driven and maneuvered over the sample. Where cargo operations will not involve forklift trucks, a similar test using a fully loaded pallet truck is to be performed.

17.7

Insulation thickness on pipes, valves, flanges and fittings is to be examined by the attending Surveyor.

17.9

Sample tests performed by the manufacturer to determine the density of the insulating material are to be presented to the Surveyor for verification that the material complies with the design specification.

17.11

Where insulating foam is intended to be applied directly to the ship's structure, the method of application and the procedure are to be approved prior to commencement of the work.

17.13

For prefabricated panels referred to in 6-2-5/5.3, the insulation material is to be in accordance with 6-2-5/17.9.

17.15

For air distribution tests, refer to 6-2-16/3.1.3.

PART

6

CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

SECTION 6 Refrigeration Machinery

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 General

1.1

The location of the refrigeration units and associated equipment such as pumps, coolers, cooling fans and motors, etc., is to be such that sufficient space is available to allow easy access during maintenance and repair.

1.3

In general, the refrigeration units for cargo refrigeration are to be completely independent of any refrigerating machinery associated with air conditioning plants or provision refrigeration installations. A combined system will be subject to special consideration on an individual basis.

1.5

An effective defrosting system suitable for the service conditions and cargo carried is to be installed.

3 Design Considerations

3.1 Design Pressures

3.1.1

The design pressure is the maximum allowable working pressure at which the system can be used. Relief valves in any part of the system are to be set such that the design pressure is not exceeded.

3.1.2

The system is to be designed such that under all normal operating and standstill conditions the design pressure is not exceeded.

3.1.3

In general, the design pressure on the high pressure side of the system is not to be less than the pressure corresponding to the condensing temperature of the refrigerant used, e.g., saturated pressure at 55°C (130°F) for refrigerants with zero or negligible glide. For zeotropic blends with significant glide, the bubble point pressure is to be used (indicated with an asterisk in 6-2-6/3.1.6).

3.1.4

The design pressure of the low pressure side of the system is not to be less than the pressure corresponding to the evaporating temperature of the refrigerant used at the discharge from the expansion valve, e.g., saturated vapor pressure at 45°C (113°F) for refrigerants with zero or negligible glide. For zeotropic blends with significant glide, the bubble point pressure is to be used (indicated with an asterisk in 6-2-6/3.1.6).

3.1.5

Where the method for defrosting is by means of circulating hot refrigerant gas, the design pressure on the low pressure side is to be the same as that on the high pressure side.

3.1.6

The minimum design pressure for the refrigerants listed is to be as follows:

Refrigerant	*High pressure side bar (kgf/cm², psi)*	*Low pressure side bar (kgf/cm², psi)*
R22	20.5 (20.9, 295)	17.1 (17.4, 250)
R717	22.4 (22.8, 325)	17.9 (18.3, 260)
R134a	13.7 (14.0, 200)	10.5 (10.7, 150)
R404a*	25.0 (25.5, 365)	19.8 (20.2, 285)
R407a*	25.2 (25.7, 365)	19.8 (20.2, 285)
R407b*	26.5 (27.0, 385)	20.9 (21.3, 305)
R407c*	23.9 (24.4, 345)	18.8 (19.2, 275)
R410a	32.8 (33.4, 475)	25.9 (26.4, 285)
R410b	32.5 (33.1, 471)	25.7 (26.2, 375)
R507	25.4 (25.9, 370)	19.9 (20.3, 290)

3.3 Capacity

3.3.1 General

3.3.1(a) At least two refrigeration units are to be provided. The aggregate capacity of the units is to be sufficient to deal adequately with the cargo as received aboard. The ambient conditions for determining the required capacity are to be based on the following conditions:

Sea water temperature	32°C (90°F)
Air temperature	35°C (95°F)
Relative humidity	75%

Where the vessel is intended to operate in regions where the temperature and the relative humidity other than those mentioned above are encountered, alternative conditions will be specially considered upon request from the Owners and/or builders.

3.3.1(b) Capacity of the refrigerating machinery is to be selected taking into account their purpose and service conditions. Where appropriate, allowance is to be made for heat generated by air circulation fans, heat produced by cargo, introduction of fresh air, heat transmission through insulation and heat input from other sources such as insulation, pipes, ducts, tank tops, steel structure, etc.

3.3.1(c) In order to compensate for the deterioration of blown foam insulation over the life of the installation, the calculated transmission heat, based upon the rated insulation performance, is to be increased by 10%, prior to inclusion in the capacity calculations.

3.3.1(d) Where refrigerated spaces are served by independent separate refrigeration units, the capacity of the units will be subject to special consideration.

3.3.2 Fruit Carriers

3.3.2(a) For the purposes of calculations, the aggregate capacity of the refrigeration system is to be such that for all the loaded refrigerated cargo spaces, under the conditions specified in 6-2-6/3.3.1, the return air temperature can simultaneously be reduced to a temperature 2°C (1°F) higher than the required steady state delivery air temperature within 24 to 36 hours.

3.3.2(b) A cool down period greater than that stated in 6-2-6/3.3.2(a) above will be specially considered when the vessel is likely to operate under conditions other than those stated in 6-2-6/3.3.1(a) or an alternative cool down period is agreed between the designers/builders and Owners.

3.3.2(c) In the event that one of the refrigeration units becomes non-operational, the capacity of the remaining unit(s) is to be sufficient to achieve and maintain the required delivered air steady state temperature when operating under the design conditions stated in 6-2-6/3.3.1.

3.3.3 Refrigerated Cargo Vessels other than Fruit Carriers

3.3.3(a) For the purposes of calculations the total aggregate capacity of the refrigeration system is to be such that minimum design temperature in all refrigerated cargo holds can be achieved under maximum loads with ambient conditions, as applicable and as specified in 6-2-6/3.3.1.

3.3.3(b) The capacity of the refrigeration system is to be sufficient to maintain the minimum design temperature in all refrigerated cargo spaces, under the conditions specified in 6-2-6/3.3.1, as applicable, with one of the units in standby condition.

3.3.4 Fish Processing Vessels

The aggregate capacity of the refrigeration system is to be in accordance with 6-2-6/3.3.3.

5 Refrigerants and Secondary Coolants

5.1

Refrigerants listed under 6-2-6/3.1.6 may be used in the refrigeration system of a refrigerated cargo vessel classed with ABS.

5.3

Use of other refrigerants will be permitted by ABS, subject to approval of the chemical properties, including toxicity, flammability and compliance with the requirements of 6-2-1/11.5.

5.5

Where it is intended to replace refrigerant in a refrigeration system onboard existing vessels under ABS Class, their use will be subject to the following:

5.5.1

Where substitute refrigerant operates at pressures greater than the system's original design pressure, details are required to be submitted to show the method used, such as calculations followed by hydrostatic tests to ensure the integrity of the existing system to withstand higher pressures under all operating and stand still conditions.

5.5.2

For those substitute refrigerants which incorporate a flammable component, precautions are to be taken to ensure that air cannot enter into the system.

5.5.3

The lubricating oil is to be soluble with the substitute refrigerant.

5.5.4

For those lubricating oils which are hygroscopic, the refrigeration system is to be effectively dehydrated before charging.

5.5.5

Poly-glycol lubricating oils should not be used in systems which previously contained chlorinated refrigerants and mineral oils.

5.5.6

The thermal stability of the lubricating oil is to be compatible with the discharge gas temperature.

5.5.7

The capacity of the pressure relief devices and the diameter and length of the discharge pipes are to comply with 6-2-6/17.15, 6-2-6/17.17 and 6-2-6/17.19.

5.5.8

The substitute refrigerant is to be compatible with the materials used in the existing system.

5.5.9

A refrigerant leakage detection system complying with 6-2-10/9 is to be provided in accordance with 6-2-10/9.1.

5.7

Hydrocarbons such as propane, butane, pentane or other similar flammable products are not permitted to be used as refrigerants in shipboard refrigeration systems.

5.9

The use of CFCs as refrigerants in shipboard refrigeration systems is not permitted by various administrations.

5.11

Solutions of sodium chloride (NaCl), calcium chloride (CaCl), magnesium chloride (MgCl) and water, commonly referred to as brine, may be used as secondary coolant in shipboard refrigeration systems. The use of other substances as secondary refrigerants will be considered, provided the flash point of the substance used is greater than 66°C (150°F).

5.13

Brine concentration is to be maintained to suit the evaporating temperature.

5.15

The refrigerant storage cylinders are to be approved by a nationally recognized agency or other similar authorized body.

7 Materials and Fabrication

7.1

Materials are to comply with the applicable requirements in Part 2, Chapter 3 and Part 4, as applicable.

7.3

Materials used for air coolers are to be corrosion-resistant or, alternatively, protected by galvanizing of the external surfaces exposed to the airflow.

7.5

Ferrous materials for refrigerant piping, valves and fittings with an intended service temperature below -18°C (0°F) are to comply with the requirements of Section 2-3-13, or with other approved specifications, except that:

7.5.1

Impact testing will not be required for austenitic stainless steel.

7.5.2

Impact testing will not be required for nut and bolt materials.

7.5.3

Impact testing will not be required if the intended service temperature is not below -29°C (-20°F), and provided the maximum fiber stress is not more than 40% of the allowable stress indicated in 4-4-1A1/Table 2 or 4-6-2/Table 1.

7.7

Seamless copper piping and seamless red brass piping, manufactured in accordance with the requirements of Section 2-3-16 or Section 2-3-17, and seamless or welded copper-nickel piping will be acceptable without impact testing.

7.9

Material for crankshafts, connecting rods, cylinders and cylinder covers, housings, rotors and rotor casings of reciprocating and rotary compressors, as applicable, is to be in accordance with the applicable requirements of this Section and Part 2, Chapter 3. Materials complying with other recognized standards will be considered.

7.11

Synthetic materials, such as neoprene, chloroprene, etc., may be used for gaskets, seals and packing in halocarbon refrigerant systems. Natural rubber is not to be used for applications in contact with the refrigerant.

7.13

Where the intended service temperature is below -18°C (0°F), ferritic steel plating used for the fabrication of refrigerant liquids receivers or other low-temperature pressure vessels is to be in accordance with 5C-8-6/Table 2. Provisions for exemptions to the toughness testing for low-stress applications in 6-2-6/7.5 may be applied to the receivers and pressure vessels.

7.15

Cast iron pipe is not to be used for refrigerant service.

7.17

The material of pipes, valves and fittings is to be in accordance with Part 2, Chapter 3 and is to be compatible with the refrigerant and, where applicable, the secondary coolant. For service where the fluid is a strong electrolyte such as brine, the materials used within the same system are to be compatible in terms of galvanic potential. In general, fabrication is to be in accordance with 2-4-2/9.5 and the following:

7.17.1 Ammonia System

Piping is to be black steel (non-galvanized). Seamless pipes and welded pipes are acceptable for use in Ammonia systems.

7.17.2 Halocarbon System

Welded or seamless copper, brass or copper-alloy pipes may be used in halocarbon systems. Piping is to be welded or brazed and pipe connections made are to be either welded or through brazed flanges. Soldering is not permitted.

Connections to valves, castings, expansion joints, spool pieces and other similar fittings is to be by welding, brazing or by use of flanges.

Magnesium alloys are not to be used where they would be in contact with any halogenated refrigerants, e.g., R22, R134a, etc.

7.19

Finned piping is acceptable for use in liquid to vapor/gas heat transfer components.

7.21

Materials used for construction of pump components which are exposed to the medium being circulated are to be suitable to withstand the effect of that medium.

9 Location and Access

9.1

The refrigeration machinery may be located in the main/auxiliary machinery spaces or in a separate dedicated space.

9.3

Spaces containing refrigeration machinery and refrigerant storage cylinders are not to have direct access to accommodation spaces. Doors are to open outwards and those not leading directly to the open deck are to be self-closing.

9.5

Refrigerant storage cylinders are to be properly secured and located in the space containing the refrigeration machinery or a dedicated space which is independently, naturally ventilated. Means for closing the vent openings from outside the dedicated space are to be provided.

9.7

Air coolers and fans are to be located in a manner which will enable easy access for the maintenance, repair and replacement of equipment with the refrigerated cargo spaces fully loaded.

11 Ventilation of Refrigeration Machinery Space

11.1

Spaces containing refrigerating machinery are to be ventilated by means of mechanical ventilation. The ventilation is to be able to provide at least 30 air changes per hour.

11.3

The ventilation ducting of spaces containing refrigerating machinery is not to be connected to the ventilation system serving the accommodation spaces, and the ventilation exhaust is to be led to the weather independently from other ventilation ducting.

11.5

The exhaust air ducts are to be air tight and the exhaust outlet is to be so positioned as to prevent recirculation to other enclosed spaces.

11.7

Means are to be provided for stopping the ventilation fans and closing the ventilation openings from outside the refrigerated machinery spaces.

13 Compressors

13.1

The crankcase of trunk piston compressors and rotor casing of rotary compressors are to be designed to withstand a pressure equal to the maximum design pressure of the high pressure side of the system.

13.3

Air-cooled compressors are to be designed for an air temperature of at least 45°C (113°F). Water cooled compressors are to be designed for a water temperature of at least 32°C (90°F)

13.5

Compressors of the positive displacement type over 10 kW (13.4 hp) are to be fitted with a relief valve or a bursting disc so arranged that the discharge is led from the high pressure side to the low pressure side in the event that the discharge valve is inadvertently closed. The capacity of the pressure relief device is to be sufficient to accommodate the discharge from the compressor when operating at full load at the maximum possible suction pressure for the refrigerant used. Alternatively, discharge may be led to deck, provided the outlets are located in accordance with 6-2-6/17.11.

13.7

Compressor vibration resulting from gas pressure pulses and inertia forces is to be taken into account in the compressor design and mounting arrangement. Acceptable mounting arrangements include resilient rubber mounts, springs, etc.

13.9

The compressor is to be equipped with safety devices to automatically stop the compressor in accordance with 6-2-10/Table 1.

13.11

All compressors are to be equipped with gauges in accordance with 6-2-10/Table 1.

15 Pressure Vessels and Heat Exchangers

15.1 General

Pressure vessels and heat exchangers under refrigerant pressure are to be constructed in accordance with Part 4, Chapter 4.

15.3 Oil Recovery Equipment

Oil separators with automatic drains are to be provided upstream of the evaporator. For compressors which have gas inter coolers, oil separators are to also be provided between the low stage discharge and the inter cooler. Arrangements for recovering oil from surge pots are to be provided.

15.5 Refrigerant Filters and Dryers

15.5.1

Filters are to be provided in the liquid line upstream of the expansion valves and in the gas line on the suction side of the compressor.

15.5.2

Where the solubility of water in the refrigerant is low, dryers are to be provided to maintain the water vapor content below the value at which free water will occur in the low pressure side of the system. The dryers are to be located upstream of the expansion valves.

15.7 Liquid Receivers

15.7.1

The refrigerating system is to be provided with a liquid receiver with shut off valves arranged to accept and capable of holding the complete refrigerant charge of the refrigerating units during servicing or repairs. Where each refrigerating unit is fitted with an individual receiver, the capacity is to be sufficient to hold the charge from that unit.

15.7.2

Receivers may be fitted with gauge glasses of the flat glass type having approved self-closing valves at each end. Tubular type gauge glasses will be considered, provided they are fitted with approved self-closing valves at each end and are protected from mechanical damage.

15.9 Expansion Valves

Expansion valves are to be suitable to achieve the required temperature for the refrigerant used.

15.11 Evaporators

Evaporators of the flooded type are to be provided with arrangements for recovering oil.

15.13 Brine Heater

Where arrangements for heating brine are by means of an auxiliary boiler, the capacity of the boiler is to be sufficient to ensure that heating of all refrigerated cargo spaces can be performed simultaneously, while supplying other shipboard consumers under normal operating conditions.

17 Safety Relief Devices

17.1

Each refrigerant system is to be provided with pressure relief devices set to relieve at a pressure not greater than the design pressure. Where relief valves are fitted, they are to be of a type not affected by back pressure.

17.3

Pressure relief devices are not to be provided with means for isolation from the part of the system they are protecting. However, where over pressure protection is by means of dual pressure relief devices, the isolation arrangement described in 6-2-6/17.7 will be acceptable.

17.5

Pressure vessels which contain liquid refrigerant and which may be isolated from the refrigeration system are to be protected by a pressure relief valve or bursting disc set to relieve at a pressure not greater than the design pressure.

17.7

Pressure vessels having an internal gross volume of 0.285 m^3 (10 ft^3) or greater are to use dual pressure relief valves or two bursting discs, or a combination thereof. These devices are to be fitted with a three-way valve to permit maintenance of either of the two relief devices without isolating the other. Where pressure relief is to the low pressure side of the refrigeration system, a single pressure relief valve may be used.

17.9

Sections of piping that can be isolated in a liquid full condition are to be provided with pressure relief valves to protect against excessive pressure due to temperature rise.

17.11

Discharge from pressure relief devices is to be led directly to the weather or the low pressure side of the refrigerant system for subsequent relief to the weather. The discharge outlet from these relief devices is to be led away from ventilation inlets and openings. Prevention against the ingress of water, dirt and debris is to be provided.

17.13

When the discharge from a pressure relief valve is led to the weather, further protection against loss of refrigerant through leakage may be provided by means of leak detectors located between the outlet and the relief valve.

17.15

The minimum required discharge capacity of the pressure relief device, in terms of air flow, for each pressure vessel is to be determined by the following formula:

$$C = f\,D\,L$$

where:

C = *Minimum required discharge capacity of the pressure relief device, in terms of air flow, kg/s (pounds per minute).*

D = *Outside diameter of the pressure vessel, in m (ft).*

L = *Length of the pressure vessel, in m (ft).*

f = *Factor applicable to type of refrigerant. The values for f of the more common refrigerants are listed in the following table:*

Refrigerant	f *metric (US units)*	f^* *metric (US units)*
R22	*0.131 (1.6)*	
R134a	*0.131 (1.6)*	
R404a	*0.18 (2.2)*	
R407a		*0.163 (2.0)*
R407b		*0.203 (2.5)*
R407c	*0.131 (1.6)*	
R410a		*0.163 (2.0)*
R410b	*0.197 (2.4)*	
R507		*0.203 (2.5)*
R717	*0.041 (0.5)*	

f^* *- Factor proposed and under consideration.*

17.17

The internal diameter of the discharge pipe from the pressure relief device is not to be less than the outlet of that device. The internal diameter of a common discharge line serving two or more pressure relief devices which may discharge simultaneously is to be based upon the sum of their outlet areas with due allowance for the pressure drop in all downstream sections.

17.19

The maximum length of the discharge pipe serving a pressure relief device is to be determined by the following formula:

$$L = \frac{FP^2 d^5}{C_r^2}$$

where:

F = *1.95 × 10^-10^ (1.88 × 10^-10^, 0.5625)*

L = *Length of the discharge pipe, m (ft)*

P = *{Set pressure of relief device × 1.1} + 1.0 bar (1.0 kgf/cm^2, 14.7 psi)*

d = *Internal diameter of discharge pipe, mm (in.)*

C_r = *Rated discharge capacity of pressure relief device, in terms of air flow, kg/s (pounds per minute).*

19 Air Coolers

19.1

The design of the air cooler coils/cooling grids is to be based upon the total heat load and service conditions specified in 6-2-6/3.3, and the air circulation rates specified in 6-2-5/9.

19.3

To minimize the dehydration of fruit cargo and the frosting of air cooler coils/cooling grids, the refrigeration system is to be designed such that under steady state conditions, the inlet temperature of the refrigerant or secondary coolant circulating in the air cooler coils/cooling grids is not greater than 5°C (9°F) below the delivery air temperature for fruit cargoes and 10°C (18°F) below the return air temperature for frozen cargoes.

19.5

For each refrigerated cargo space over 300 m^3 (10,600 ft^3) cooled by air coolers, the air cooler coils are to be divided into at least two independent sections so that any one of them may be isolated without affecting the operation of the others. Alternatively, at least two independent air coolers are to be fitted.

19.7

A defrosting system is to be installed.

21 Cooling Grids

For each refrigerated cargo space over 300 m^3 (10,600 ft^3) cooled by cooling grids, the cooling grids are to consist of at least two independent sections so that any one of them may be isolated without affecting operation of the others.

23 Piping Systems

23.1 Design Considerations

23.1.1

Pipes, valves and fittings are to be generally in accordance with the requirements of Part 4, Chapter 6.

23.1.2

Refrigerant piping is to be designed to resist collapse when subjected to the drying procedure described in 6-2-16/1.3.

23.1.3

Where liquid refrigerant is being pumped near its saturation pressure, the refrigerant pump is to have a sufficient net liquid column above the pump centerline to provide the pressure required to cause liquid flow into the pump suction without flashing.

23.1.4

Arrangements for preventing slugs of oil or liquid refrigerant entering the compressor suction are to be provided. Any liquid collected may be returned to the system by satisfactory means.

23.1.5

Bulkhead and deck penetrations of refrigerant/secondary coolant pipes whose working temperature is below the normal ambient temperature are to be constructed so that the pipes do not come in direct contact with the steel members of the ship's structure.

23.1.6

Where liquid refrigerant is circulated through the system by pumps, the system is to be provided with a dedicated, readily interchangeable standby pump capable of replacing, without reduction in capacity, any operating pump.

23.1.7

Where secondary coolant is circulated through the system by pumps, the system is to be provided with a dedicated, readily interchangeable standby pump capable of replacing, without reduction in capacity, any operating pump.

23.1.8

Brine tanks are preferably to be of a closed type and have ventilating pipes led to the weather away from ventilation inlets and openings to accommodation spaces. Wire gauze is to be fitted to the ventilating pipe outlets.

23.1.9

Where open type brine tanks are installed, the compartments in which they are located are to be adequately ventilated to prevent accumulation of objectionable vapor.

23.1.10

Where necessary, the refrigeration units may be interconnected on the discharge and/or suction side to facilitate operation of the individual compressors with each condenser and, where applicable, with each brine cooler.

23.3 Corrosion Prevention and Insulation

23.3.1

Refrigerant/secondary coolant pipes within refrigerated chambers or embedded in the insulation and all refrigerant/secondary coolant pipes with working temperatures below ambient temperature are to be protected externally against corrosion. Steel pipes are to be galvanized on the outside or protected against corrosion by other equally effective methods approved by ABS.

23.3.2

Brine pipes are not to be galvanized internally.

23.3.3

Pipes welded or threaded in place, such as at pipe and flange connections, are to have their corrosion protection reinstated by an approved method.

23.3.4

All pipes indicated in 6-2-6/23.3 as well as valves and fittings whose working temperature is below the normal ambient temperature are to be effectively insulated. The insulation is to be sufficiently thick to prevent the formation of moisture on the pipe surface at a relative humidity of 90%. The insulation is to be free of discontinuities and must be protected where there is a danger of damage, and its final layer must be resistant to moisture penetration.

23.5 Valves and Fittings

23.5.1

Gate valves, ball valves and plug cocks are not to be fitted in the liquid refrigerant circuit unless consideration is given to the expansion of liquid trapped in the valve cavities when the valve or cock is closed.

23.5.2

Valves in the refrigerant circuit are to be fitted with removable sealing caps or other alternative means to retain any leakage that may pass through valve glands and seals. However, remote controlled valves or manual valves subject to regular operation, such as manifold valves, will be subject to special consideration and may be accepted without the removable caps.

23.5.3

Filters, strainers and refrigerant dryers are to be provided with isolation arrangements to enable their cleaning/replacement.

23.5.4

Automatic expansion valves are to be provided with manually operated bypass valves. Alternatively, duplicate automatic expansion valves will be accepted.

25 Tests and Inspections

25.1 Compressor

25.1.1

The Surveyor is to verify the materials used but need not witness the material tests.

25.1.2

The pressure boundary components of the compressor are to be hydrostatically tested in the presence of the attending Surveyors to 1.5 times the design pressure.

25.1.3

In addition to the hydrostatic test specified in 6-2-6/25.1.2, the compressors are to be leak tested in the presence of the attending Surveyor at the design pressure on the LP and HP side, as appropriate. This leak test may be performed using the mediums referenced in 6-2-16/1.1.

25.1.4

After completion of the tests referred to in 6-2-6/25.1.2, functional and capacity testing of the compressor is to be carried out at the manufacturer's plant in the presence of the Surveyor in accordance with an approved program. The functional tests should include recording of the refrigerant used, temperatures, pressures, testing of alarms and shut down, pressure relief devices and vibration measurements to ensure that the limits do not exceed those proposed by the manufacturer and that other features relating to the performance of the equipment are in accordance with the specification. Similarly, during the capacity test, power consumption and the refrigeration loads are to be recorded.

A certificate documenting the functional and capacity tests that were performed will be issued by the attending Surveyor.

25.3 Pressure Vessels

25.3.1

Pressure vessels including condensers, coolers and heaters under refrigerant pressure are to be hydrostatically tested by the manufacturer in the presence of the attending Surveyor to a test pressure equal to 1.5 times the design pressure. The condenser, heaters and evaporators are to be pressure-tested on both tube and shell sides.

25.3.2

Pressure vessels in the refrigerant and the brine system are to be leak-tested and the procedure followed is to be in accordance with 6-2-16/1.1.3.

25.5 Piping

25.5.1

After fabrication (e.g., bending, attachment of flanges and fittings, etc.), all refrigerant and brine pipes are to be subjected to a hydrostatic test pressure at 1.5 times the design pressure in the presence of the attending Surveyor. Alternatively, the test may be performed pneumatically using a suitable inert gas such as nitrogen.

25.5.2

The refrigerant and the brine piping are to be leak-tested at the design pressure in accordance with the procedures in 6-2-16/1.1.3.

25.5.3

For tests after installation, refer to Section 6-2-16.

25.7 Pumps

25.7.1

Refrigerant pumps and brine pumps are to be tested at the manufacturer's plant in the presence of the Surveyor. The pumps are to meet the hydrostatic and capacity test requirements of 4-6-1/7.5.2.

25.7.2

The refrigerant and the brine pumps are to be leak-tested at the design pressure in accordance with 6-2-16/1.1.3.

25.9 Relief Devices

The setting of the relief devices are to be verified by the Surveyor.

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SECTION 7 Ancillary Systems

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 Cooling Water Systems

1.1 Design Considerations

1.1.1

Cooling water pipes, valves and fittings are to be in accordance with the requirements of Part 4, Chapter 6.

1.1.2

The supply of cooling water for condensers is to be available from at least two independent sea connections, one preferably to be on the port and the other on the starboard side.

1.1.3

The maximum cooling water velocity through each condenser is not to exceed manufacturer's recommendations.

1.3 Pumps

At least two independent pumps are to be installed for the supply of cooling water to the refrigeration unit(s), one of which is to act as a standby. The standby pump may be used for other general service duties, except oil and bilge systems, provided its capacity is sufficient to simultaneously maintain the required supply of cooling water to the refrigeration unit(s).

1.5 Shell Connections

1.5.1

Shell connections are to be in accordance with the requirements of 4-6-2/9.13.

1.5.2

If the elevation of the condenser relative to the light water line is such that the manufacturer's recommended back pressure cannot be maintained in the overboard discharge line, then the overboard valve is to be of a spring loaded type.

3 Bilge and Drainage Systems

The refrigerating machinery space is to be efficiently drained. Bilge arrangements are to be in accordance with 4-6-4/5.

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SECTION 8 Fire Extinguishing Systems and Equipment

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 Cargo Spaces

Refrigerated cargo spaces are to be provided with a fixed fire extinguishing system complying with the requirements of 4-7-2/7.1.1. Where gas smothering system is used, the arrangements are to be in accordance with 4-7-3/3.

3 Refrigeration Machinery Spaces

Where refrigeration machinery is located in a dedicated space, at least two portable fire extinguishers complying with 4-7-3/15 are to be provided in the space. One of the required portable fire extinguishers is to be stowed near the entrance to the space.

5 Refrigerant Storage Space

Spaces other than those referred to in 6-2-8/3 above, which contain refrigerant cylinders, are to be provided with at least one portable fire extinguisher complying with 4-7-3/15, which is to be stowed near the entrance to the space.

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SECTION 9 Electrical Systems

1 General

Except as noted herein, compliance with Part 4, Chapter 8, as applicable, is required.

3 Cable Installation

Cables are not to be installed behind nor imbedded in the insulation. They may, however, pass through such insulation at right angles, provided they are protected by a continuous pipe with a stuffing tube at one end. For deck penetrations, these stuffing tubes are to be at the upper end of the pipe and for bulkhead penetrations, on the un-insulated side of the bulkhead.

5 Electrical Installation in Refrigerating Machinery Room and Cargo Hold

5.1

Electrical accessories such as switches, detectors, junction boxes, etc. installed in the refrigerating machinery room are to have IP44 enclosure and all other electrical equipment is to have IP22 enclosure.

5.3

Electrical equipment installed in the cargo holds is to be protected from mechanical damage. All electrical equipment in the cargo holds is to have IP55 enclosure.

5.5

Electrical equipment installed in the ammonia refrigerating machinery spaces is to be in accordance with 6-2-11/13.3.

7 Power Supply

Where the refrigerating plant is electrically driven, the electrical power is to be available from at least two generating sets. The capacity of the generating sets is to be such that, in addition to ensure the operation of the services essential for the propulsion and safety of the ship and services for providing minimum comfortable conditions of habitability, as required by 4-8-2/3.1, the following conditions are met:

7.1

Aggregate capacity of the generators is to be sufficient to supply the power to the refrigerating plant(s) mentioned in 6-2-6/3.3.2(a) or 6-2-6/3.3.3(a). Where the vessel is designed for the simultaneous carriage of integral refrigerated containers on deck, the aggregate capacity of the generators is to be sufficient to supply power to the refrigerated cargo spaces mentioned above and all of the electrical power sockets for these containers, to enable all modes of operations including cool down.

7.3

Where, due to operational requirements, it is not necessary to supply power simultaneously to all the electrical sockets, where fitted, for the refrigerated containers on deck, alternative aggregate capacity of power supply from the generators to that required in 6-2-9/7.1 will be considered.

7.5

With any one generator out of action, the remaining generator(s) are to be capable of supplying sufficient power to the refrigerating plant(s) and/or electrical power sockets in order to achieve and maintain the required steady state temperature in all of the loaded cargo spaces and/or containers when operating under the conditions specified in 6-2-6/3.3.1, subject to the applicability of 6-2-9/7.3.

9 Transformer

9.1

Where the refrigerating plants are supplied by power through transformers or converters, the system is to be so arranged as to ensure continuity of the power supply to the refrigerating plants, as follows:

With any one transformer or converter out of action, a standby transformer or converter is to be capable of supplying the power to the refrigerating plants. Alternatively, this requirement may be satisfied, provided there are alternative arrangements for supplying power to the circuit upon failure of the transformer or converter.

11 System Design

Coordinated tripping is to be provided between feeder and branch circuit protective devices for refrigerating plants.

13 Testing and Inspection

13.1 Motor Control Centers and Distribution Boards

Motor control centers used for refrigerant plants are to be tested in the presence of the Surveyor in accordance with 4-8-3/5.11.3.

For distribution boards, the tests as per 4-8-3/5.11.3 may be carried out by the manufacturer whose certificate of tests will be acceptable.

13.3 Motors

Motors of 100 kW (135 hp) and over are to be tested in the presence of the Surveyor in accordance with 4-8-3/Table 3. For motors below 100 kW (135 hp), the tests as per 4-8-3/Table 3 may be carried out by the manufacturer whose certificates of tests will be acceptable.

13.5 Electrical Installation

Testing of the electrical installation for refrigeration machinery is to be carried out in accordance with 4-8-4/29.

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SECTION 10 Instrumentation, Control and Monitoring

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 General

1.1

The control and monitoring systems are to ensure that the selected carriage temperature for the individual cargo spaces is maintained during all service conditions. The monitoring system is to be provided for refrigerating machinery and refrigerated cargo space temperatures.

1.3

For fruit carriers, the monitoring and control systems are additionally to ensure that the CO_2 levels in cargo spaces are continuously monitored and the levels selected are not exceeded during all service conditions.

3 Control

3.1

Control, instrumentation and monitoring necessary for operation may be provided at or in the proximity to the refrigeration machinery, the centralized control and monitoring station of the propulsion machinery, the navigation bridge or other similar spaces.

3.3

Where the refrigeration machinery is remotely controlled from the centralized control and monitoring station of the propulsion machinery, the navigation bridge or other similar spaces, means of independent controls and instrumentation and monitoring necessary for operation are to be provided at or in the proximity to the refrigeration machinery, together with means provided locally to disconnect or override associated remote controls.

3.5

See 6-2-10/Table 1 for required displays and alarms.

3.7

The control and monitoring for the temperature of circulating air entering and leaving each air cooler is to be independent from each other.

5 Temperature Measuring Equipment

5.1 Minimum Number of Sensors

For guidance, the minimum required number of sensors in a refrigerated space is to be determined based on the capacity and geometry of the space, as follows:

4 for up to 250 m^3 (8,828 ft^3) space

5 for up to 400 m^3 (14,124 ft^3) space

6 for up to 700 m^3 (24,178 ft^3) space

7 for up to 1200 m^3 (42,373 ft^3) space

8 for up to 1900 m^3 (67,090 ft^3) space

10 for up to 2800 m^3 (98,870 ft^3) space

5.3 Location of Sensors

In addition to 6-2-10/5.1, in each refrigerated space with forced air circulation through air coolers, at least one sensor is required for the circulating air. See also 6-2-10/3.7.

5.5 Remote Temperature Measurement

5.5.1

Sensors in refrigerated spaces are to be arranged in such a way that temperature reading is possible without entering the spaces.

5.5.2

Each refrigerated cargo space is to be provided with at least two temperature measuring instruments with separate power supply such that the temperature measurement of the space is possible in the event of a fault in any one of the measuring instruments.

5.5.3

Temperature reading devices or similar means are to be fitted for maintaining a log of cargo hold temperature.

5.5.4

Where temperature measuring systems are supplied by an individual source of power supply, such as transformer, converter or battery, a stand-by source of power is to be provided. Alternatively, this requirement may be satisfied, provided there are alternative arrangements for supplying power to the circuit upon failure of the transformer, converter or battery.

5.5.5

Number and arrangement of the remote temperature measuring system sensing elements is to comply with 6-2-10/5.1 and 6-2-10/5.3. The temperature sensing elements are to be permanently connected to their instruments and well-protected against damage.

5.7 Accuracy, FSD (Full Scale Deflection) Range

5.7.1

The measuring range of the system is to cover the entire anticipated temperature range plus an additional ±5°C (9°F).

5.7.2

The accuracy of the temperature measuring equipment is to be within ±0.5°C (0.9°F) for frozen cargo and ±0.2°C (0.4°F) for fruit.

5.7.3

Accuracy of instrumentation to a value higher than that stated in 6-2-10/5.7.2 above is required by some Administrations, depending on the cargoes carried. Accordingly, due attention is to be given to the requirements of various Port States during the design stages of the temperature monitoring and control systems, if it is intended for the vessels to transport cargoes to and from these ports.

7 CO_2 Measuring Equipment

All refrigerated cargo spaces intended for carriage of fruit are to be fitted with permanently installed equipment for indication of CO_2 content. The sensors are to be suitably positioned in the cargo spaces and are to be located away from the fresh air ducts.

9 Refrigerant Leakage Detection

9.1

Where the quantity of the refrigerant charge in the largest system exceeds the following per unit volume of the spaces in which it is located, the spaces containing the refrigerating machinery, and in the case of a direct expansion system, the refrigerated cargo spaces, are to be provided with a refrigerant leakage detection system complying with 6-2-10/9.3 and 6-2-10/9.5.

Refrigerant	*Concentration, kg/m^3 (lb/ft^3)*
R22	*0.14 (0.009)*
R134a	*0.25 (0.016)*
R404a	*0.48 (0.030)*
R407a	*0.33 (0.021)*
R407b	*0.35 (0.022)*
R407c	*0.35 (0.022)*
R410a	*0.44 (0.028)*
R410b	*0.43 (0.027)*
R507	*0.49 (0.031)*

9.3

The Refrigerant vapor detection system is to give an alarm and start mechanical ventilation in the event of refrigerant concentration exceeding the time-weighted average to which personnel may be repeatedly exposed to in the spaces.

9.5

The refrigerant vapor detection system referenced in 6-2-10/9.3 is also to be arranged to give an alarm and start mechanical ventilation when the refrigerant concentration exceeds a level where oxygen levels in the refrigerant machinery space are below 19.5% by volume. Alternatively, sensors for monitoring the oxygen level in the machinery space may be fitted and arranged to give an alarm should oxygen level drop below 19.5%.

11 Instrumentation and Monitoring

The indications and alarms in accordance with 6-2-10/Table 1 are to be provided at or in the proximity to the refrigeration machinery, the centralized control and monitoring station of the propulsion machinery, the navigation bridge or other similar spaces.

13 Alarm Call Button

All refrigerated spaces and air cooler rooms are to be fitted with at least one alarm call button located near the exit.

15 Automatic Controls

15.1 General

Where automatic control is fitted, compliance with the following is required. Additionally, the arrangements are to be in compliance with 6-2-10/1 through 6-2-10/13.

The control systems are to be designed to automatically maintain the selected carriage temperature in the individual cargo spaces and, additionally, for fruit carriers, the CO_2 level.

15.3 Control and Monitoring

15.3.1

The alarms and the indication as listed in 6-2-10/Table 1 are to be provided at the locations mentioned in 6-2-10/3.1.

15.3.2

Instrumentation and means of independent control and monitoring necessary for operation are to be provided at or in the proximity of the refrigeration machinery.

15.3.3

Adequate arrangements are to be provided to disable the automatic control mode and restore manual control.

15.5 Alarm Systems

15.5.1

Alarm systems are to be of the self-monitoring type and designed so that a fault in the alarm system will cause it to fail to the alarmed condition.

15.5.2

Alarming of other faults that may occur during the acknowledgment process is not to be superseded by such action.

15.5.3

Alarm systems are to be provided with effective means of testing.

15.7 Computer Based Systems

15.7.1

Where alarms are displayed on a visual display unit, they are to appear in the sequence in which the incoming signals are received and are to have priority, regardless of the mode the visual display unit is in.

15.7.2

The computer program and associated data considered to be essential for the operation of the system is to be stored in non-volatile memory.

15.7.3

Software is to be validated in accordance with a national, international or other recognized standard and demonstrated for verification.

15.9 Testing of Equipment

Testing of equipment associated with automatic or remote control systems, monitoring systems and computer-based systems is to be in accordance with Section 4-9-8.

For equipment that has been certified by ABS on an individual basis or certified under the ABS Type Approval Program, the tests previously carried out for compliance with Section 4-9-8 will be accepted, provided that the equipment being proposed is identical to the one previously tested.

17 Testing after Installation on Board

The following tests are to be carried out to the satisfaction of the Surveyor:

17.1

Local control of the refrigerating machinery is to be demonstrated. This is to include a demonstration of independent manual control and the disconnection or override of the automatic control system.

17.3

Where automatic control or remote control is provided, the ability to control from a remote control station is to be demonstrated. This is to include a demonstration to disable the automatic control mode and restore manual controls.

17.5

The required alarm control systems and displays are to be verified for satisfactory operation at the predefined set points.

17.7

The following equipment or systems are to be tested:

- The accuracy of the temperature measuring equipment in accordance with 6-2-10/5.7.
- CO_2 measuring system for refrigerated cargo spaces in accordance with 6-2-10/7.
- *Refrigerant leakage detection system in accordance with 6-2-10/9.*
- *Alarm call button in accordance with 6-2-10/13.*

TABLE 1
Instrumentation and Alarms

Item		*Display*	*Alarm*	*Remarks*
Compressor	Automatic stop		Activated	
	Lubricating oil	*Pressure	Low	Automatic stop (Low pressure)
	Driving motors	Running	Stop	
	Available driving motors	Running	Start	For auto start
	Discharge line - Pressure	*Pressure	High	Automatic stop (High pressure)
	- Temperature	*Temperature	High/Low	Automatic stop (High temp.)
	- Superheat	*Temperature	High	Automatic stop
	Suction line - Pressure	*Pressure	Low	Automatic stop (Low pressure)
	- Temperature	*Temperature	High	
	- Superheat	*Temperature	Low	Automatic stop (Low temp.)
	Intermediate stage (if fitted)	*Pressure	High	Automatic stop (High pressure)
Brine Lines	Brine pumps	Running	Stop	
	Available pumps	Running	Start	For auto start
	Brine cooler - inlet/outlet	*Temperature	High (outlet)	
	Pressure line	*Pressure	Low	
	Header tank	*Level	Low	
Condenser	Cooling water pumps	Running	Stop	
	Available cooling water pump	Running	Start	
	Cooling water - inlet	*Temperature		
	Cooling water - outlet	*Temperature	High	
Refrigerant receiver	Level	*Level	High/Low	
Refrigerating Machinery space	O_2 content (or, excessive refrigerant vapor content)		below 19.5% (excessive)	
Refrigerant leakage	Concentration in Refrigerating machinery space		Leakage (ppm above as per 6-2-10/9.3)	
	Concentration in Refrigerated spaces		Leakage (10 ppm)	Direct system
	Detection system		Failure	
Refrigerated spaces	Temperature measuring	Temperature	Deviation from set point	
	Left/Right hand cooler delivery air/return air	Temperature	Deviation from set point	
	CO_2 content	Percentage	Higher than the set point	For fruit carriers
	Fresh air fan (Full/Half speed)	Stop/Running/ Auto	Failure	
	Ventilation fan (Full/Half speed)	Stop/Running	Failure	For fruit carriers
Relative Humidity	Percentage		Deviation from set point	
Defrost	Time duration		Disabled	

Note: Those devices marked (*) are to be provided at or in the proximity to the refrigeration machinery.

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SECTION 11 Ammonia Refrigeration System

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 General

1.1

Refrigerating machinery using ammonia is to be designed, constructed and installed in accordance with the requirements of this Section and other applicable requirements the Rules.

1.3

Ammonia may be used only as a primary refrigerant in indirect refrigeration systems.

1.5

Ammonia refrigerant for use in direct expansion systems on-board refrigerated fish carriers will be specially considered subject to an assessment of all the features necessary to ensure the safety of the installation.

3 Design Considerations

3.1 Location of Refrigeration Machinery

3.1.1

Refrigerating units and associated equipment which contain ammonia are to be located in a dedicated space.

3.1.2

The dedicated space referred to in 6-2-11/3.1.1 is to be separated by gastight steel bulkheads and decks from other spaces.

3.3 Access and Openings

3.3.1

Access doors to the refrigerated machinery space are to be in accordance with the following requirements:

3.3.1(a) A minimum of two access doors located as far apart as possible are to be provided, one of which is to lead directly to the open deck. Water screens are to be provided above access doors, operable manually from outside the compartment.

3.3.1(b) The access doors are to be gastight and self-closing with no holdback arrangements and are to open outward from the refrigeration machinery space.

3.3.1(c) Access doors are not to open to the accommodation spaces.

3.3.1(d) Where one access is from a Category "A" machinery space, it is to be fitted with double door separation having a minimum space of 1.5 m (4.9 ft) between each door. The doors are to be self-closing and gastight with no holdback arrangements and the space between each door is to be provided with an independent ventilation system, the exhaust from which is to be led to atmosphere. Alternative access arrangements will be specially considered, provided a similar level of safety is maintained.

3.3.2

Access corridors leading to the refrigerating machinery space are to be ventilated by means of an independent mechanical exhaust system. This will not be required if the ventilation system required by 6-2-11/3.5 is also arranged to draw from the access corridors.

3.3.3

Duct, pipe and cable penetrations of bulkheads and decks of the ammonia refrigerating machinery spaces are to be made gastight.

3.5 Ventilation of the Refrigeration Machinery Space

The ammonia refrigerating machinery space is to be efficiently ventilated by means of mechanical exhaust ventilation designed in accordance with the following requirements:

3.5.1

The ventilation system is to be independent of other shipboard ventilation systems.

3.5.2

The ventilation system is to be designed for continuous operation.

3.5.3

The capacity of the ventilation system is to be of sufficient capacity to ensure at least 30 air changes per hour based on the total empty volume of the space.

3.5.4

Means are to be provided for stopping the ventilation fans and closing the ventilation openings from a readily accessible position.

3.5.5

Air inlet openings are to be positioned as low as practicable in the spaces being ventilated and exhaust openings as high as practicable to ensure that no ammonia accumulates in the space.

3.5.6

Exhaust duct outlets are to be positioned at least 10 m (33 ft.) from air intake openings, openings to accommodation spaces and other enclosed areas, and at least 2 m (6.5 ft.) above the open deck.

3.5.7

Ventilation fans are to be of non-sparking construction in accordance with 4-8-3/11.

3.7 Emergency Ventilation of Ammonia Refrigeration Machinery Space (2010)

Ammonia refrigerating machinery spaces are to be provided with a mechanical exhaust type gas evacuation system to quickly dissipate a catastrophic leak of ammonia to reduce the risk of fire and explosion. The system is to be designed and constructed in accordance with the following requirements:

3.7.1

The gas evacuation system is to be independent of other shipboard ventilation systems; however, it need not be independent of the ventilation system required in 6-2-11/3.5.

3.7.2

The gas evacuation system is to be arranged to automatically start when the concentration of ammonia in the space exceeds 300 ppm.

3.7.3

The combined capacity of the ventilation and gas evacuation fans is to be based upon the larger of the following

[A] A volume to ensure 40 air changes per hour based on the total empty volume of the space;

or;

[B] The capacity calculated using the following formula:

$Q = k\,G^{0.5}$

where

k = *0.07 (3.66)*

Q = *minimum combined capacity, in m^3/s (ft^3/s)*

G = *mass of ammonia in the largest refrigerating unit, in kg (lbs)*

3.7.4

The gas evacuation system controls are to be positioned outside of the space.

3.7.5

The exhaust duct outlets are to be positioned at least 10 m (33 ft.) from air intake openings, openings to accommodation spaces and other enclosed areas, and at least 2 m (6.5 ft.) above the open deck. In addition, the vent outlets are to be directed upward and arranged such as to ensure the discharge of any ammonia vapors would be away from accommodations and other occupied areas.

3.7.6

Gas evacuation fans are to be of non-sparking construction in accordance with 4-8-3/11.

3.9 Drainage of Ammonia Refrigeration Machinery Space (2010)

3.9.1

The ammonia refrigerating machinery space(s) is to be fitted with an independent bilge system.

3.9.2

The deck plating is to be arranged to facilitate easy cleaning and drying. No other plating above the deck is to be provided.

3.9.3

Where a deluge system (see 6-2-11/3.11) is fitted, the drainage and pumping arrangements are to be such as to prevent the build-up of free surfaces. The drainage system is to be sized to remove not less than 125% of the capacity of the water-spraying system.

3.11 Deluge System (2010)

Where a water deluge system is fitted, the emergency gas evacuation system in 6-2-11/3.7 may be reduced by 20%. The water deluge system arrangements are to be as follows:

3.11.1

The system is to be independent but may also be used for supply to the water screens required by 6-2-11/3.3.1(a).

3.11.2

The deluge system is to contain fresh water through a pressurized system.

3.11.3

The discharge nozzles in the space(s) protected are to be positioned such that the spray is directed over the entire area containing the ammonia refrigeration machinery.

3.11.4

The pressurized system is to consist of two pumps, a tank with a capacity to maintain discharge for a period of 30 minutes to all of the nozzles simultaneously in the protected space(s), the tank to be fitted with adequate safety relief arrangements, pressure gauge(s), level control and level gauge.

3.11.5

Means are to be provided to automatically maintain the required pressure and the water level in the tank. In the event of low pressure or the low level, an audible alarm is to sound in the refrigeration machinery room, refrigeration cargo control room, if fitted, and the engine room.

3.11.6

The water deluge system is to be arranged to automatically start when the concentration of ammonia in the space exceeds 300 ppm.

3.11.7

The electrical equipment in the ammonia refrigeration compartment is to be to IP55 enclosure.

3.13 Storage of Ammonia Cylinders

3.13.1

A maximum of 140 kg (308 lb) of reserve ammonia may be stored in the refrigerating machinery space. Reserve ammonia in excess of this amount is to be stored in a separate storage space designed and constructed in accordance with the requirements of this Section, unless 6-2-11/3.13.7 is applicable.

3.13.2

Portable steel ammonia storage cylinders satisfying the requirements of 6-2-6/5.15 are to be stowed in an efficiently ventilated dedicated space.

3.13.3

The ammonia storage space is to comply with the requirements of 6-2-11/3.1.2, 6-2-11/3.3.3 and 6-2-11/3.9.

3.13.4

Access doors to the storage space are to be in accordance with 6-2-11/3.3.1, except that two doors are not required.

3.13.5

The storage space is to be provided with a mechanical ventilation system complying with 6-2-11/3.5. Where the storage space is adjacent to the refrigerating machinery space, a common ventilation system servicing both spaces may be accepted.

3.13.6

Means for secure stowage and handling of the steel storage cylinders are to be provided.

3.13.7

Where due to limited space, the provision of a separate storage space is impracticable, alternative solutions, such as location of the storage cylinders in the space containing the ammonia refrigeration machinery, will be subject to special considerations, provided that the water deluge system and the leakage detection system is extended to take account of the additional ammonia stored in the space.

5 Materials *(2010)*

5.1

Components in contact with ammonia are not to contain copper, zinc, cadmium or alloys of these materials.

5.3

Components of rubber or plastic materials likely to be exposed to ammonia are not to be used.

5.5

Material for sea water cooled condensers is to be corrosion resistant to sea water.

7 Personnel Safety Equipment

7.1

An eye wash and shower unit are to be provided immediately outside of the refrigerating machinery room.

7.3 *(2010)*

The following safety equipment is to be provided and stored in a readily accessible, protected location outside of the refrigerating machinery room and is to be in addition to the equipment required by 4-7-3/15.5:

- *At least two sets of ammonia protective clothing, including refrigerant gas mask, helmet, boots and gloves.*
- *At least two sets of fireman's outfits complying with 4-7-3/15.5.*
- *Two or more power driven air compressors, to recharge breathing apparatus cylinders.*
- *One heavy duty adjustable wrench.*
- *Bottles of boric acid, vinegar and eye cups.*

9 Safety Devices

9.1

A rupture disc is not to be used in series with the safety relief valve.

9.3

The discharge from safety relief valves on the ammonia side is to be led into the sea below the lightest water line or into the water dump tank near the bottom of the tank.

9.5

Ammonia refrigeration systems are to be provided with automatic air purging devices. The discharge from the purging devices is to be led overboard below the lightest water line or to the water dump tank such that the discharge opening is submerged at all times. Where the connection is lead overboard, the discharge pipe is to be of heavy grade.

9.7

Where condensers are cooled by fresh water which is re-circulated, the fresh water system is to be equipped with pH meters to activate audible and visual alarms in the event of an ammonia leak.

11 Piping Arrangements

11.1

Ammonia pipes are to have provision for expansion and contraction encountered in service. The use of metallic flexible hoses for this purpose will be subject to approval by ABS.

11.3

Where flexible bellows are intended to be used in the ammonia refrigerant system, details and test data to show their suitability for the intended service are to be submitted.

11.5

Joints for piping conveying ammonia are to be butt welded, as far as practicable. For pipes up to 25 mm (1 in.) nominal diameter, socket welded joints may be accepted. Flanged joints are to be kept to a minimum and precautions are to be taken prior to disconnecting any such joints during repair and maintenance.

11.7

Piping for discharge of cooling sea water from the condenser is to be independent of other sea water piping systems and is to be led directly overboard without passing through accommodations or Category A machinery spaces.

11.9

Oil traps and oil drains are to be provided at the low points of the refrigerant system. Gauge lines and branches to level controls are not to be in locations where oil is likely to accumulate.

11.11

Overboard discharges are to be in accordance with 4-6-2/9.13.

13 Electrical

13.1 General

Except as noted herein, compliance with Section 6-2-9 is required.

13.3 Equipment and Installation in Hazardous Area

Ammonia refrigerating machinery spaces and storage spaces are considered hazardous locations. Electrical equipment and wiring are not to be installed in such locations unless essential for operational purposes. Where electrical equipment is installed in the above spaces, the following conditions are to be met:

13.3.1

Electrical equipment operated in the event of ammonia leakage, such as vapor detection and alarm system, is to be intrinsically safe type.

13.3.2

Emergency lighting fixtures of explosion proof type are to be provided in the above spaces. The switches for the lights are to be double pole type and located outside these spaces.

13.3.3

Electrical motors for gas evacuation fans or ventilation fans, if used for the gas evacuation system, are not to be located in the fan ducts or inside the ammonia refrigerating machinery spaces. They are to be located outside of the hazardous areas.

13.3.4 (2014)

For electrical equipment other than those referenced in 6-2-11/13.3.1 and 6-2-11/13.3.2, means are to be provided for automatic de-energizing when the concentration of ammonia vapor in the space exceeds 10,000 ppm.

13.3.5

Cables in these spaces are to be armored and the penetrations are to be through gas tight fittings.

15 Instrumentation, Control and Monitoring

15.1 General

Instrumentation, control and monitoring for the ammonia refrigeration system is to be in accordance with Section 6-2-10 and the following requirements.

15.3 Ammonia Vapor Detection and Alarm System

15.3.1

An ammonia vapor detection and alarm system is to be provided for the following locations:

15.3.1(a) The refrigerating machinery spaces; one detector per 36 m^2 (387 ft^2) of the space floor area.

15.3.1(b) One detector in the exhaust duct of the refrigerating machinery space ventilation system.

15.3.1(c) The access corridors leading to the ammonia refrigerating machinery spaces.

15.3.1(d) One detector in the ammonia storage space.

15.3.2

If the concentration of ammonia exceeds 25 ppm, the detectors are to activate audible and visual alarms. In addition, if the concentration of ammonia exceeds 300 ppm, the detectors in the refrigerating machinery space are to stop the refrigerating plant and activate the gas evacuation system.

15.3.3

Additional ammonia vapor detectors set to provide an alarm in a continuously manned space if the ammonia concentration exceeds 500 ppm are to be provided in the discharge pipes from safety relief valves.

15.3.4

Note that the refrigerant leakage detection system required in 6-2-11/15.3.1 is in lieu of the system required by 6-2-10/9.

15.5 Instrumentation and Alarms

The alarms listed in 6-2-11/Table 1 are to be provided at the locations specified in 6-2-10/3.

TABLE 1
Instrumentation and Alarms

Item		*Display*	*Alarm*	*Remarks*
Condenser	Leakage of ammonia into cooling fresh water system	pH Meter	Leakage	Where condensers are cooled by fresh water
Water Dump Tank	Level	Level	Low	
Ammonia Vapor Detection	Location mentioned in 6-2-11/15.3.2		Exceed 25 ppm	
	Discharge pipes from safety relief valves		Exceed 500 ppm	
	Refrigerating plant automatic stop		Stop (300 ppm)	
	Activation of gas evacuation		Activation	

17 Tests and Inspections

17.1

Gas tightness of openings or doors referred to in 6-2-11/3.3 is to be verified by the attending Surveyor.

17.3 *(2014)*

Electrical isolation of the refrigeration equipment at the set limit of 10,000 ppm of ammonia is to be demonstrated.

17.5

Ventilation air changes are to be verified by the attending Surveyor.

17.7

Satisfactory operational test of the emergency ventilation system is to be verified by the attending Surveyor.

17.9

Ammonia vapor detection and alarm system is to be demonstrated. This is to include a demonstration of the required audible and visual alarms and stopping the refrigerating plant and activation of the gas evacuation system in accordance with 6-2-11/15.3.

17.11

The required alarms and displays are to be verified for satisfactory operation at the predefined set points.

17.13

Automatic de-energizing of non-intrinsically safe electrical equipment required in 6-2-11/13.3.4 is to be demonstrated in the presence of the Surveyor.

PART

6

CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

SECTION 12 Controlled Atmosphere Systems

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 General

1.1

The requirements of this Section are applicable to systems installed onboard, either temporary or permanent, for generating nitrogen enriched gases and its supply to the refrigerated cargo spaces and to control the atmosphere in those spaces. Generation and supply of other non-toxic gases for this purpose will be subject to special consideration.

1.3

Portable nitrogen-generating equipment intended to serve multiple refrigerated cargo holds is to comply with all the relevant requirements of this Chapter and is to be approved in consideration with the number of specific refrigerated cargo spaces it is intended to serve.

1.5

The nitrogen-generating equipment is to be designed, manufactured and installed in accordance with good commercial practice and is to be suitable for intended service conditions including the marine environment. All pressure-retaining components are to comply with the requirements of Part 4, Chapter 4 and Part 4, Chapter 6, as appropriate.

3 Design Considerations

3.1

The controlled atmosphere plant is to be able to achieve and maintain the O_2 levels in the designated spaces within a range between 2% and 10% by volume. However, O_2 levels outside of this range will be considered, depending on the cargoes carried.

3.3

The controlled atmosphere plant is to be capable of controlling the CO_2 levels in the designated spaces within a specified range by means of nitrogen purge, mechanical scrubbing or other acceptable means.

3.5

Where CO_2 levels are to be controlled by nitrogen purge, the capacity of the nitrogen generator must take account of the respiration rate of the cargo and the maximum required level of CO_2 which is to be maintained in the designated spaces.

3.7 Capacity

3.7.1

The minimum required nitrogen generator capacity is to be such that the oxygen content in the cargo space can be reduced to a value below 5% within 24 hours after sealing of the cargo space or container, in accordance with the following equation:

$$Q = 0.07\ V$$

where:

Q = Hourly nitrogen generating capacity when delivering nitrogen having a purity of 97%, in m^3 (ft^3), at standard atmospheric conditions of pressure and temperature

V = General cargo carrier: Total empty volume of all cargo spaces which are to be supplied with nitrogen simultaneously, in m^3 (ft^3).

Container carrier: Total empty volume of all containers which are to be supplied with nitrogen simultaneously, in m^3 (ft^3)

General cargo and container carrier: Sum of the volumes calculated above, in m^3 (ft^3).

3.7.2

The required capacity of the nitrogen generator may vary due to variations in types of cargoes, sealing arrangements and other relevant parameters and therefore is to be specified by the designer/Owner. The specified capacity of the system is to be indicated on the submitted plans.

3.9

The nitrogen generator is to be capable of delivering its rated capacity against a back pressure at the cargo space inlet equal to the pressure setting of the PV valve which is protecting that space.

5 Nitrogen Generator Compressor

5.1

Nitrogen-generating systems utilizing compressors are to be provided with two or more compressors and prime movers which together will be capable of delivering the rated capacity. Each compressor is to be sized so that with one compressor out of operation, the system is to be able to maintain the O_2 content in all designated cargo spaces within the range specified in 6-2-12/3.1.

5.3

Alternatively, one compressor and prime mover may be accepted if the compressor is capable of delivering the specified capacity and provided that spares for the compressor and prime mover are carried to enable any failure of the compressor and prime mover to be rectified onboard.

5.5

Materials for crankshafts, connecting rods, cylinders and cylinder covers, housings, rotors and rotor casings of reciprocating and rotary compressors, as applicable, are to be in accordance with the applicable requirements of Part 2, Chapter 3. Materials to other recognized standards will be considered. Material tests need not be witnessed by the Surveyor.

5.7

Air-cooled compressors are to be designed for an air temperature of at least 45°C (113°F). Water-cooled compressors are to be designed for a water temperature of at least 32°C (90°F).

7 *Location and Access for Compartments Containing Gas Generating Equipment* (2010)

7.1

The air compressor and the nitrogen generator may be installed in the engine room or in a separate compartment.

7.3

Where a separate compartment is provided, it is to be:

7.3.1

Treated as "other machinery spaces" with respect to fire protection

7.3.2

Positioned outside the cargo area

7.3.3

Fitted with an independent mechanical extraction ventilation system providing at least six (6) air changes per hour

7.3.4

Fitted with a low oxygen alarm

7.3.5

Arranged with no direct access to accommodation spaces, service spaces and control stations

7.5

Where fitted, a nitrogen receiver/buffer tank may be installed either in a dedicated compartment or in the separate compartment containing the air compressor and the generator. Where the nitrogen receiver/buffer tank is installed in an enclosed space, the access is to be arranged only from the open deck and the access door is to open outwards. Permanent ventilation and alarm are to be fitted as in 6-2-12/7.3.3 and 6-2-12/7.3.4 above.

In order to permit maintenance, means of isolation are to be fitted between the generator and the receiver.

7.7

Where the gas generating equipment is located in a container positioned on the open deck, the following requirements are to be met:

7.7.1

The container is to be provided with a mechanical ventilation system of the exhaust type giving at least 6 air changes per hour based on total volume of the container.

7.7.2

The outlets of the ventilation exhaust ducts from the container are to be located such that the exhaust cannot enter enclosed spaces on the vessel.

7.7.3

Means for stopping the ventilation fans and closing all of the openings to the gas generator container are to be from outside.

7.7.4

Unrestricted access to the container is to be possible under all loading conditions.

7.7.5

Two portable fire extinguishers complying with 4-7-3/15.1 are to be provided inside the container, of which one is to be stowed near the entrance to the container. Where the compressors are driven by internal combustion engines and the fuel tanks are located inside the container, an approved fixed fire extinguishing system complying with 4-7-3/3 may be required, depending upon the arrangement.

7.7.6

Notices are to be posted to indicate that the container is a dangerous area and may contain a level of oxygen which will cause asphyxiation and will not support human life due to presence of an inert gas.

7.7.7

Means are to be provided for stopping the gas generator from outside of the container.

7.7.8

The container is to be properly secured to the vessel. The container is to be designed considering proper support for the equipment and is to be suitable for the marine environment. In this regard, reference may be made to the ABS Guide for Certification of Container Securing Systems*, the ABS* Rules for Certification of Cargo Containers *and the available certifications contained therein.*

9 Gas and Compressed Air Piping System

9.1 Installation

9.1.1

Where flexible hoses on deck are intended to be used for the supply of nitrogen gas to the refrigerated cargo spaces, they are to be of an approved type complying with the requirements of 4-6-2/5.7 Means are to be provided for protecting these hoses against damage.

9.1.2

Vessels utilizing either portable or fixed nitrogen generation equipment are to be fitted with a permanently installed piping system complying with Part 4, Chapter 6 for the supply and distribution of nitrogen (N_2) gas. A positive closing isolation valve is to be fitted in the gas supply line at the inlet to the refrigerated cargo space. This valve arrangement is to be in accordance with 6-2-12/9.3.1 or 6-2-12/9.3.2.

9.1.3

Exhaust of O_2 and N_2 enriched gases from nitrogen generators are to be led to a safe location in the weather, at least 2 m (6.5 ft) above the open deck and 5 m (16.5 ft) away from ventilation inlets and openings to enclosed spaces

9.1.4

Gas pipes are not to pass through accommodation spaces, ducts or tunnels.

9.1.5

Gas pipes passing through service, machinery and control spaces are to be led through gas tight pipes.

9.3 Valve and Fittings

9.3.1

Each gas inlet line to an individual controlled atmosphere space is to be equipped with two shut-off valves and an intermediate vent valve. Discharge from the vent valve is to be led to a safe location in the weather, at least 2 m (6.5 ft) above the open deck and 5 m (16.5 ft) away from ventilation inlets and openings to enclosed spaces. The shut-off valves are to be provided with arrangements for locking in the closed position.

9.3.2

If a portable nitrogen generating plant is used, the arrangement in 6-2-12/9.3.1 may be dispensed with if it is not possible to supply nitrogen to more than one space at a time. In this case, each permanent gas inlet line is to be equipped with a screw down non return valve provided with arrangements for locking it in the closed position.

9.3.3

Filters are to be provided in the air supply to membrane separators and pressure swing adsorption carbon beds to ensure filtration of oil, debris and water particulate.

11 Safety Relief Devices

11.1

Safety relief devices are to be provided in each section of pipe that may be isolated by valves and may build up a pressure in excess of the design pressure. Discharges from relief valves on gas lines are to be led to the weather, at least 2 m (6.5 ft) above the open deck and 5 m (16.5 ft) away from ventilation inlets and openings to enclosed spaces.

11.3

Each air compressor for the nitrogen generating plant is to be provided with a relief valve on the discharge side.

11.5

Pressure vessels with isolating valves are to be equipped with a pressure relief valve set to relieve at a pressure not greater than the design pressure.

13 Cargo Spaces Under Controlled Atmosphere and Adjacent Spaces

13.1 General

13.1.1

Where the tween-deck spaces within cargo holds are fitted with separate means of maintaining controlled atmosphere conditions, each tween-deck space is to be considered an independent gas tight compartment. For container carriers where the containers stowed under deck are supplied with a low oxygen atmosphere, each container is to be considered a gas tight compartment.

13.1.2

Each cargo space under controlled atmosphere conditions is to be made gas tight, as far as practicable The arrangements are to be such as to ensure that when cargo space is pressurized with an over pressure of 20 mm of water column, the time taken for a 40% pressure drop is greater than 16 minutes.

13.1.3

Hatch covers and doors to spaces under controlled atmosphere are to be provided with locking arrangements and warning notices informing about the low oxygen atmosphere.

13.1.4

Warning notices are to be posted at all openings to spaces under controlled atmosphere to prevent inadvertent opening while the space is under the controlled atmosphere.

13.3 Pressure and Vacuum Considerations

13.3.1

Each cargo space or compartment under controlled atmosphere is to be provided with a pressure and vacuum relief valve (PV valve) to limit the positive and negative pressure below that for which the space is designed.

13.3.2

The pressure relieving capacity of the PV valve is to be such as to ensure that the pressure in the space does not exceed the design limits referred to in 6-2-12/13.3.1 above, when the gas generating unit is delivering at its maximum capacity to a single cargo space or compartment. Consideration is also to be given to pressure changes caused by defrost cycles.

13.3.3

Outlets of PV valves are to be located at least 2 m (6.5 ft.) above the open deck and 5 m (16.5 ft.) away from air inlets and openings to accommodation spaces, service spaces, machinery spaces and other similar manned spaces.

13.3.4

The PV valves are to be of a type suitable to satisfy the requirements of 6-2-12/13.3.1 and are to be capable of operating at ambient temperatures of 0°C (32°F) or less.

13.3.5

Arrangements for the protection of cargo spaces or compartments against over- or underpressure other than those referred to above will be the subject of special consideration.

13.5 Bilge and Drainage Arrangements

13.5.1

Liquid sealed traps on drains from cargo spaces, air cooler trays, etc. are to have sufficient liquid head to withstand the design over pressure when the Controlled Atmosphere system is in operation. Ship motions and over pressure of air circulation fans are to be considered when determining the required liquid head.

13.5.2

The liquid in the liquid seal traps is to be of a type that will not freeze or evaporate under any ambient condition.

13.5.3

Spaces under controlled atmosphere are not to have bilge wells or drain tanks common with spaces not intended for controlled atmosphere.

13.5.4

Where it is intended to gain access to the tween-deck spaces referred to in 6-2-12/13.1.1, any open-ended interconnecting pipe work between such spaces is to be arranged to prevent nitrogen gas from escaping from one gas tight space to another.

13.7 Ventilation

13.7.1

The ventilation inlets and outlets of cargo spaces under controlled atmosphere are to be provided with positive closing gas tight valves.

13.7.2

All ventilation outlets from spaces under controlled atmosphere are to be located at least 2 m (6.5 ft.) above the open deck and 5 m (16.5 ft.) away from air inlets and openings to accommodation spaces, service spaces, machinery spaces and other similar manned spaces.

13.7.3

Suitable arrangements for gas freeing the spaces under controlled atmosphere conditions are to be provided. Air circulation and ventilation fans may be used for this operation. The ventilation outlets used for gas freeing are to be directed vertically upwards.

13.7.4

Compartments other than tanks, void spaces or other similar areas where personnel do not normally have access, which are adjacent to refrigerated cargo spaces under controlled atmosphere, and other normally accessible spaces containing gas piping where gas leakage may create an oxygen deficient atmosphere, are to be provided with permanent mechanical ventilation systems of the positive pressure type with a capacity of at least 2 air changes per hour based on total volume of the space. The ventilation is to be able to be controlled from outside of the space. The permanent ventilation outlets are to be located in accordance with 6-2-12/13.7.2.

13.7.5

Cargo spaces with containers under controlled atmosphere which are required to be entered by personnel are to be provided with ventilation arrangements which are capable of maintaining a minimum of 19% oxygen (by volume) throughout the space when operating under the conditions specified in 6-2-12/3.5. Ventilation rate calculations are to be based upon a 100% gas leakage rate from the containers into the cargo space. The ventilation is to be able to be controlled from outside of the space. The permanent ventilation outlets are to be in accordance with 6-2-12/13.7.2.

15 Instrumentation, Control and Monitoring

15.1 General

15.1.1

In addition to Section 6-2-10, compliance with the following is required.

15.1.2

Within the specified ranges, the levels of O_2 and CO_2 are to be able to be maintained with an accuracy within ±0.2%.

15.1.3

A permanently installed monitoring system is to be arranged to display the O_2 and CO_2 content in all spaces under controlled atmosphere. The equipment for measuring CO_2 content may be the same as that required in 6-2-10/7.

15.1.4

Injection of nitrogen and removal of CO_2 may be arranged either manually or automatically.

15.3 Sampling

15.3.1

The permanently installed monitoring system is to be provided with independent sampling lines or gas sensors for each cargo space under controlled atmosphere.

15.3.2

Where the sampling lines are connected to a monitoring unit which is located in an enclosed space, that space is to be ventilated at a rate which is at least equivalent to the sampling flow rate.

15.3.3

The exhaust gases from measuring and analysis devices are to be discharged to a safe location on the open deck without creating a back pressure. The exhaust outlets are to be positioned in accordance with 6-2-12/13.7.2.

15.3.4

Sampling line arrangements are to be such as to prevent condensation and freezing of water in the lines under all operating conditions. Inlets of sampling lines are to be provided with filters to prevent dirt and debris entering the lines.

15.3.5

In addition to the sampling line or gas sensor required in 6-2-12/15.3.1, another closeable sampling line is to be provided for each cargo space under controlled atmosphere. This line is to be arranged for attachment of portable O_2 and CO_2 measuring devices as close as possible to the space served.

15.3.6

Portable equipment for measuring O_2 and CO_2 is to be available onboard at all times.

15.5 Analyzing

15.5.1

If an automatic control system is installed, gas analyzing equipment independent from the one used by the monitoring system is required. Separate gas sampling lines are to be provided for both systems.

15.5.2

Where a gas monitoring system with sequential analyzing is arranged, the system is to be designed so that each measuring point is analyzed at hourly intervals or other suitable duration specified by the Owner/builder. The instruments are to clearly indicate the space being analyzed. Direct readout of the gas quantity in any space under controlled atmosphere is to be available on demand.

15.5.3

Gas analyzing equipment is to be capable of self-calibration and manual calibration with known gases at both zero and full scale. The accuracy of the O_2 analyzers is to be within ±0.1% by volume. The accuracy of the CO_2 analyzers is to be within ±0.25% by volume.

15.7 Precaution for Low Level of O_2

15.7.1

The following spaces are to be provided with permanently installed equipment for monitoring O_2 content and be capable of alarming when the O_2 level is low:

15.7.1(a) All normally accessible spaces adjacent to spaces under controlled atmosphere.

15.7.1(b) Cargo spaces not under controlled atmosphere adjacent to spaces under controlled atmosphere and spaces where gas leakage may create an oxygen deficient atmosphere, e.g., spaces containing scrubber units or gas piping.

15.7.1(c) Cargo spaces which contain containers under controlled atmosphere.

15.7.1(d) Ship compartments or containers housing gas generating equipment.

15.7.2

An automatic pre-discharge warning alarm is to be fitted in each space under controlled atmosphere. The alarm is to be arranged to give audible signals continuously for 60 seconds before the gas discharge into that space commences. The alarm may be connected with the O_2 analyzer in a manner that it does not sound if the oxygen level in the space is below 14% by volume.

15.9 Monitoring and Alarm

The conditions as per 6-2-12/Table 1 are to be individually alarmed at the manned station for the spaces specified in 6-2-10/3.

17 Electrical

17.1 General

Except as noted herein, compliance with Section 6-2-9 is required.

17.3 Power Supply

17.3.1

Aggregate capacity of the electrical generators is to be sufficient to simultaneously supply the power to the entire controlled atmosphere system and the refrigerating system referred to in 6-2-9/7.1.

17.3.2

With any one generator out of action, the remaining generator(s) is to be capable of supplying the power to the controlled atmosphere system, excluding standby units, and the refrigerating system referred to in 6-2-9/7.3.

17.3.3

The power for the controlled atmosphere system is to be supplied from the main switchboard by feeders separate from those for other systems.

17.3.4

As an alternative to 6-2-12/17.3.2, an independent generating set providing power for the controlled atmosphere system may be accepted, provided arrangements are made to connect the controlled atmosphere system to the ship service generators, which are to have adequate total capacity to simultaneously carry the rated load of the controlled atmosphere system and the services essential for the propulsion and safety of the ship, services for providing minimum comfortable conditions of habitability and the entire refrigerating system.

17.5 Cable Penetration

Cable penetrating the boundaries of the gas generator compartment and spaces under controlled atmosphere are to be arranged gastight by use of cable glands.

19 Ethylene and Carbon Dioxide Scrubbers

19.1

Permanently installed piping complying with Part 4, Chapter 6 is to be provided between the scrubber units and the cargo spaces under controlled atmosphere.

19.3

The piping referred to in 6-2-12/19.1 is to be in accordance with 6-2-12/9.1.4 and 6-2-12/9.1.5.

19.5

Positive closing isolation valves are to be fitted at the connections with the cargo spaces under controlled atmosphere.

19.7

Exhausts from the scrubber units are to be led to a safe location in the weather, at least 2 m (6.5 ft) above the open deck and 5 m (16.5 ft) away from air inlets and openings to accommodation spaces, machinery spaces and other similar manned spaces.

21 Humidification Equipment

Where the cargo space under controlled atmosphere is equipped with a humidification system to control relative humidity of the space, the humidification system is to be in accordance with the following requirements:

21.1

For general guidance, the humidification system is to be capable of increasing the relative humidity in each of the intended cargo spaces up to a level of 90% at the specified space temperatures and maintain the selected level constant within ±5%.

21.3

The humidification system lines in the refrigerated cargo spaces are to be installed to facilitate ease of drainage and are to be provided with suitable heating arrangements, as applicable.

21.5

Permanently installed equipment for monitoring relative humidity in the cargo spaces is to be provided.

21.7

The deviation of relative humidity from the predetermined set point in each cargo space is to be individually alarmed at the monitoring station.

23 Personnel Safety Equipment

23.1

Means are to be provided to re-oxygenate the cargo spaces and compartments prior to gaining entry into the spaces which were under controlled atmosphere conditions. Until the O_2 levels which are considered safe for entry have been achieved, entry into such spaces is to be prevented.

23.3

At least ten portable oxygen monitors with alarms are to be provided onboard.

23.5

At least one portable gas analyzer capable of measuring O_2 levels in the atmosphere is to be provided onboard for use prior to entry into the spaces under controlled atmosphere. This portable gas analyzer is in addition to the equipment required in 6-2-12/15.3.6.

23.7

A means of two-way communication is to be provided between the cargo spaces under controlled atmosphere and the nitrogen release control station. If portable radiotelephone apparatus are adopted to comply with this requirement, at least three sets are to be provided onboard. This equipment is in addition to the equipment required by SOLAS Chapter III, Regulation 6.

23.9

One set of oxygen resuscitation equipment is to be provided onboard.

23.11

Two self-contained breathing apparatus equipped with built in radio communication and lifeline with a belt are to be provided onboard together with fully charged spare air bottles with a total free air capacity of 3600 liter (950 US gallons) for each breathing apparatus. This equipment is in addition to the equipment required by SOLAS Chapter II-2, Regulation 17.

25 Operations, Equipment and Procedures Manual

An Operations, Equipment and Procedures Manual is to be available onboard. The manual is to provide the following information:

25.1

General information about controlled atmospheres including explanation such as what is controlled atmosphere, need for controlled atmosphere, method of controlling atmosphere composition, danger associated with oxygen depleted atmosphere, insidious leakage of gas, etc.

25.3

Complete description of the ship's controlled atmosphere installation and diagrammatic arrangements showing the details of the gas tight compartments.

25.5

Procedures for gas freeing of Controlled Atmosphere (CA) spaces, methods of ascertaining adequacy of oxygen prior to entry, methods of communication in CA spaces.

25.7

Procedures for entering the CA spaces after gas freeing.

25.9

Procedures for loading adjacent cargo spaces.

25.11

Procedures prior to starting controlled atmosphere equipment.

25.13

Procedures for opening shut-off valves on nitrogen distribution branch lines and attachment of nitrogen distribution hoses, where applicable.

25.15

Procedures for functional testing portable gas generating unit each time it is placed onboard.

25.17

Procedures during the voyage with controlled atmosphere.

25.19

Equipment maintenance procedures and list of spare parts.

25.21

Operation, maintenance and calibration instructions for all types of gas detecting, analyzing and alarming equipment onboard associated with controlled atmosphere system.

25.23

Emergency procedures related to erroneous instrumentation.

25.25

Emergency procedures related to personnel overcome by oxygen deficiency.

25.27

Emergency procedures related to entry using breathing apparatus.

25.29

Instructions for atmosphere testing and gas freeing of spaces without permanent ventilation.

27 Tests and Inspections

27.1

Compressor parts subject to elevated pressure are to be hydrostatically tested at the manufacturer's plant in the presence of the ABS Surveyors to 1.5 times their respective design pressure.

27.3

After completion, functional and capacity testing of the nitrogen generator is to be carried out at the manufacturer's plant in the presence of the Surveyor in accordance with an approved program. The functional tests should include testing of alarms, shut downs and pressure relief devices. Capacity and quality of the nitrogen produced may alternatively be verified onboard, in the presence of the Surveyor.

27.5

Air leakage test for cargo spaces are to be witnessed by the attending Surveyor.

27.7

Sample lines are to be tested in the presence of the attending Surveyor for leakage and blockage.

27.9

The setting of the PV valves is to be verified by the attending Surveyor.

27.11

The accuracy of the levels of O_2 and CO_2 in all spaces under controlled atmosphere is to be verified by the attending Surveyor in accordance with 6-2-12/15.1.2.

27.13

Accuracy of the O_2 analyzers and CO_2 analyzers is to be verified by the attending Surveyor in accordance with 6-2-12/15.5.3.

27.15

Low level alarm of O_2 and automatic nitrogen pre-discharge warning alarm are to be demonstrated in accordance with 6-2-12/15.7.

27.17

The required alarms and displays are to be verified for satisfactory operation at the predefined set points.

27.19

The requirements in 6-2-9/13 are to be complied with, as applicable.

TABLE 1
Instrumentation and Alarms

Item		*Display*	*Alarm*	*Remarks*
Compressor	Automatic stop		Activated	
	Lubricating oil	Pressure	Low	Automatic stop (Low pressure)
	Discharge line – Pressure	Pressure	High	Automatic stop (High pressure)
	Suction line – Pressure	Pressure	Low	
O_2 Content	Spaces under controlled atmosphere	Content	Deviation from set point	6-2-12/15.1.3
	Accessible spaces/cargo spaces adjacent to spaces under C.A.	Content	Low	6-2-12/15.7.1(a) and 6-2-12/15.7.1(b)
	Gas generating compartments	Content	Low	6-2-12/15.7.1(d)
	Gas generating container	Content	Low	6-2-12/15.7.1(d)
	Cargo spaces containing containers under controlled atmosphere	Content	Low	6-2-12/15.7.1(c)
	Accessible spaces containing scrubber units and gas piping	Content	Low	6-2-12/15.7.1(b)
CO_2 Content	Space under controlled atmosphere	Content	Deviation from set point	
Gas Measuring System	Failure		Failure	
	Accuracy		Out of range	
Humidification System	Relative humidity	Relative humidity	Deviation from set point	If humidification system is fitted

PART
6

CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

SECTION 13 Refrigerated Cargo Container Carrier

1 General

1.1

Insulated containers are not considered part of the classed installation. However, for installations where the containers are supplied with cooled air from the vessel's refrigeration system, in accordance with 6-2-1/7.1.3 and 6-2-1/7.1.4, the requirements of this Section will apply.

1.3

Where requested, insulated containers will be certified in accordance with the *Rules for Certification of Cargo Containers*.

3 Porthole Refrigerated Cargo Container Carrier

3.1 Design Considerations

3.1.1

The vessel's refrigeration system is to be designed, constructed and installed in accordance with the requirements of this Section and other applicable requirements.

3.1.2

Where cargo cells are to be insulated, the arrangements are to be in accordance with the applicable requirements of Section 6-2-5.

3.1.3

Space heating of the cargo cells will be subject to special consideration.

3.1.4

The air circulation and fresh air ventilation system serving the containers is to be based upon the air volume of each empty container connected to the system. Air circulation for each connected container is to be 50 to 70 air changes per hour for fruit cargoes and 30 to 40 air changes per hour for frozen cargoes.

3.1.5

Fresh air ventilation for each container is to be at least two (2) air changes per hour.

3.1.6

Means are to be provided for monitoring CO_2 levels in each air cooler battery.

3.3 Ducts and Couplings

3.3.1

Ducts, couplings and air cooler casings are to be airtight, as established by tests conducted in accordance with 6-2-13/7.

3.3.2

Where a container stack is supplied with cooled air from its own air cooler, the air flow to each connected container is to be within ±5% of the design value.

3.3.3

Insulation installed on the inside of ducts is to be of a type that is not affected by moisture and is resistant to abrasion. The properties required by 6-2-5/5.11 are also applicable.

3.3.4

Where couplings are pneumatically actuated, the compressed air piping, valves and fittings are to be in accordance with Part 4, Chapter 6 and are to be protected against freezing.

3.3.5

The compressed air system referred to in 6-2-13/3.3.4 is to incorporate moisture traps to ensure the air supply is sufficiently dry to prevent ice formation when cargo cell temperatures are below 0°C (32°F).

3.3.6

In order to protect against icing, the outer surface of the coupling connections is to be insulated.

3.5 Air Coolers

When the total internal volume of all containers connected to a single air cooler exceeds 300 m^3 (10,593 ft^3), the air cooler coils are to be divided into at least two independent sections such that any one of them may be isolated without affecting the operation of the other. Alternatively, at least two independent air coolers are to be fitted.

3.7 Instrumentation, Control and Monitoring

Except as noted herein, refrigerating machinery plants and machinery spaces are to comply with the requirements in Section 6-2-10.

3.7.1 Temperature Monitoring

3.7.1(a) Delivery and return air ducts for each container are to be fitted with a thermometer. Where a group of containers is being served by one air cooler with common fans, the individual thermometers may be replaced by common thermometers for the delivery air.

3.7.1(b) Remote temperature monitoring of delivery and return air ducts is to comply with the requirements of 6-2-10/5.5.2, 6-2-10/5.5.3 and 6-2-10/5.7, except that the sensors in the delivery air ducts need not be connected to separate measuring instruments if the delivery air temperature is monitored locally.

3.7.1(c) The sensors are to be permanently connected to their instruments and protected against damage.

3.7.2 Monitoring

The display and alarms are to be provided in accordance with 6-2-13/Table 1 at the locations specified in 6-2-10/3.

TABLE 1
Instrumentation and Alarms

Item	*Display*	*Alarm*
Return air / Delivery air Temperature	Temperature	Deviation from set point
CO_2 Level in each Air Cooler Battery	Content	High

3.9 Electrical

The requirements in Section 6-2-9 are applicable.

3.11 Automatic Control

Where automatic control is provided for refrigerating machinery, compliance with 6-2-10/15, as applicable, is required.

5 Integral Refrigerated Cargo Container Carrier

(1 July 2016) Note: Please refer to the ABS *Guide for Carriage of Integral Refrigerated Containers On Board Ships*, for additional requirements.

5.1 Design Considerations

5.1.1

Where water-cooled condensers are provided, the cooling water flow rate is to be between 11 and 26 liters per minute.

5.1.2

Cooling water systems are to be in accordance with Section 6-2-7, as applicable.

5.1.3

Cargo cells containing containers are to be provided with sufficient air freshening capability to dissipate metabolic gas and also to ensure that the cell temperature does not exceed 10°C (18°F) above ambient while operating under the conditions specified in 6-2-6/3.3.1.

5.1.4

Where refrigerated cargo containers are carried in open hatch or hatchless cargo holds of a container vessel, the ventilation, bilge, hold temperature, etc. will be subject of special consideration.

5.3 Instrumentation, Control and Monitoring

Monitoring in accordance with the following 6-2-13/Table 2 is to be provided at a location specified in 6-2-10/3.

TABLE 2
Instrumentation and Displays

Item	*Display*
Power Supply (Monitoring)	Status
Compressor Running	Running
Defrost	Activate
Temperature in range	Temperature

5.5 Electrical

5.5.1

The requirements in Section 6-2-9 and the following are to be complied with.

5.5.2

Receptacles and plugs of different electrical ratings are not to be interchangeable. They are to be in accordance with ISO standard 1496-2 or equipment compatible with ISO standard.

5.7 Automatic Control

Where an automatic control system is provided, compliance with 6-2-10/15 is required.

7 Tests and Inspections

7.1 Porthole Refrigerated Cargo Container Carrier

7.1.1

Measurements are to be carried out in the presence of the attending Surveyor during on-board trials to demonstrate the air circulation and ventilation rates are as per 6-2-13/3.1.4 and 6-2-13/3.1.5.

7.1.2

The air tightness required by 6-2-13/3.3.1 is considered satisfactory when the leakage rate does not exceed 0.5% of the total volumetric flow rate at the design pressure. Tests to establish compliance are to be conducted on the installed system in the presence of the attending Surveyor.

7.1.3

Tests to establish that the cooled air distribution is in compliance with 6-2-13/3.3.2 are to be conducted on the installed system in the presence of the attending Surveyor.

7.1.4

Compressed air lines connected to the coupling actuators referred to in 6-2-13/3.3.4 are to be tested to 1.5 times the design pressure.

7.1.5

The electrical test requirements in 6-2-9/13 are to be complied with.

7.1.6

The alarms and displays required by 6-2-13/3.7 are to be verified for satisfactory operation at pre-defined set points.

7.1.7

In order to simplify shipboard testing, each type of air ducting system with couplings, an air cooler and circulating fans which are completely assembled at the manufacturer's plant may be tested prior to installation onboard in accordance with the following requirements:

7.1.7(a) The test is to be performed in accordance with an approved test program in the presence of the Surveyor.

7.1.7(b) The *k*-values for the duct and cargo cells are to be established as per the requirements of 6-2-16/3.3.3.

7.1.7(c) Air leakage rate for the air distribution duct, couplings and air cooler casings is to be measured.

7.1.7(d) Air distribution in the air ducting system for a stack of containers is to be measured.

7.1.7(e) Air circulating fans are to be tested in accordance with 6-2-16/3.1.1.

7.3 Integral Refrigerated Cargo Container Carrier

7.3.1

The design values required for compliance with 6-2-13/5.1.3 are to be shown on the ventilation fan capacity curve and, by performing on-board trials in the presence of the attending Surveyor, the capacity curve is to be verified at the prevailing ambient conditions.

7.3.2

The electrical test requirements in 6-2-9/13 are to be complied with.

7.3.3

The alarms and displays, where fitted for compliance with 6-2-13/5.3, are to be verified for satisfactory operation at the pre-defined set points.

7.3.4

Cooling water flow rate to the condensers is to be measured for compliance with 6-2-16/1.13.

7.3.5

Air freshening ventilation fans for cargo cells are to be tested in accordance with applicable requirements in 6-2-16/3.1.1.

PART
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CHAPTER **2 Vessels Intended to Carry Refrigerated Cargoes**

SECTION **14 Refrigerated Edible Bulk Liquid Tanker**

Note: Text in *italics* is considered necessary as conditions of classification (i.e., compulsory requirements). (See 6-2-1/5.3.)

1 General

1.1

The requirements of this Section are applicable to vessels defined in 6-2-1/13.25 requiring the notation referred to in 6-2-1/7.1.5 for the carriage of refrigerated edible bulk liquids.

1.3

Unless otherwise stated in this Section, the requirements of these Rules are applicable.

1.5

Due regard is to be given to the requirements of the Flag State and the Port State for the carriage and transportation of edible products.

3 Design Considerations

The material used is to be in accordance with the requirements of Part 2 and the following:

3.1

Materials used for the construction of the cargo containment, the associated piping, pumps and valves are to be suitable to withstand the design service temperatures, pressures and are to be compatible with the products carried. Materials incompatible with the edible products being carried are not to be used. Details of the materials are to be submitted for review.

3.3

The use of non-metallic materials for the cargo piping system will be the subject of special consideration. Accordingly, relevant details are to be submitted for review.

5 Hull Structure

For design and construction of the hull structure, refer to Part 3.

7 Cargo Containment System

7.1 Cargo Tanks

7.1.1

Cargo tanks, both independent and integral, are to be designed and constructed in accordance with the applicable requirements in Part 3. Integral tanks are also to comply with the requirements for integral tanks on chemical carriers in 5C-9-4/1.1 and 5C-9-4/1.3.

7.1.2

Independent pressurized tanks referred to in 6-2-1/13.27.4 are to be designed and constructed in accordance with Part 4, Chapter 4, as applicable.

7.1.3

The supports for the independent cargo tank(s) are to be designed in accordance with the requirements of a recognized national or international pressure vessel design code to withstand the static and dynamic loads with liquid full cargo tanks.

7.1.4

The independent cargo tanks are to be fitted with anti-flotation devices, as necessary. The loads on the anti-flotation devices are to assume cargo tanks empty and the hold spaces flooded.

7.1.5

Where the cargo tanks are located in hold spaces, the void spaces are to be made accessible to enable inspection and examination of the containment pressure boundaries and insulation (if fitted).

7.3 Cargo Tank Protection

7.3.1

Cargo tanks are to be fitted with pressure/vacuum valves, as applicable, to prevent over- or underpressurization. The discharges from the valves from a cargo tank may be led to another cargo tank, provided the cargo tanks are independent of each other and it is not possible to pressurize or vacuum all the tanks simultaneously through a common system. Alternatively, the discharge from the cargo tank valves may be led to the hold bilges.

7.3.2

The setting of the cargo tank pressure/vacuum valve(s) is to be in accordance with 5C-1-7/11.11.2 and 5C-8-1/1.7.1 and the arrangement is to be such that the valve(s) remain connected directly to the cargo tanks at all times, except during maintenance and repair.

7.3.3

For cargo tanks fitted with inerting facilities, see 6-2-14/9.5.2.

9 Cargo Loading and Unloading System

9.1 Cargo Piping

9.1.1

A permanently installed cargo loading and unloading system is to be fitted. There are to be a minimum of two pumps capable of taking suction from each cargo tank. Where submersible pumps are used, only one cargo pump per tank may be used, provided that an alternative method of pumping cargo is available onboard the vessel. This alternative method may be by means of pressurizing the cargo tanks.

9.1.2

Means are to be provided for isolation of each cargo tank in the loading and unloading lines.

9.1.3

Pipes, valves and the fittings in the cargo system are to comply with the requirements of Part 4, Chapter 6.

9.1.4

Cargo loading and unloading lines are to be protected against over pressurization by pressure relief valves. The discharge from the relief valves may be led to the cargo tanks.

9.3 Cargo Pumps

Where the cargo unloading is through cargo pumps other than submersible pumps, they are to be accessible for maintenance and repair.

9.5 Inert Gas System

Where cargo tanks are provided with facilities to supply inert gas into the vapor spaces, the arrangements are to be in accordance with the following requirements:

9.5.1

The location of the inert gas generating plant or the storage of the reserve inert gas is subject to approval by ABS.

9.5.2

The cargo tanks are to be fitted with pressure/vacuum valves to ensure against over- or underpressurization. The outlets from the pressure/vacuum valves are to be situated at least 5 m (16.5 ft.) from any openings and air intakes to the accommodation and service spaces.

11 Refrigeration System

11.1

The refrigeration machinery is to comply with the requirements of Section 6-2-6, as applicable.

11.3

Where a direct expansion system is used whereby the refrigerant is circulated through the cooling coils in the cargo tanks, the design of the coils are to be such as to ensure that there is no possibility of leakage of the refrigerant into the cargo. Details in this regard are to be submitted for review.

11.5

Where an indirect expansion system is used, the secondary coolant must not be detrimental to the cargo.

13 Ancillary Systems

13.1 Cargo Tank Sounding Arrangements

Cargo tanks are to be provided with means for assessing the liquid levels in the tanks. The system may be a permanently fixed or a temporary arrangement.

13.3 Cargo Tank Ventilation

Means for ventilating the cargo tanks during loading and unloading is to be fitted. For tanks supplied with inert gas, refer to 6-2-14/9.5.

13.5 Hold Space Bilge Arrangement

13.5.1

A permanently fixed bilge system is to be provided for emptying out the hold space bilges. This system need not be independent of the Bilge system required by Part 4, Chapter 6.

13.5.2

Where the discharge from the cargo tank relief valves is led to the hold bilges, a bilge high level alarm is to be fitted to give an audible and visual alarm in the engine room or the bridge.

13.7 Hold Space Ventilation Arrangements

The hold spaces are to be provided with adequate ventilation, where applicable.

15 Tests and Inspections

15.1

Tests and inspections of the refrigerating machinery and associated systems are to be in accordance with 6-2-6/25, as applicable.

15.3

Tests and inspection of the vessel and its machinery, other than the refrigeration machinery, are to be in accordance with applicable Sections of the Rules.

PART
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CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

SECTION 15 Refrigerated Fish Carrier

1 General

1.1

The requirements of this section are applicable to fishing vessels defined in 6-2-1/13.29 requiring the notation referred to in 6-2-1/7.1.6.

1.3

Unless otherwise stated in this Section, these Rules and Part 5, Chapter 12 of the *Rules for Building and Classing Steel Vessels Under 90 meters (295 feet) in Length* are applicable.

1.5

Due regard is to be given to the requirements of the Flag State and Port State for the carriage and transportation of edible products.

3 Design Considerations

For design considerations, reference is to be made to the applicable requirements of this Chapter.

5 Materials

Generally, the materials used are to be in accordance with the requirements of Part 2 and the applicable sections of this Chapter.

7 Hull Structures

7.1

For design and construction of the hull structure, refer to the applicable parts of these Rules and the *Rules for Building and Classing Steel Vessels Under 90 meters (295 feet) in Length*, as applicable.

7.3

Where fishing vessels are moored against the mother vessels during unloading at sea, fenders or other similar means for the protection of the shell plating may be required. Where such an arrangement is fitted, the shell plating in way of the protection is to be adequately strengthened.

9 Refrigerated Cargo Spaces

9.1

The refrigerated spaces are to comply with the applicable requirements of Section 6-2-5.

9.3

Equipment and fittings such as electric lights, etc. are to be suitably protected to prevent damage during loading and unloading of cargo.

11 Refrigeration System

11.1

The refrigeration machinery is to comply with the requirements of Section 6-2-6, as applicable.

11.3

Where an ammonia refrigeration system is used, reference is to be made to 6-2-11/1.5.

13 Refrigerated Sea Water Tanks (RSW Tank)

13.1

Each RSW tank is to be provided with appropriate venting and sounding arrangements. The arrangements to assess the liquid levels in the tanks may be permanently installed or a temporary arrangement.

13.3

Where cooling coils are used in the tanks using ammonia as the refrigerant, refer to the requirements of 6-2-15/11.3.

13.5

Where an RSW tank is intended to carry dry fish in bulk, in addition to the requirements for refrigerated spaces, the following arrangements are to be provided:

13.5.1

The tank is to be provided with a bilge well and a permanent connection to the bilge system, unless the tanks are provided with independent bilge systems.

13.5.2

Arrangements are to be made for blanking off sea water piping.

15 Plate Freezers

15.1

Insulation and piping in plate freezers is to be protected from moveable parts of the system.

15.3

Flexible hoses in the system are to be of the armored type suitable for the services intended.

15.5

Piping, including flexible hoses, is to comply with the requirements of 6-2-15/11.1 and 6-2-15/11.3.

17 Tests and Inspections

17.1

The tests and inspections of the refrigeration machinery and associated system is to be in accordance with 6-2-6/25, as applicable.

17.3

Tests and inspection of the vessel and its machinery, other than the refrigeration machinery, are to be in accordance with the applicable Sections of these Rules or the *Rules for Building and Classing Steel Vessels Under 90 meters (295 feet) in Length*, as applicable.

PART

6

CHAPTER 2 Vessels Intended to Carry Refrigerated Cargoes

SECTION 16 Testing

1 On Board Tests After Installation – (Commissioning)

1.1 Piping

1.1.1

All refrigerant and brine piping welded joints are to be hydrostatically tested to a pressure of 1.5 times the respective design pressure. Alternatively, 100 percent nondestructive radiographic or ultrasonic testing of the welded joints may be carried out.

1.1.2

After completion of tests required in 6-2-16/1.1.1, and being completely installed and assembled, but before the application of the insulation, a leak test is to be carried out on the refrigerant and brine systems by use of nitrogen or other suitable gases at pressures not less than the design pressures of the respective systems.

Where defrosting is intended by hot refrigerant gas, the design pressure for the leak test on the low pressure side is to be the same as the high pressure side.

1.1.3

The leak test may be carried out using following methods:

1.1.3(a) By submerging the refrigerant and brine piping and equipment and applying the pressure referred to in 6-2-16/1.1.2.

1.1.3(b) By building up an initial pressure of 0.5 to 1.0 bar (0.5 to 1.0 kgf/cm^2, 7 to 14 psi) in the refrigerant and brine piping systems and checking for leaks at the pressure by either soapy water test, tracer, or detectors. If no leaks are detected or leaks found are dealt with satisfactorily, the pressure is to be increased gradually to the respective design pressures of the systems. The pressure is to be maintained for a predetermined period and pressures deviations are to be recorded.

1.1.3(c) Other alternative, effective methods similar to those described above, subject to the satisfaction of the attending Surveyor.

1.1.4

Following completion of the above-mentioned tests, the refrigerant piping systems are to be flushed with dry nitrogen to ensure dryness and cleanliness.

1.3

Before charging with refrigerant, the entire refrigeration system is to be evacuated using vacuum pumps.

1.5

After completing the pressure tests above, all refrigerant and brine pipes are to be examined under working pressure.

1.7

The refrigeration plant is to be operated to demonstrate its ability to modulate the refrigeration capacity in single and multiple compressor operation with all possible variations in the cross over connections that can be made with compressors, condensers and evaporators.

1.9

Verify operation of thermostats, solenoid valves, expansion valves, bypass valves, evaporator brine line valves and condenser water regulators and other such similar devices.

1.11

Plant safety valves and other similar safety devices are to be verified for satisfactory operation.

1.13

Cooling water flow rates through the condenser are to be measured to determine that the velocities do not exceed the maximum design values while operating with the main cooling water pump and then the standby pumps.

1.15

The satisfactory operation of the automatic or manual oil refrigerant separation system is to be verified to ensure that separated oil is returned to the compressors such that the oil levels between the compressors are balanced.

1.17

After initial startup, the refrigeration monitoring system and the automatic control system, where fitted, is to be verified for satisfactory operation.

1.19

Effective operation of the refrigerant leakage detection system is to be demonstrated.

3 Performance Test

3.1 Air Circulation and Fresh Air Ventilation

3.1.1

All fans for air circulation and fresh air ventilation of cargo spaces are to be tested at the full rated speeds of volumetric flow rates referred to in 6-2-5/9. The testing is to include measurements of pressure difference across the fans and power consumption. The anemometer or other similar measuring devices may be situated on the suction side of the cargo hold. These measuring devices are to be calibrated to the satisfaction of the attending Surveyor.

3.1.2

The air circulation distribution pattern in the refrigerated cargo spaces is to be checked.

3.1.3

The air distribution measurement referred to in 6-2-16/3.1.2 is to be carried out to verify the design values specified by the manufacturers and to ensure that there are no areas of insufficient air flow.

3.3 Refrigeration Machinery and Insulation Test

3.3.1 General

The following refrigeration machinery and insulation tests are to verify that the plant has sufficient refrigeration capacity as required by 6-2-6/3.3 relative to the insulation and other heat loads to achieve and maintain the minimum design temperature, which will be the basis of the notations referred to in 6-2-1/7.

3.3.2 Pull Down Test

Upon completion of the commissioning test referred to in 6-2-16/1, all openings to the cargo spaces including the air freshening vents are to be closed.

The refrigerated cargo hold spaces are to be warmed up to ambient atmospheric temperature by means of running air circulation fans and brine pumps, if fitted.

The refrigeration plant is to be started and run at full capacity under automatic control using all compressors and set at maximum design condensing temperature. The refrigeration machinery should continue to run until the minimum design temperature in all cargo spaces has been achieved. The operation of the refrigeration machinery is to be monitored to ensure satisfactory operation within design parameters.

3.3.3 Heat Balance Test

Upon achieving the minimum design temperature of the refrigerated spaces, after the test specified in 6-2-16/3.3.2, a heat balance test is to be initiated by switching one compressor to manual and remainder switched off and allowing the temperature to stabilize at approximately the minimum design temperature or at least minus 20°C (68°F) and held at these temperatures for a sufficient period of time, generally about 24 hours, to remove the residual heat in the insulation and achieve a balanced condition.

The condition is considered to be balanced when the mean temperature in the refrigerated cargo space does not vary by more than ±0.5°C (±1°F) in each hour. The balanced condition should be planned to be achieved during the time of day when the outside temperature is as constant as possible. During the stabilization period, the collection of data is to be taken initially every six (6) hours and every hour for the last six (6) hours.

For this test, at least the following data are to be recorded:

3.3.3(a) The outside temperatures of the shell, bulkheads and decks enclosing the refrigerated cargo spaces.

3.3.3(b) The internal temperatures of the cargo spaces.

3.3.3(c) The suction and discharge pressure of the compressors.

3.3.3(d) The actual voltage and amperage of the compressor electric motor.

3.3.3(e) Heat inputs to the refrigerated spaces from fan motors, lighting fixtures, heat tracing on drain pumps, etc.

3.3.3(f) The rate of cooling water flowing through the condensers.

3.3.3(g) The inlet and outlet temperatures of the condenser cooling water.

3.3.3(h) Upon achieving stabilized temperatures, calculations of the k values based on the heat balance test mentioned above are to be carried by the yard/builder and submitted to ABS for review. For these calculations, the air cooler overall heat transfer coefficient at the design conditions is to be taken equal to that measured during the heat balance test. Similarly, the condenser overall heat transfer coefficient at the stated maximum sea water temperature is to be taken equal to that measured during the heat balance test.

3.3.4 Refrigerated Port Hole Type

For container carriers described in 6-2-13/3, a full functional test of all refrigerated cargo spaces may not be required if an operational test equivalent to that described herein is performed onboard with at least one cell of containers installed and the following requirements are satisfied:

3.3.4(a) Cooling air to the containers is supplied exclusively by air ducts tested in accordance with 6-2-13/7.

3.3.4(b) The builder demonstrates by calculating, using data obtained during testing described above, to show that the refrigerating machinery has sufficient capacity.

3.3.4(c) It is to be demonstrated that the cell conditioning, if fitted, is sufficient to maintain the cell at a temperature which is in excess of the minimum design temperature of the structural steel.

3.3.5 Insulation Test

After the cargo spaces have been stabilized for the heat balance test in 6-2-16/3.3.3, the outside surfaces of the bulkheads, shell, decks, doors and other opening covers, as well as duct, pipe and cable penetrations are to be checked for excessive condensation or frost indicative of voids and thermal bridges in the insulation.

3.3.6 Temperature Rise Test

For the temperature rise test, the refrigerating machinery is to be stopped and all of the heat input sources shut off after stabilization, as in 6-2-16/3.3.3, and at least the following data are to be recorded once per hour over a six-hour period.

3.3.6(a) The outside temperature of the entire shell enclosing the refrigerated cargo space such as ambient, sea water, tanks, engine room.

3.3.6(b) The internal temperature of the cargo space:

i) The test is to be performed at the time of the day when the outside temperature is as constant as possible.

ii) The calculations of the *k* values is to be carried out by the yard/builder and submitted to ABS for review, together with a drawing showing precise locations and position of the values recorded for this test.

3.3.7 Defrosting Test

After satisfactory completion of the heat balance test and temperature rise test, the cooler batteries are to be defrosted to demonstrate the ability to completely defrost. The Surveyor to verify that the system for removing defrost water is operating satisfactorily.

3.3.8 Multiple Compartment Temperature Test

Where the design parameter specified requires multiple temperature configurations, a test is to be carried out to demonstrate this capability for the refrigerated spaces.

3.3.9 Heating Capacity Test

Where the design parameter specified requires a heating capacity to be available for the refrigerated compartments, a test is to be carried out to demonstrate the capability of the heating system.

3.3.10 Automatic Control System

Where automatic control systems are fitted, the tests referred to under 6-2-16/3.3.2, 6-2-16/3.3.6, 6-2-16/3.3.7 and 6-2-16/3.3.8 are to be conducted utilizing the control system.