Amendment on 27 June 2024
Resolved by Technical Committee on 30 January 2024

Machinery of Polar Class Ships

Object of Amendment

Rules for the Survey and Construction of Steel Ships Part I

Reason for Amendment

IACS Unified Requirements (UR) I3 (Corr.1) specifies requirements for the machinery of Polar Class Ships, and these requirements have already been incorporated into the NK Rules.

However, the UR contains a number of paragraphs that are reserved for future additions to allow for requirements for certain types of machinery to be further discussed, but this has resulted in a lack of requirements for said types of machinery.

For this reason, IACS conducted a comprehensive review of the UR, which included an examination of the design requirements for propeller blades, shafts, propeller lines component, steering systems, etc. and other requirements for icebreakers. In addition, the consistency between various design requirements stipulated in the Finnish-Swedish Ice Class Rules (FSICR) and the UR was considered. As a result of its review, IACS adopted UR I3 (Rev.2) in January 2023.

Accordingly, relevant requirements are amended based on the UR I3 (Rev.2).

Outline of the Amendment

The main contents of this amendment are as follows:

(1) Clarifies classification of materials for machinery.
(2) Clarifies the handling of various design loads.
(3) Clarifies the handling of fatigue design of propeller blades and propulsion shafting systems.
(4) Specifies design requirements for blade bolts, propeller bosses, controllable pitching mechanisms, propeller lines component, starting arrangement for main engine, emergency power supply and ladder actuators for steering systems.

Effective Date and application

This amendment applies to ships for which the date of contract for construction is on or after 1 July 2024.

ID: DD23-31

Amended	Original	Remarks
RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS	**RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS**	
Part I SHIPS OPERATING IN POLAR WATERS, POLAR CLASS SHIPS AND ICE CLASS SHIPS	**Part I SHIPS OPERATING IN POLAR WATERS, POLAR CLASS SHIPS AND ICE CLASS SHIPS**	
ANNEX 1 SPECIAL REQUIREMENTS FOR THE MATERIALS, HULL STRUCTURES, EQUIPMENT AND MACHINERY OF POLAR CLASS SHIPS	**ANNEX 1 SPECIAL REQUIREMENTS FOR THE MATERIALS, HULL STRUCTURES, EQUIPMENT AND MACHINERY OF POLAR CLASS SHIPS**	
Chapter 2 MATERIALS AND WELDING	**Chapter 2 MATERIALS AND WELDING**	
2.1 Material	**2.1 Material**	UR I3(Rev.2) Para. 3
2.1.5 Materials for Machinery Parts Exposed to Sea Water	**2.1.5 Materials for Machinery Parts Exposed to Sea Water**	Para. 3.1
1 Materials exposed to sea water, such as propeller blades, propeller hub and cast thruster bodies are to have an elongation of not less than 15 % for the *U*14*A* test specimen in **Part K of the Rules**.	Materials exposed to sea water, such as propeller blades, propeller hub and blade bolts are to have an elongation of not less than 15 % for the *U*14*A* test specimen in **Part K of the Rules**.	
2 Materials other than bronze and austenitic steel are to have an average impact energy value of 20 *J* at -10 °C for the *U*4 test specimen in **Part K of the Rules**.	Materials other than bronze and austenitic steel are to have an average impact energy value of 20 *J* at -10 °C for the *U*4 test specimen in **Part K of the Rules**.	
3 Materials are also to be in accordance with the requirement in **Chapter 5** and **Chapter 6, Part K of the Rules** that apply to ice class ships.	(Newly added)	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
2.1.6 Materials for Machinery Parts Exposed to Sea Water Temperatures	**2.1.6 Materials for Machinery Parts Exposed to Sea Water Temperatures**	Para. 3.2
1 Except for bronze and austenitic steel, materials exposed to sea water temperatures are to have an average impact energy value of 20 *J* at -10°C for the *U*4 test specimen in **Part K of the Rules**. Materials are also to be in accordance with requirements in **Chapter 5** and **Chapter 6, Part K of the Rules** that apply to ice class ships.	Materials exposed to sea water temperatures are to be of steel or other ductile material approved by the Society. The materials are to have an average impact energy value of 20 *J* at -10°C for the *U*4 test specimen in **Part K of the Rules**.	
2 This **2.1.6** applies to components such as but not limited to blade bolts, controllable pitch mechanisms, shaft bolts, propeller shafts and strut-pod connecting bolts.	(Newly added)	
3 This **2.1.6** does not apply to surface hardened components, such as bearings and gear teeth or sea water cooling lines (heat exchangers, pipes, valves, fittings etc.).	(Newly added)	
4 Definitions for structural boundaries exposed to sea water temperatures are the level of 0.3 *m* below the *LIWL* as specified in **2.1.3**.	(Newly added)	
2.1.7 Materials for Machinery Parts Exposed to Low Air Temperatures	**2.1.7 Materials for Machinery Parts Exposed to Low Air Temperatures**	Para. 3.3
1 Except for bronze and austenitic steel, materials of machinery and foundations exposed to low air temperatures are to be of steel or other ductile materials approved by the Society. The materials are to have an average impact energy value of 20 *J* obtained at 10°C below the lowest design temperature for the *U*4 test specimen in **Part K of the Rules**.	Materials of essential components exposed to low air temperatures are to be of steel or other ductile materials approved by the Society. The materials are to have an average impact energy value of 20 *J* obtained at 10°C below the lowest design temperature for the *U*4 test specimen in **Part K of the Rules**.	
2 This **2.1.7** does not apply to surface hardened components, such as bearings and gear teeth. In addition, definitions for structural boundaries exposed to low air temperatures are the level of 0.3 *m* below the *LIWL* as specified in **2.1.3**.	(Newly added)	

Amended	Original	Remarks
Chapter 4 MACHINERY INSTALLATIONS	**Chapter 4 MACHINERY INSTALLATIONS**	
4.1 General	**4.1 General**	
4.1.1 Scope	**4.1.1 Scope**	Para. 1
1 The requirements of this chapter apply to main propulsion, steering gear, emergency and essential auxiliary systems essential for the safety of the ship and survivability of the crew.	The requirements of this chapter apply to main propulsion, steering gear, emergency and essential auxiliary systems essential for the safety of the ship and survivability of the crew.	
2 Ship operating conditions are to be in accordance with **Chapter 1**.	(Newly added)	
3 This **chapter** applies in additional to requirements applicable to ships operating in open water.	(Newly added)	
4.1.2 Drawings and Data	**4.1.2 Drawings and Data**	Para. 2.1
Drawings and data to be submitted in this chapter are as follows:	Drawings and data to be submitted in this chapter are as follows:	
(1) Details of the intended environmental operational conditions and the required polar class for the machinery, if different from the polar class of hull structure	(1) Details of the environmental conditions and the required polar class for the machinery, if different from the polar class of hull structure	
(2) Detailed drawings and descriptions of the main propulsion , steering, emergency and auxiliary systems (including information on essential main propulsion load control functions)	(2) Detailed drawings of the main propulsion machinery (including information on essential main propulsion load control functions)	
(3) Operational limitations of the main propulsion, steering, emergency and auxiliaries	(3) Operational limitations of the main propulsion, steering, emergency and essential auxiliaries	
(4) Descriptions detailing where main, emergency and auxiliary systems are located and how they are protected to prevent problems from freezing, ice and snow accumulation	(4) Descriptions detailing how main, emergency and auxiliary systems are located and protected to prevent problems from freezing, ice and snow	
(5) Evidence of their capability to operate in intended environmental conditions	(5) Evidence of their capability to operate in intended environmental conditions	

Amended	Original	Remarks
(6) Calculations and documentation indicating compliance with the requirements of this chapter (7) Drawings and data which are deemed necessary by the Society	(6) Calculations and documentation indicating compliance with the requirements of this chapter (7) Drawings and data which are deemed necessary by the Society	
4.1.3 System Design	**4.1.3 System Design**	Para. 2.2
1 Additional fire safety measures are to be arranged in accordance with the requirements in **5.2.3, 7.4, 10.2.1-2, 10.5.3-1** and **10.5.5-2, Part R of the Rules**.	**1** Additional fire safety measures are to be arranged in accordance with the requirements in **5.2.3, 7.4, 10.2.1-2, 10.5.3-1** and **10.5.5-2, Part R of the Rules**.	
2 Any automation plant (control, alarm, safety and indication systems) for essential systems installed is to be maintained in accordance with the requirements in **Chapter 4 of the Rules for Automatic and Remote Control Systems**.	**2** Any automation plant (control, alarm, safety and indication systems) for essential systems installed is to be maintained in accordance with the requirements in **Chapter 4 of the Rules for Automatic and Remote Control Systems**.	
3 Systems subject to damage by freezing are to be drainable.	**3** Systems subject to damage by freezing are to be drainable.	
4 Polar class ships classed *PC*1 to *PC*5 are to have means provided to ensure sufficient vessel operation in the case of propeller damage including a controllable pitch mechanism.	**4** Single screw polar class ships classed *PC*1 to *PC*5 are to have means provided to ensure sufficient vessel operation in the case of propeller damage including a controllable pitch mechanism.	
5 "Sufficient vessel operation" means that vessels are to be able to reach safe haven (i.e. a safe location) where repairs can be undertaken. This may be achieved either by temporary repairs at sea or by towing, assuming either is available under conditions approved by the Society.	(Newly added)	
6 Means are to be provided to free stuck propellers by turning in the reverse direction. This is to also be possible for propulsion plants intended for unidirectional rotation.	(Newly added)	
7 Propellers are to be fully submerged at the *LIWL*.	(Newly added)	

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td>4.2 Materials</td><td>(Newly added)</td><td></td></tr>
<tr><td>4.2.1 General
1 Materials for machinery parts are to be in accordance with 2.1.5, 2.1.6 and 2.1.7.
2 Materials for machinery parts are to be ductile material approved by the Society.
3 Ferritic nodular cast iron may be used for machinery parts other than bolts. In such cases, the values of average absorbed energy at the testing temperatures specified in 2.1.5, 2.1.6 and 2.1.7 are to be applied at 10 <i>J</i>.</td><td>(Newly added)</td><td>Para. 3</td></tr>
<tr><td>4.3 Definitions</td><td>(Newly added)</td><td>Para. 4</td></tr>
<tr><td>4.3.1 Definition of Symbols
Symbols are as defined in Table 4.3.1-1.</td><td>(Newly added)</td><td>Para. 4.1</td></tr>
<tr><td>Table 4.3.1-1 Definition of Symbols
<table>
<tr><th>Symbol</th><th>Unit</th><th>Definition</th></tr>
<tr><td>c</td><td>m</td><td>chord length of blade section</td></tr>
<tr><td>$c_{0.7}$</td><td>m</td><td>chord length of blade section at 0.7<i>R</i> propeller radius</td></tr>
<tr><td>D</td><td>m</td><td>propeller diameter</td></tr>
<tr><td>d</td><td>m</td><td>external diameter of propeller hub (at propeller plane)</td></tr>
<tr><td>d_{pin}</td><td>mm</td><td>diameter of shear pin</td></tr>
<tr><td>D_{limit}</td><td>m</td><td>limit value for propeller diameter</td></tr>
<tr><td>EAR</td><td>-</td><td>expanded blade area ratio</td></tr>
<tr><td>F_b</td><td>kN</td><td>maximum backward blade force for the ship's service life (negative value)</td></tr>
<tr><td>F_{ex}</td><td>kN</td><td>ultimate blade load resulting from blade failure through plastic bending</td></tr>
<tr><td>F_f</td><td>kN</td><td>maximum forward blade force for the ship's service life (positive value)</td></tr>
<tr><td>F_{ice}</td><td>kN</td><td>ice load on propeller blade</td></tr>
</table></td><td>(Newly added)</td><td>Table 1</td></tr>
</table>

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks

$(F_{ice})_{max}$	kN	maximum ice load for the ship's service life
h_0	m	depth of the propeller centreline from lower ice waterline (LIWL)
(H_{ice})	m	ice block dimension for propeller load definition
I	kgm^2	equivalent mass moment of inertia of all parts on engine side of component under consideration
I_t	kgm^2	equivalent mass moment of inertia of the whole propulsion system
k	-	shape parameter for Weibull distribution
m	-	slope for S-N curve in log/log scale
M_{BL}	kNm	blade bending moment
MCR	-	maximum continuous rating
N	-	number of ice load cycles
n	rpm	propeller rotational speed
n_n	rpm	nominal propeller rotational speed at MCR in free running condition
N_{class}	-	reference number of ice impacts per propeller revolution per ice class
N_{ice}	-	total number of ice load cycles on propeller blade for the ship's service life
N_R	-	reference number of ice load cycles for equivalent fatigue stress (10^8 cycles)
N_Q	-	number of propeller revolutions during a milling sequence
$P_{0.7}$	m	propeller pitch at 0.7*R* radius
$P_{0.7n}$	m	propeller pitch at 0.7*R* radius at MCR in free running open water condition
$P_{0.7b}$	m	propeller pitch at 0.7*R* radius at MCR in bollard condition
PCD	m	pitch circle diameter
$Q(\varphi)$	kNm	Torque
Q_{Amax}	kNm	maximum response torque amplitude as a simulation result
Q_{emax}	kNm	maximum engine torque
$Q_F(\varphi)$	kNm	Ice torque excitation for frequency domain calculations
Q_{fr}	kNm	friction torque in pitching mechanism; reduction of spindle torque

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks

Q_{max}	*kNm*	maximum torque on the propeller resulting from propeller/ice interaction
Q_{motor}	*kNm*	electric motor peak torque
Q_n	*kNm*	nominal torque at *MCR* in free running open water condition
$Q_r(t)$	*kNm*	response torque along the propeller shaft line
Q_{peak}	*kNm*	maximum of the response torque $Q_r(t)$
Q_{smax}	*kNm*	maximum spindle torque of the blade for the ship's service life
Q_{sex}	*kNm*	extreme spindle torque corresponding to the blade failure load F_{ex}
Q_{vib}	*kNm*	Vibratory torque at considered component, taken from frequency domain open water Torsional Vibration Calculation (*TVC*)
R	*m*	propeller radius
S	-	safety factor
S_{fat}	-	safety factor for fatigue
S_{ice}	-	ice strength index for blade ice force
r	*m*	blade section radius
T	*kN*	hydrodynamic propeller thrust in bollard condition
T_b	*kN*	maximum backward propeller ice thrust for the ship's service life
T_f	*kN*	maximum forward propeller ice thrust for the ship's service life
T_n	*kN*	propeller thrust at *MCR* in free running condition
T_r	*kN*	maximum response thrust along the shaft line
T_{kmax}	*kNm*	maximum torque capacity of flexible coupling
T_{kmax2}	*kNm*	T_{kmax} at $N = 1$ load cycle
T_{max1}	*kNm*	T_{kmax} at $N = 5 \times 10^4$ load cycle
T_{kv}	*kNm*	vibratory torque amplitude at $N = 10^6$ load cycles
ΔT_{kmax}	*kNm*	maximum range of T_{kmax} at $N=5 \times 10^4$ load cycles
t	*m*	maximum blade section thickness
z	-	number of propeller blades
z_{pin}	-	number of shear pins
α_i	*deg*	duration of propeller blade/ice interaction expressed in

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks

		rotation angle
γ_s	-	the reduction factor for fatigue; scatter and test specimen size effect
γ_v	-	the reduction factor for fatigue; variable amplitude loading effect
γ_m	-	the reduction factor for fatigue; mean stress effect
ρ	-	a reduction factor for fatigue correlating the maximum stress amplitude to the equivalent fatigue stress for 10^8 stress cycles
$\sigma_{0.2}$	MPa	proof yield strength (at 0.2 % plastic strain) of material
σ_{exp}	MPa	mean fatigue strength of blade material at 10^8 cycles to failure in sea water
σ_{fat}	MPa	equivalent fatigue ice load stress amplitude for 10^8 stress cycles
σ_{fl}	MPa	characteristic fatigue strength for blade material
σ_{ref1}	MPa	reference stress $\sigma_{ref1} = 0.6\sigma_{0.2} + 0.4\sigma_u$
σ_{ref2}	MPa	reference stress whichever is less $\sigma_{ref2} = 0.7\sigma_u$ or $\sigma_{ref2} = 0.6\sigma_{0.2} + 0.4\sigma_u$
σ_{st}	MPa	maximum stress resulting from F_b or F_f
σ_u	MPa	ultimate tensile strength of blade material
$(\sigma_{ice})_{bmax}$	MPa	principal stress caused by the maximum backward propeller ice load
$(\sigma_{ice})_{fmax}$	MPa	principal stress caused by the maximum forward propeller ice load
σ_{mean}	MPa	mean stress
$(\sigma_{ice})_A(N)$	MPa	blade stress amplitude distribution

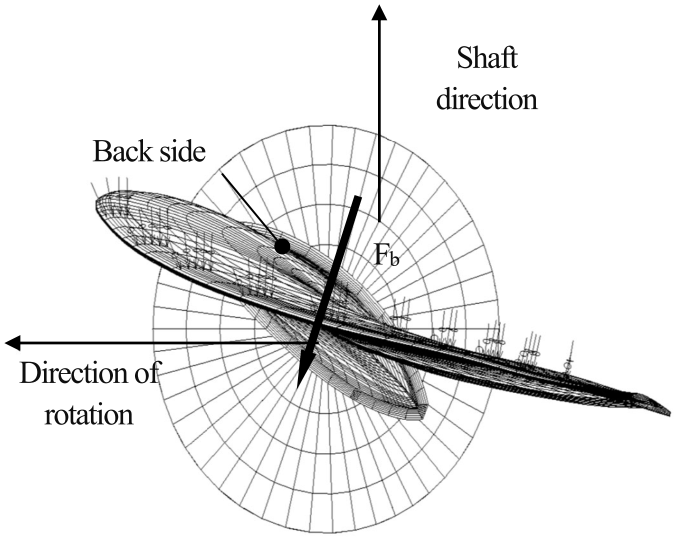
<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr>
<td>
4.3.2 Definition of Loads
Loads are as defined in Table 4.3.2-1.
Table 4.3.2-1 Definition of Loads
<table>
<tr><th></th><th>Definition</th><th>Use of the load in design process</th></tr>
<tr><td>F_b</td><td>The maximum lifetime backward force on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to 0.7R chord line. (See Fig. 4.3.2-1)</td><td>Design force for strength calculation of the propeller blade.</td></tr>
<tr><td>F_f</td><td>The maximum lifetime forward force on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to 0.7R chord line.</td><td></td></tr>
<tr><td>Q_{smax}</td><td>The maximum lifetime spindle torque on a propeller blade resulting from propeller/ice interaction, including hydrodynamic loads on that blade.</td><td>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.</td></tr>
<tr><td>T_b</td><td>The maximum lifetime thrust on propeller (all blades) resulting from propeller/ice interaction. The direction of the thrust is the propeller shaft direction and the force is opposite to the hydrodynamic thrust.</td><td>Is used for estimation of the response thrust T_r. T_b and T_f can be used as an estimate of excitation for axial vibration calculations. However, axial vibration calculations are not required in the rules.</td></tr>
<tr><td>T_f</td><td>The maximum lifetime thrust on propeller (all blades) resulting from propeller/ice interaction. The direction of the thrust is the propeller shaft direction acting in the direction of</td><td></td></tr>
</table>
</td>
<td>(Newly added)

(Newly added)</td>
<td>Para 4.2

Table 2</td>
</tr>
</table>

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

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

Amended	Original	Remarks
Fig. 4.3.2-1 Direction of the Backward Blade Force resultant taken Perpendicular to the Chord Line at Radius 0.7R. (Ice contact pressure at leading edge is shown with small arrows.)  	(Newly added)	Figure 1
4.4 Design Loads **4.4.1 General** **1** In the design of the propeller, propulsion shafting system and power transmission system, the following are to be taken into account. (1) Maximum backward blade force (2) Maximum forward blade force (3) Maximum blade spindle torque (4) Maximum propeller ice torque (5) Maximum propeller ice thrust (6) Design torque on propulsion shafting system	**4.2 Design Loads** **4.2.1 General** **1** In the design of the propeller, propulsion shafting system and power transmission system, the following are to be taken into account. (1) Maximum backward blade force (2) Maximum forward blade force (3) Maximum blade spindle torque (4) Maximum propeller ice torque (5) Maximum propeller ice thrust (6) Design torque on propulsion shafting system	Para. 5

Amended	Original	Remarks
(7) Maximum thrust on propulsion shafting system	(7) Maximum thrust on propulsion shafting system	
(8) Blade failure load	(8) Blade failure load	
2 The loads specified in **-1** are to comply with the following:	**2** The loads specified in **-1** are to comply with the following:	Para. 5.1
(1) The ice loads cover open and ducted type propellers situated at the stern of a ship having controllable pitch or fixed pitch blades. Ice loads on bow mounted propellers are to receive special consideration.	(1) The ice loads cover open and ducted type propellers situated at the stern of a ship having controllable pitch or fixed pitch blades. Ice loads on bow propellers and pulling type propellers are to receive special consideration.	
(2) The given loads in this chapter are expected, single occurrence, maximum values for the whole ships service life for normal operation conditions, including loads resulting from directional change of rotation where applicable. The loads do not cover off-design operational conditions, for example when a stopped propeller is dragged through ice.	(2) The given loads in this chapter are expected, single occurrence, maximum values for the whole ships service life for normal operation conditions The loads do not cover off-design operational conditions, for example when a stopped propeller is dragged through ice.	
(3) The loads apply also for propeller ice interaction for azimuthing and fixed thrusters with geared transmissions or integrated electric motors (i.e. "geared and podded propulsors"). However, such load models do not include propeller/ice interaction loads when ice enters the propeller of turned azimuthing thrusters from the side (i.e. radially) or loads when ice blocks hit the propeller hubs of pulling propellers. ice loads resulting from ice impacts on the body of azimuthing thrusters are to be estimated on a case by case basis.	(3) The loads apply also for azimuthing (geared and podded) thrusters considering loads due to propeller ice interaction. However, ice loads due to ice impacts on the body of azimuthing thrusters are not covered by this chapter.	
(4) The loads are total loads including ice-induced loads and hydrodynamic loads (unless otherwise stated) during ice interaction that are to be applied separately (unless otherwise stated) and are intended for component strength calculations only.	(4) The loads are total loads (unless otherwise stated) during interaction and are to be applied separately (unless otherwise stated) and are intended for component strength calculations only.	
(5) The load specified in **-1(1)** above is the maximum force experienced during the lifetime of the ship that bends	(Newly added)	

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td>propeller blades backwards when propellers mill ice blocks while rotating ahead. The load specified in -1(2) above is the maximum force experienced during the lifetime of the ship that bends propeller blades forwards when propellers mill ice blocks while rotating ahead. Since these loads originate from different propeller/ice interaction phenomena, which do not act simultaneously, they are to be applied separately.</td><td></td><td></td></tr>
<tr><td>4.4.2 Polar Class Factors
1 The ice thickness H_{ice} (m) and the ice strength index S_{ice} for each polar class are to be taken as specified in Table 4.4.2-1.
2 The design ice block to be considered is to be obtained by the following formula:
$$H_{ice} \times 2H_{ice} \times 3H_{ice} \ \ (m)$$
3 The design ice block H_{ice} and ice strength index S_{ice} are to be used for the estimation of propeller ice loads.</td><td>(Newly added)</td><td>Para. 5.2</td></tr>
<tr><td>Table 4.4.2-1 Values of H_{ice} and S_{ice}
<table>
<tr><th>Polar class</th><th>H_{ice}</th><th>S_{ice}</th></tr>
<tr><td>PC1</td><td>4.0</td><td>1.2</td></tr>
<tr><td>PC2</td><td>3.5</td><td>1.1</td></tr>
<tr><td>PC3</td><td>3.0</td><td>1.1</td></tr>
<tr><td>PC4</td><td>2.5</td><td>1.1</td></tr>
<tr><td>PC5</td><td>2.0</td><td>1.1</td></tr>
<tr><td>PC6</td><td>1.75</td><td>1</td></tr>
<tr><td>PC7</td><td>1.5</td><td>1</td></tr>
</table></td><td>(Newly added)</td><td>Table 3</td></tr>
</table>

Amended	Original	Remarks
4.4.3 Maximum Backward Blade Force **1** The maximum backward blade force which bends a propeller blade backwards when a propeller mills an ice block while rotating ahead is to be given by the following formulae: (1) For open propellers: when $D < D_{\text{limit}}$ $F_b = 27S_{ice}\left(\frac{n}{60}D\right)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3}D^2$ (*kN*) when $D \geq D_{\text{limit}}$ $F_b = 23S_{ice}(H_{ice})^{1.4}\left(\frac{n}{60}D\right)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3}D$ (*kN*) where $D_{\text{limit}} = 0.85(H_{ice})^{1.4}$ (*m*) (2) For ducted propellers : when $D < D_{\text{limit}}$ $F_b = 9.5S_{ice}\left(\frac{n}{60}D\right)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3}D^2$ (*kN*) when $D \geq D_{\text{limit}}$ $F_b = 66S_{ice}(H_{ice})^{1.4}\left(\frac{n}{60}D\right)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3}D^{0.6}$ (*kN*) where $D_{\text{limit}} = 4H_{ice}$ (*m*) (Deleted) (Deleted) *D* : Propeller diameter (*m*) *EAR* : Expanded blade area ratio *n* : Nominal rotational propeller speed (*rpm*) at maximum continuous revolutions in free running condition for controllable pitch propellers and 85 % of the nominal rotational propeller speed at maximum continuous revolutions in free running	**4.2.2 Maximum Backward Blade Force** **1** The maximum backward blade force which bends a propeller blade backwards when a propeller mills an ice block while rotating ahead is to be given by the following formulae: (1) For open propellers: when $D < D_{\text{limit}}$ $F_b = 27S_{ice}\left(\frac{n}{60}D\right)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3}D^2$ (*kN*) when $D \geq D_{\text{limit}}$ $F_b = 23S_{ice}(H_{ice})^{1.4}\left(\frac{n}{60}D\right)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3}D$ (*kN*) where $D_{\text{limit}} = 0.85(H_{ice})^{1.4}$ (*m*) (2) For ducted propellers : when $D < D_{\text{limit}}$ $F_b = 9.5S_{ice}\left(\frac{n}{60}D\right)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3}D^2$ (*kN*) when $D \geq D_{\text{limit}}$ $F_b = 66S_{ice}(H_{ice})^{1.4}\left(\frac{n}{60}D\right)^{0.7}\left(\frac{EAR}{Z}\right)^{0.3}D^{0.6}$ (*kN*) where $D_{\text{limit}} = 4H_{ice}$ (*m*) H_{ice} : Ice thickness (*m*) for machinery strength design specified in **Table 4.2.2-1**. S_{ice} : Ice strength index for blade ice force specified in **Table 4.2.2-1**. *D* : Propeller diameter (*m*) *EAR* : Expanded blade area ratio *n* : Nominal rotational propeller speed (*rpm*) at maximum continuous revolutions in free running condition for controllable pitch propellers and 85 % of the nominal rotational propeller speed at maximum continuous revolutions in free running	Para. 5.3.1 Para. 5.3.4

Amended	Original	Remarks
condition for fixed pitch propellers (regardless of driving engine type)	condition for fixed pitch propellers (regardless of driving engine type)	
2 For ships affixed with the additional notation "*Icebreaker*" (abbreviated to *ICB*), the maximum backward blade force F_b specified in **-1** above is to be multiplied by a factor of 1.1.	(Newly added)	
3 The maximum backward blade force F_b is to be applied as a uniform pressure distribution to an area of the blade for the following load cases as specified in **Table 4.4.5-1** and **Table 4.4.5-2**.	**2** The maximum backward blade force F_b is to be applied as a uniform pressure distribution to an area of the blade for the following load cases.	
(1) For open propellers:	(1) For open propellers:	
(a) Load case 1 in **Table 4.4.5-1**	(a) F_b specified in **-1(1)** is applied to an area from $0.6R$ to the tip and from the blade leading edge to a value of 0.2 chord length. (See load case 1 in **Table 4.2.2-2**)	
(b) load case 2 in **Table 4.4.5-1**	(b) A load equal to 50% of F_b specified in **-1(1)** is applied on the propeller tip area outside of $0.9R$. (See load case 2 in **Table 4.2.2-2**)	
(c) For reversible propellers, Load case 5 in **Table 4.4.5-1**	(c) For reversible propellers, a load equal to 60% of F_b specified in **-1(1)** is applied to an area from $0.6R$ to the tip and from the blade trailing edge to a value of 0.2 chord length. (See load case 5 in **Table 4.2.2-2**)	
(2) For ducted propellers:	(2) For ducted propellers:	
(a) Load case 1 in **Table 4.4.5-2**	(a) F_b specified in **-1(2)** is applied to an area from $0.6R$ to the tip and from the blade leading edge to a value of 0.2 chord length. (See load case 1 in **Table 4.2.2-3**)	
(b) For reversible propellers,Load case 5 in **Table 4.4.5-2**	(b) For reversible propellers, a load equal to 60 % of the F_b specified in **-1(2)** is applied to an area from $0.6R$ to the tip and from the blade trailing edge to a value of 0.2 chord length. (See load case 5 in **Table 4.2.2-3**)	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr>
<td>(Deleted)</td>
<td>Table 4.2.2-1 Values of H_{ice} and S_{ice}
<table>
<tr><th>Polar class</th><th>H_{ice}</th><th>S_{ice}</th></tr>
<tr><td>PC1</td><td>4.0</td><td>1.2</td></tr>
<tr><td>PC2</td><td>3.5</td><td>1.1</td></tr>
<tr><td>PC3</td><td>3.0</td><td>1.1</td></tr>
<tr><td>PC4</td><td>2.5</td><td>1.1</td></tr>
<tr><td>PC5</td><td>2.0</td><td>1.1</td></tr>
<tr><td>PC6</td><td>1.75</td><td>1</td></tr>
<tr><td>PC7</td><td>1.5</td><td>1</td></tr>
</table>
</td>
<td>Relocation to Table 4.4.2-1</td>
</tr>
<tr>
<td>

4.4.4 Maximum Forward Blade Force

1 Maximum forward blade force which bends a propeller blade forwards when a propeller interacts with an ice block while rotating ahead is to be given by the following formulae:

(1) For open propellers:

when $D < D_{\text{limit}}$

$$F_f = 250\left(\frac{EAR}{Z}\right)D^2 \quad (kN)$$

when $D \geq D_{\text{limit}}$

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

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

(2) For ducted propellers:

when $D \leq \mathrm{D}_{\text{limit}}$

$$F_f = 250\left(\frac{EAR}{Z}\right)D^2 \quad (kN)$$

when $D > D_{limit}$

</td>
<td>

4.2.3 Maximum Forward Blade Force

1 Maximum forward blade force which bends a propeller blade forwards when a propeller interacts with an ice block while rotating ahead is to be given by the following formulae:

(1) For open propellers:

when $D < D_{\text{limit}}$

$$F_f = 250\left(\frac{EAR}{Z}\right)D^2 \quad (kN)$$

when $D \geq D_{\text{limit}}$

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

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

(2) For ducted propellers:

when $D \leq \mathrm{D}_{\text{limit}}$

$$F_f = 250\left(\frac{EAR}{Z}\right)D^2 \quad (kN)$$

when $D > D_{limit}$

</td>
<td>Para. 5.3.2
Para. 5.3.5</td>
</tr>
</table>

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
$F_f = 500H_{ice}\left(\frac{EAR}{Z}\right)\left(\frac{1}{1-\frac{d}{D}}\right)D$ (kN)	$F_f = 500H_{ice}\left(\frac{EAR}{Z}\right)\left(\frac{1}{1-\frac{d}{D}}\right)D$ (kN)	
where $D_{limit} = \frac{2}{\left(1-\frac{d}{D}\right)}H_{ice}$ (m)	where $D_{limit} = \frac{2}{\left(1-\frac{d}{D}\right)}H_{ice}$ (m)	
(Deleted) d : Propeller boss diameter (m) Z : Number of propeller blades	H_{ice}, D and EAR : As specified in **4.2.2-1**. d : Propeller boss diameter (m) Z : Number of propeller blades	
2 The maximum forward blade force F_f is to be applied as a uniform pressure distribution to an area of the blade for the following load cases as specified in **Table 4.4.5-1** and **Table 4.4.5-2**.	**2** The maximum forward blade force F_f is to be applied as a uniform pressure distribution to an area of the blade for the following load cases.	
(1) For open propellers:	(1) For open propellers:	
(a) Load case 3 in **Table 4.4.5-1**	(a) F_f specified in **-1(1)** is applied to an area from 0.6R to the tip and from the blade leading edge to a value of 0.2 chord length. (See load case 3 in **Table 4.2.2-2**)	
(b) Load case 4 in **Table 4.4.5-1**	(b) A load equal to 50% of F_f specified in **-1(1)** is applied on the propeller tip area outside of 0.9R. (See load case 4 in **Table 4.2.2-2**)	
(c) For reversible propellers, Load case 5 in **Table 4.4.5-1**	(c) For reversible propellers, a load equal to 60% of F_f specified in **-1(1)** is applied to an area from 0.6R to the tip and from the blade trailing edge to a value of 0.2 chord length. (See load case 5 in **Table 4.2.2-2**)	
(2) For ducted propellers:	(2) For ducted propellers:	
(a) Load case 3 in **Table 4.4.5-2**	(a) F_f specified in **-1(2)** is applied to an area from 0.6R to the tip and from the blade leading edge to a value of 0.5 chord length. (See load case 3 in **Table 4.2.2-3**)	
(b) For reversible propellers, Load case 5 in **Table 4.4.5-2**	(b) For reversible propellers, a load equal to 60% of F_f specified in **-1(2)** is applied to an area from 0.6R to the tip and from the blade trailing edge to a value of 0.2 chord length. (See load case 5 in **Table 4.2.2-3**)	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

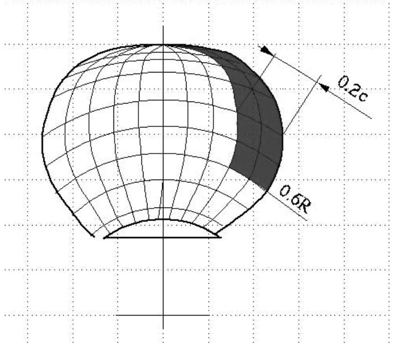
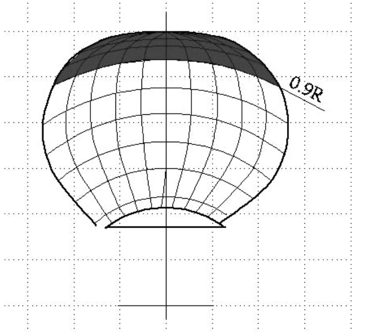
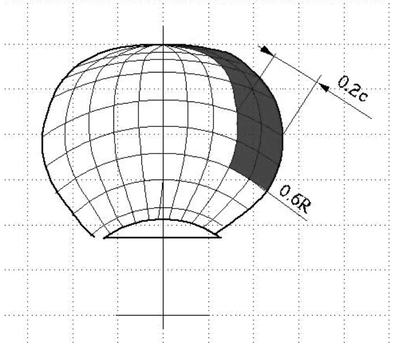
Amended	Original	Remarks
4.4.5 Loaded area on the blade **1** Loaded area on the blade of the Maximum forward blade force and maximum backward blade force are to be in accordance with **Table 4.4.5-1** and **Table 4.4.5-2.**	**(Newly added)** (Newly added)	Para. 5.3.3

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
Table 4.5.5-1 Load Cases for Open Propeller	Table 4.2.2-2 Load Cases for Open Propeller	Table 4

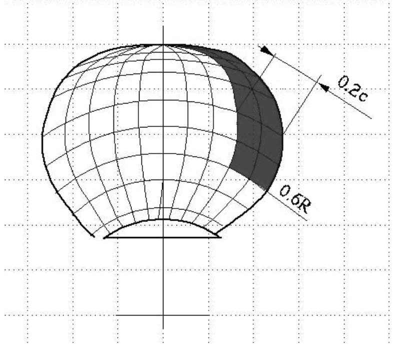
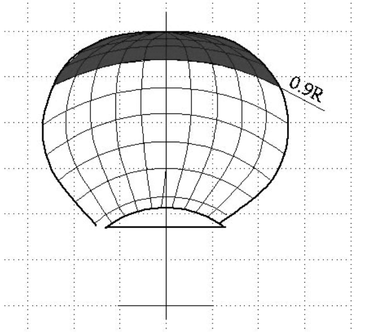
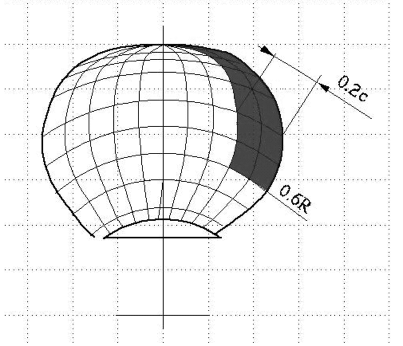
Amended

Table 4.5.5-1 Load Cases for Open Propeller

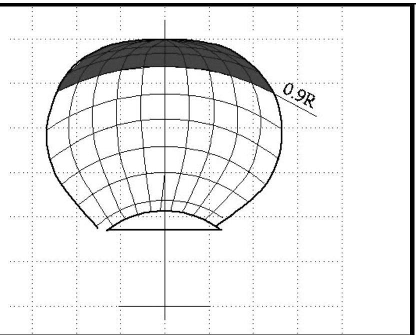
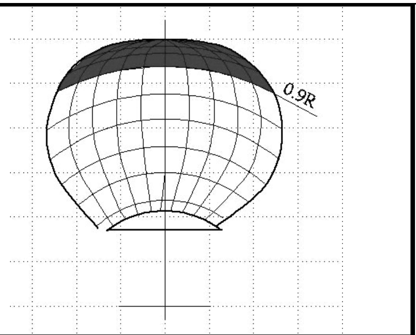
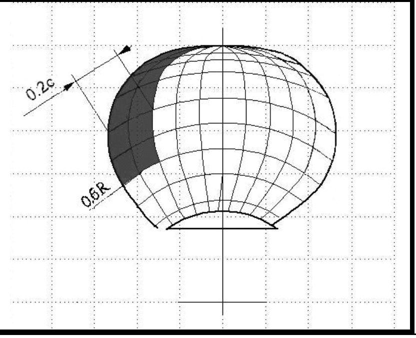
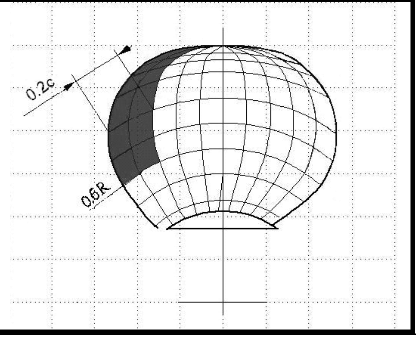
	Force	Loaded area	Right handed propeller blade seen from back
Load case 1	F_b	Uniform pressure applied on the back of the blade (suction side) to an area from $0.6R$ to the tip and from the leading edge to 0.2 times the chord length	
Load case 2	50% of F_b	Uniform pressure applied on the back of the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

Original

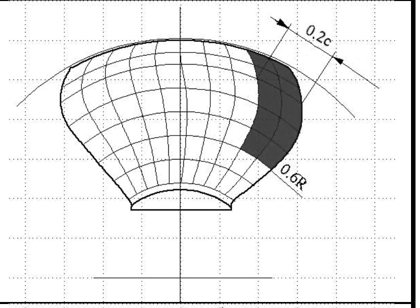
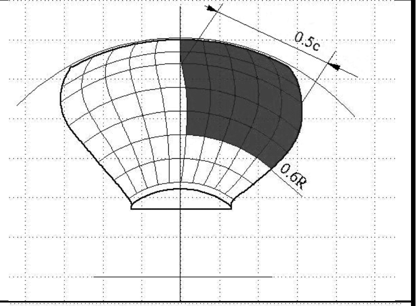
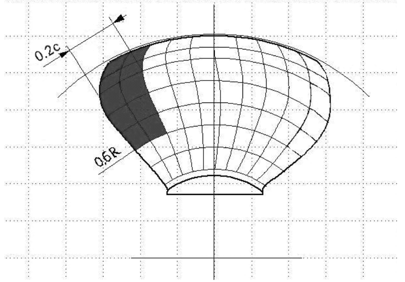
Table 4.2.2-2 Load Cases for Open Propeller

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

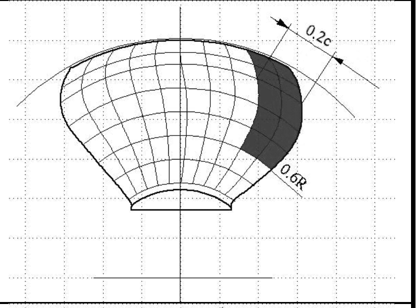
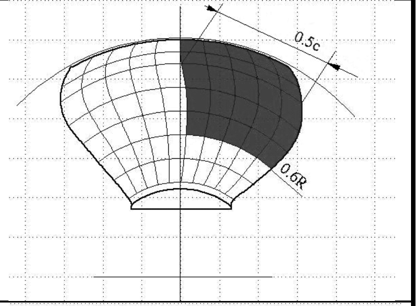
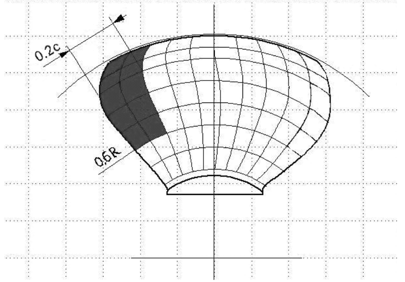
Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended				Original				Remarks
Load case 4	50% of F_f	Uniform pressure applied on the propeller face (pressure side) on the propeller tip area outside of $0.9R$ radius		Load case 4	50% of F_f	Uniform pressure applied on the propeller face (pressure side) on the propeller tip area outside of $0.9R$ radius		
Load case 5	60% of F_f or F_b which one is greater	Uniform pressure applied on propeller face (pressure side) to an area from $0.6R$ to the tip and from the trailing edge to 0.2 times the chord length		Load case 5	60% of F_f or F_b which one is greater	Uniform pressure applied on propeller face (pressure side) to an area from $0.6R$ to the tip and from the trailing edge to 0.2 times the chord length		

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr>
<td>
Table 4.5.5-2 Load Cases for Ducted Propeller
<table>
<tr><th></th><th>Force</th><th>Loaded area</th><th>Right handed propeller blade seen from back</th></tr>
<tr><td>Load case 1</td><td>F_b</td><td>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</td><td> </td></tr>
<tr><td>Load case 3</td><td>F_f</td><td>Uniform pressure applied on the blade face (pressure side) to an area from the leading edge to 0.5 times the chord length</td><td> </td></tr>
<tr><td>Load case 5</td><td>60% of F_f or F_b which one is greater</td><td>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</td><td> </td></tr>
</table>

4.4.6 Maximum Blade Spindle Torque

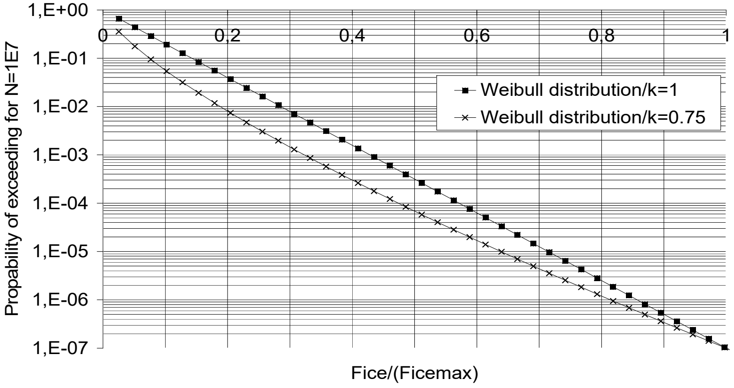
Spindle torque around the spindle axis of the blade fitting is to be calculated both for the load case specified in **4.4.3** and **4.4.4** for
</td>
<td>
Table 4.2.2-3 Load Cases for Ducted Propeller
<table>
<tr><th></th><th>Force</th><th>Loaded area</th><th>Right handed propeller blade seen from back</th></tr>
<tr><td>Load case 1</td><td>F_b</td><td>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</td><td> </td></tr>
<tr><td>Load case 3</td><td>F_f</td><td>Uniform pressure applied on the blade face (pressure side) to an area from the leading edge to 0.5 times the chord length</td><td> </td></tr>
<tr><td>Load case 5</td><td>60% of F_f or F_b which one is greater</td><td>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</td><td> </td></tr>
</table>

4.2.4 Maximum Blade Spindle Torque

Spindle torque around the spindle axis of the blade fitting is to be calculated both for the load case specified in **4.2.2** and **4.2.3** for
</td>
<td>Table 5
Para. 5.3.7</td>
</tr>
</table>

Amended	Original	Remarks
F_b and F_f. Where these spindle torque values are less than the default value obtained from the following formula, the default value is to be used. $Q\_smax = 0.25FC\_0.7$ (kNm) where $C_{0.7}$: Length (m) of the blade chord at $0.7R$ radius F : Either F_b determined in **4.4.3-1** or F_f determined in **4.4.4-1,** whichever has the greater absolute value.	F_b and F_f. Where these spindle torque values are less than the default value obtained from the following formula, the default value is to be used. $Q\_smax = 0.25FC\_0.7$ (kNm) where: $C_{0.7}$: Length (m) of the blade chord at $0.7R$ radius F : Either F_b determined in **4.2.2-1** or F_f determined in **4.2.3-1,** whichever has the greater absolute value.	
4.4.7 Frequent Distributions for Propeller Blade Loads **1** A Weibull-type distribution (probability that F_{ice} exceeds $(F_{ice})_{max}$), as given in **Fig. 4.4.7-1**, is to be used for the fatigue design of blades. $P\left(\frac{F_{ice}}{(F_{ice})_{max}} \geq \frac{F}{(F_{ice})_{max}}\right) = \exp\left(-\left(\frac{F}{(F_{ice})_{max}}\right)^k \ln(N_{ice})\right)$ where F_{ice}: Random variable for ice loads (kN) on the blade that satisfies $0 \leq F_{ice} \leq (F_{ice})_{max}$ $(F_{ice})_{max}$: Maximum ice load for ship's service life (kN) k: Shape parameter for the Weibull-type distribution in witch the following definitions apply: Open propeller: k =0.75 Ducted propeller: k =1.0 N_{ice}: Total number of ice loads on propeller blade for ship's service life	**(Newly added)** (Newly added)	Para. 5.3.8

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
2 A blade stress amplitude distribution in accordance with -1 above is to be determined in accordance with the formula: $(\sigma_{ice})_A(N) = (\sigma_{ice})_{Amax} \cdot \left(1 - \frac{\log(N)}{\log(N_{ice})}\right)^{\frac{1}{k}}$ where $(\sigma_{ice})_{Amax} = \frac{(\sigma_{ice})_{fmax} - (\sigma_{ice})_{bmax}}{2}$	(Newly added)	
Fig.4.4.7-1 Weibull-type Distribution (probability that F_{ice} exceeds $(F_{ice})_{max}$) Used for Fatigue Designs  	(Newly added)	Figure 2

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td>

4.4.8 Number of Ice Loads

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

$$N_{ice} = k_1 k_2 N_{class} \frac{n_n}{60}$$

where

N_{class}: Reference number of loads for ice classes, as specified in **Table 4.4.8-1**

n_n: Nominal propeller rotational speed at maximum continuous revolutions in free running condition (rpm)

k_1: Propeller location factor, as specified in **Table 4.4.8-2**

k_2: The submersion factor k_2 is determined from the following equation.

$$k_2 = \begin{array}{ll} 0.8 - f & : f < 0 \\ 0.8 - 0.4f & : 0 \leq f \leq 1 \\ 0.6 - 0.2f & : 1 < f \leq 2.5 \\ 0.1 & : f > 2.5 \end{array}$$

where

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

h_0: The depth of the propeller centreline at the lower ice waterline ($LIWL$) of the ship (m). If h_0 is not known, $h_0 = 2/D$.

</td><td>

(Newly added)

(Newly added)

</td><td>Para 5.3.9</td></tr>
<tr><td>

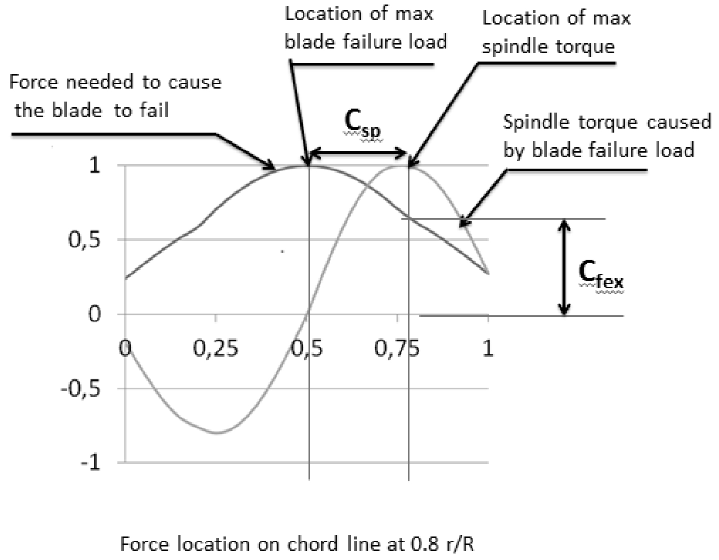
Table 4.4.8-1 Reference Number of Loads for Polar Classes N_{class}

<table>
<tr><th>Polar class</th><th>PC1</th><th>PC2</th><th>PC3</th><th>PC4</th><th>PC5</th><th>PC6</th><th>PC7</th></tr>
<tr><td>N_{class}</td><td>$21 \cdot 10^6$</td><td>$17 \cdot 10^6$</td><td>$15 \cdot 10^6$</td><td>$13 \cdot 10^6$</td><td>$11 \cdot 10^6$</td><td>$9 \cdot 10^6$</td><td>$6 \cdot 10^6$</td></tr>
</table>

</td><td>(Newly added)</td><td>Table 6</td></tr>
</table>

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td>Table 4.4.8-2 Propeller Location Factor k_l
<table>
<tr><th>Factor</th><th>Centre propeller
Bow first operation</th><th>Wing propeller
Bow first operation</th><th>Pulling propeller (wing and centre)
Bow propeller or Stern first operation</th></tr>
<tr><td>k_l</td><td>1</td><td>2</td><td>3</td></tr>
</table></td><td>(Newly added)</td><td></td></tr>
<tr><td>2 For ships is affixed with the additional notation “<i>Icebreaker</i>” (abbreviated to <i>ICB</i>), the number of loads specified in -1 above is to be multiplied by a factor of 3.</td><td>(Newly added)</td><td></td></tr>
<tr><td>3 For components that are subject to loads resulting from propeller/ice interaction with the propeller blades, the number of load cycles (N_{class}) is to be multiplied by the number of propeller blades (Z).</td><td>(Newly added)</td><td></td></tr>
<tr><td>4.4.9 Blade Failure Load</td><td>4.2.9 Blade Failure Load</td><td>Para. 5.4</td></tr>
<tr><td>1 Bending Force, F_{ex}
(1) Bending force is to be obtained by the following formula:
$F_{ex} = \frac{0.3ct^2\sigma_{ref}}{0.8D-2r} \times 10^3$ (<i>kN</i>)
where
$\sigma_{ref1} = 0.6\sigma_{0.2} + 0.4\sigma_u$ (<i>MPa</i>)
σ_u: Minimum tensile stress of blade material (<i>MPa</i>)
$\sigma_{0.2}$: Minimum yield stress or 0.2 % proof strength of blade material (<i>MPa</i>)
<i>c</i>, <i>t</i> and <i>r</i>: The actual chord length, thickness and radius of the cylindrical root section of the blade at the weakest section outside root fillet (typically will be at the termination of the fillet into the blade profile), respectively</td><td>1 The blade failure load is to be given by the following formula:
$\frac{0.3ct^2\sigma_{ref}}{0.8D-2r} \times 10^3$ (<i>kN</i>)
where
$\sigma_{ref} = 0.6\sigma_{0.2} + 0.4\sigma_u$ (<i>MPa</i>)
σ_u: Ttensile stress of blade material (<i>MPa</i>)
$\sigma_{0.2}$: Yield stress or 0.2 % proof strength of blade material (<i>MPa</i>)
<i>c</i>, <i>t</i> and <i>r</i>: The actual chord length, thickness and radius of the cylindrical root section of the blade at the weakest section outside root fillet (typically will be at the termination of the fillet into the blade profile), respectively</td><td>Para. 5.4.1</td></tr>
<tr><td>(2) The bending force in (1) above is the minimum load required resulting in blade failure through plastic bending,</td><td>2 The force is to be acting at 0.8<i>R</i> in the weakest direction of the blade and at a spindle arm of 2/3 the distance of the axis</td><td></td></tr>
</table>

Amended	Original	Remarks
and is to be calculated iteratively along the radius of the blade root to 0.5R assumed to be acting at 0.8R in the weakest direction. (3) Blade failure loads may be obtained alternatively by means of an appropriate stress analysis, reflecting the non-linear plastic material behaviour of the actual blade. In such cases, blade failure areas may be outside root sections. Blades are regarded as having failed if their tips are bent into offset positions by more than 10 % of the propeller diameter D. **2** Spindle Torque, Q_{sex} (1) The maximum spindle torque due to a blade failure load acting at 0.8R is to be determined. The force that causes blade failure typically reduces when moving from the propeller centre towards the leading and trailing edges, and maximum spindle torque will occur at certain distances from the blade centre of rotation. This maximum spindle torque is to be defined by an appropriate stress analysis or by using the following equation: $Q_{sex} = \max(C_{LE0.8}; 0.8C_{TE0.8})C_{spex}F_{ex}$ (kNm) where $C_{spex} = C_{sp}C_{fex} = 0.7\left(1-\left(\frac{4EAR}{z}\right)^3\right)$ C_{sp}: Non-dimensional parameter taking into account the spindle arm C_{fex}: Non-dimensional parameter taking into account the reduction of blade failure force at the location of maximum spindle torque $C_{LE0.8}$: Leading edge portion of the chord length at 0.8R	of blade rotation of the leading and trailing edge whichever is greater. (Newly added) (Newly added)	Para. 5.4.2

Amended	Original	Remarks
$C_{TE0.8}$: Trailing edge portion of the chord length at 0.8R If C_{spex} is below 0.3, a value of 0.3 is to be used for C_{spex}. (2) Spindle torque values due to blade failure loads across the entire chord length area are shown in **Fig. 4.4.9-1**. Fig.4.4.9-1 Schematic figure showing blade failure loads and related spindle torques when force acts at different locations on chord lines at radius 0.8R 	(Newly added)	Figure 3
4.4.10 Maximum Propeller Ice Thrust Maximum propeller ice torque applied to the shaft is to be given by the following formula. However, the propeller/ice interaction loads where an ice blocks hit the propeller hubs of pulling propellers are not to be included.	**4.2.6 Maximum Propeller Ice Thrust** Maximum propeller ice torque applied to the shaft is to be given by the following formula.	Para. 5.5.1

Amended	Original	Remarks
(1) Maximum forward propeller ice thrust $T_f = 1.1\ F_f\ (kN)$ (2) Maximum backward propeller ice thrust $T_b = 1.1\ F_b\ (kN)$ where F_f: As determined in **4.2.3-1** F_b: As determined in **4.2.2-1**	(1) Maximum forward propeller ice thrust $T_f = 1.1\ F_f\ (kN)$ (2) Maximum backward propeller ice thrust $T_b = 1.1\ F_b\ (kN)$ where F_f: As determined in **4.2.3-1** F_b: As determined in **4.2.2-1**	
4.4.11 Design Thrust along Propulsion Shaft Lines	**4.2.8 Maximum Thrust on Propulsion Shafting System**	Para. 5.5.2
1 Design thrust along the propeller shaft line is to be given by the following formulae. The greater value of the forward and backward directional load is to be taken as the design load for both directions.	Maximum response thrust along the propeller shaft line is to be given by the following formulae.	
(1) Maximum shaft thrust forwards: $T_r = T_n + 2.2T_f \quad (kN)$ (2) Maximum shaft thrust backwards: $T_r = 1.5T_b \quad (kN)$ where: T: Propeller bollard thrust (kN) If not known, T is to be taken as specified in **Table 4.4.11-1** T_f and T_b: Maximum propeller ice thrust (kN) determined in **4.4.10** 2.2 and 1.5: Thrust magnification factors due to axial vibration	(1) Maximum shaft thrust forwards: $T_r = T_n + \alpha 2.2T_f \quad (kN)$ (2) Maximum shaft thrust backwards: $T_r = \beta 1.5T_b \quad (kN)$ where: T_n: Propeller bollard thrust (kN) If not known, T_n is to be taken as specified in **Table 4.2.8-1** T_f and T_b: Maximum propeller ice thrust (kN) determined in **4.2.6** α and β: Thrust magnification factors due to axial vibration given by the following Alternatively the factors may be calculated by dynamic analysis. $\alpha = 2.2$ $\beta = 1.5$	
2 For pulling type propellers the ice interaction loads on propeller hubs are to be considered in addition to **-1** above.	(Newly added)	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr>
<td>

Table 4.4.11-1 Value of T

<table>
<tr><td>Propeller type</td><td>T</td></tr>
<tr><td>Controllable pitch propellers (open)</td><td>$1.25T_n$</td></tr>
<tr><td>Controllable pitch propellers (ducted)</td><td>$1.1T_n$</td></tr>
<tr><td>Fixed pitch propellers driven by turbine or electric motor</td><td>T_n</td></tr>
<tr><td>Fixed pitch propellers driven by diesel engine (open)</td><td>$0.85T_n$</td></tr>
<tr><td>Fixed pitch propellers driven by diesel engine (ducted)</td><td>$0.75T_n$</td></tr>
</table>

Notes:
T_n: Nominal propeller thrust (kN) at maximum continuous revolutions in free running open water conditions

</td>
<td>

Table 4.2.8-1 Value of T_n

<table>
<tr><td>Propeller type</td><td>T_n</td></tr>
<tr><td>Controllable pitch propellers (open)</td><td>$1.25T$</td></tr>
<tr><td>Controllable pitch propellers (ducted)</td><td>$1.1T$</td></tr>
<tr><td>Fixed pitch propellers driven by turbine or electric motor</td><td>T</td></tr>
<tr><td>Fixed pitch propellers driven by diesel engine (open)</td><td>$0.85T$</td></tr>
<tr><td>Fixed pitch propellers driven by diesel engine (ducted)</td><td>$0.75T$</td></tr>
</table>

T: Nominal propeller thrust (kN) at maximum continuous revolutions in free running open water conditions

</td>
<td>Table 7</td>
</tr>
<tr>
<td>

4.4.12 Maximum Propeller Ice Torque

Maximum propeller ice torque applied to the propeller is to be given by the following formulae:

(1) For open propellers:

when $D < D_{\mathrm{limit}}$

$Q_{\max} = k_{open}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}\left(\frac{n}{60}D\right)^{0.17}D^3$ (kNm)

when $D \geq D_{\mathrm{limit}}$

$Q_{\max} = 1.9k_{open}\left(1-\frac{d}{D}\right)(H_{ice})^{1.1}\left(\frac{P_{0.7}}{D}\right)^{0.16}\left(\frac{n}{60}D\right)^{0.17}D^{1.9}$ (kNm)

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

k_{open}: Factor for open propeller of each polar class is given below.

$PC1$ to $PC5$ $k_{open} = 14.7$
$PC6$ to $PC7$ $k_{open} = 10.9$

(2) For ducted propellers:

when $D < D_{\mathrm{limit}}$

</td>
<td>

4.2.5 Maximum Propeller Ice Torque

Maximum propeller ice torque applied to the propeller is to be given by the following formulae:

(1) For open propellers:

when $D < D_{\mathrm{limit}}$

$Q_{\max} = 105S_{qice}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}\left(\frac{t_{0.7}}{D}\right)^{0.6}\left(\frac{n}{60}D\right)^{0.17}D^3$ (kNm)

when $D \geq D_{\mathrm{limit}}$

$Q_{\max} = 202S_{qice}(H_{ice})^{1.1}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}\left(\frac{t_{0.7}}{D}\right)^{0.6}\left(\frac{n}{60}D\right)^{0.17}D^{1.9}$ (kNm)

where $D_{limit} = 1.81H_{ice}$ (m)
(Newly added)

(2) For ducted propellers:

when $D \leq D_{\mathrm{limit}}$

</td>
<td>Para. 5.6.1
Para. 5.6.2</td>
</tr>
</table>

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
$Q_{\max} = k_{ducted}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}\left(\frac{n}{60}D\right)^{0.17} D^3$ (kNm) when $D \geq D_{limit}$ $Q_{\max} = 1.9k_{ducted}\left(1-\frac{d}{D}\right)(H_{ice})^{1.1}\left(\frac{P_{0.7}}{D}\right)^{0.16}\left(\frac{n}{60}D\right)^{0.17} D^{1.9}$ (kNm) where: $D_{limit} = 1.8H_{ice}$ (m) k_{ducted}: Factor for open propeller of each polar class is given below. PC1 to PC5 $k_{ducted} = 10.4$ PC6 to PC7 $k_{ducted} = 7.7$ (Deleted) (Deleted) $P_{0.7}$: Propeller pitch (m) at 0.7R For controllable pitch propellers, $P_{0.7}$ is to correspond to maximum continuous revolutions 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 maximum continuous revolutions in free running open water condition. (Deleted) n : Rotational propeller speed (rpm) at bollard condition If not known, n is to be taken as specified in **Table 4.4.12-1**.	$Q_{\max} = 74S_{qice}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}\left(\frac{t_{0.7}}{D}\right)^{0.6}\left(\frac{n}{60}D\right)^{0.17} D^3$ (kNm) when $D > D_{limit}$ $Q_{\max} = 141S_{qice}(H_{ice})^{1.1}\left(1-\frac{d}{D}\right)\left(\frac{P_{0.7}}{D}\right)^{0.16}\left(\frac{t_{0.7}}{D}\right)^{0.6}\left(\frac{n}{60}D\right)^{0.17} D^{1.9}$ (kNm) where: $D_{limit} = 1.8H_{ice}$ (m) (Newly added) H_{ice}, D and d : As specified in **4.2.2-1** and **4.2.3-1** S_{qice} : Ice strength index for blade ice torque specified in **Table 4.2.5-1** $P_{0.7}$: Propeller pitch (m) at 0.7R For controllable pitch propellers, $P_{0.7}$ is to correspond to maximum continuous revolutions 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 maximum continuous revolutions in free running condition. $t_{0.7}$: Maximum thickness (mm) at 0.7R n : Rotational propeller speed (rpm) at bollard condition If not known, n is to be taken as specified in **Table 4.2.5-2**.	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr>
<td>
(Deleted)

Table 4.4.12-1 Rotational Propeller Speed
<table>
<tr><td>Propeller type</td><td>n</td></tr>
<tr><td>Controllable pitch propellers</td><td>n_n</td></tr>
<tr><td>Fixed pitch propellers driven by turbine or electric motor</td><td>n_n</td></tr>
<tr><td>Fixed pitch propellers driven by diesel engine</td><td>$0.85n_n$</td></tr>
</table>
Notes:
n_n : Nominal rotational speed (rpm) at maximum continuous revolutions in free running condition
</td>
<td>
Table 4.2.5-1 Value of S_{qice}
<table>
<tr><td>Polar class</td><td>S_{qice}</td></tr>
<tr><td>$PC1$</td><td>1.15</td></tr>
<tr><td>$PC2$</td><td>1.15</td></tr>
<tr><td>$PC3$</td><td>1.15</td></tr>
<tr><td>$PC4$</td><td>1.15</td></tr>
<tr><td>$PC5$</td><td>1.15</td></tr>
<tr><td>$PC6$</td><td>1</td></tr>
<tr><td>$PC7$</td><td>1</td></tr>
</table>

Table 4.2.5-2 Rotational Propeller Speed
<table>
<tr><td>Propeller type</td><td>n</td></tr>
<tr><td>Controllable pitch propellers</td><td>n_n</td></tr>
<tr><td>Fixed pitch propellers driven by turbine or electric motor</td><td>n_n</td></tr>
<tr><td>Fixed pitch propellers driven by diesel engine</td><td>$0.85n_n$</td></tr>
</table>
n_n : Nominal rotational speed (rpm) at maximum continuous revolutions in free running condition
</td>
<td></td>
</tr>
</table>

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
4.4.13 Ice Torque Excitations **1** General (1) The given excitations are used to estimate the maximum torque likely to be experienced once during the service life of the ship. The load cases in this paragraph are intended to reflect the operational loads on the propulsion system when the propeller interacts with ice and the corresponding reaction of the complete system. The ice impact and system response cause loads in the individual shaft line components. The ice torque Q_{max} may be taken as a constant value in the complete speed range. When considerations at specific shaft speeds are performed a relevant Q_{max} may be calculated using the relevant speed. (2) Diesel engine plants without an elastic coupling are to be calculated at the least favourable phase angle for ice versus engine excitation, when calculated in time domain. In addition, engine firing pulses are to be included in the calculations and their standard steady state harmonics can be used. The phase angles between ice and gas force excitation do not need to be considered in frequency domain analyses, and misfiring does not need to be considered at all. (3) If there is a blade order resonance just above *MCR* speed, calculations are to cover the rotational speeds up to 105 % of *MCR* speed. (4) See **4.4.15**.	**(Newly added)** (Newly added)	Para. 5.6.3
2 Time domain calculation (1) The propeller ice excitation torque for shaft line transient dynamic analysis in the time domain is defined as a sequence of blade impacts which are of half sine shape and	(Newly added)	Para. 5.6.3.1

Amended	Original	Remarks

occur at the blade. The torque due to a single blade ice impact as a function of the propeller rotation angle is then given by the following formulae:

(a) when $0 \leq \varphi - 360x \leq \alpha_i$ (*deg*)

$$Q(\varphi) = C_q Q_{max} \sin\left(\varphi(180/\alpha_i)\right)$$

(b) when $\alpha_i \leq \varphi - 360x \leq 360$ (*deg*)

$$Q(\varphi) = 0$$

where

φ: Rotation angle from when the first impact occurs

X: Integer revolutions from the time of first impact

C_q: As specified in **Table 4.4.13-1**

a_i: Duration of propeller blade/ice interaction expressed in rotation angle as specified in **Table 4.4.13-1**

Table 4.4.13-1 Values of C_q and a_i

Torque excitation	Propeller-ice interaction	C_q	a_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 360/(2・Z) *deg*)	0.5	45	45	36	30
Case 4	Single ice block	0.5	45	45	36	30

(2) Total ice torque is obtained by summing the torque of single blades, while taking account of the phase shift 360 *deg* /*Z*.

(3) At the beginnings and ends of milling sequences (within the calculated duration), linear ramp functions are to be used to

Original: (Newly added)

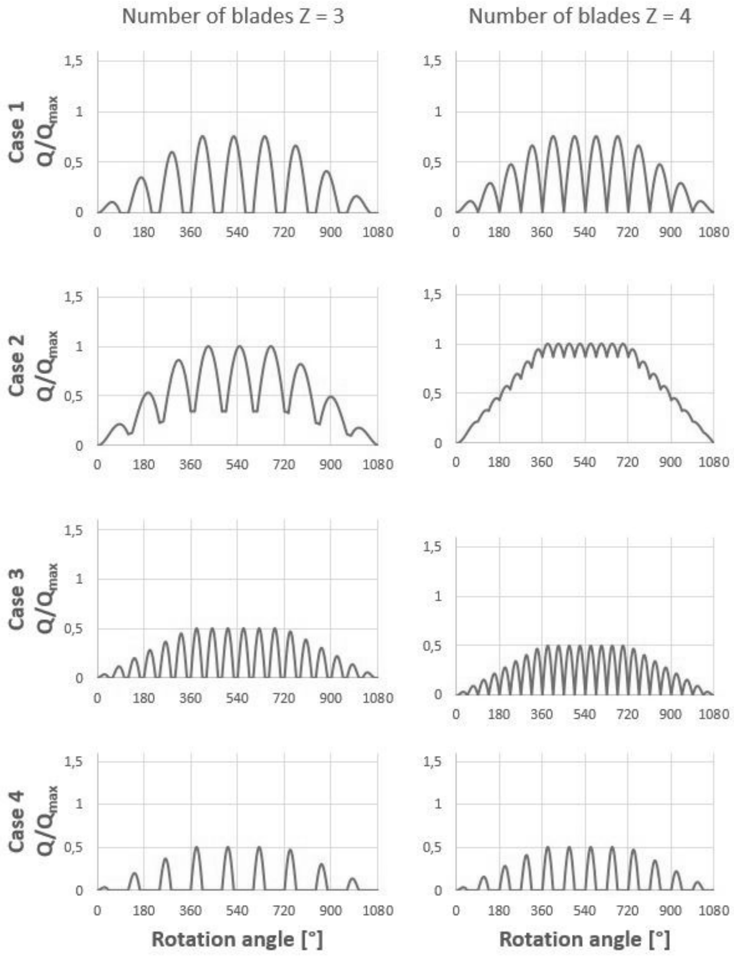
Remarks:

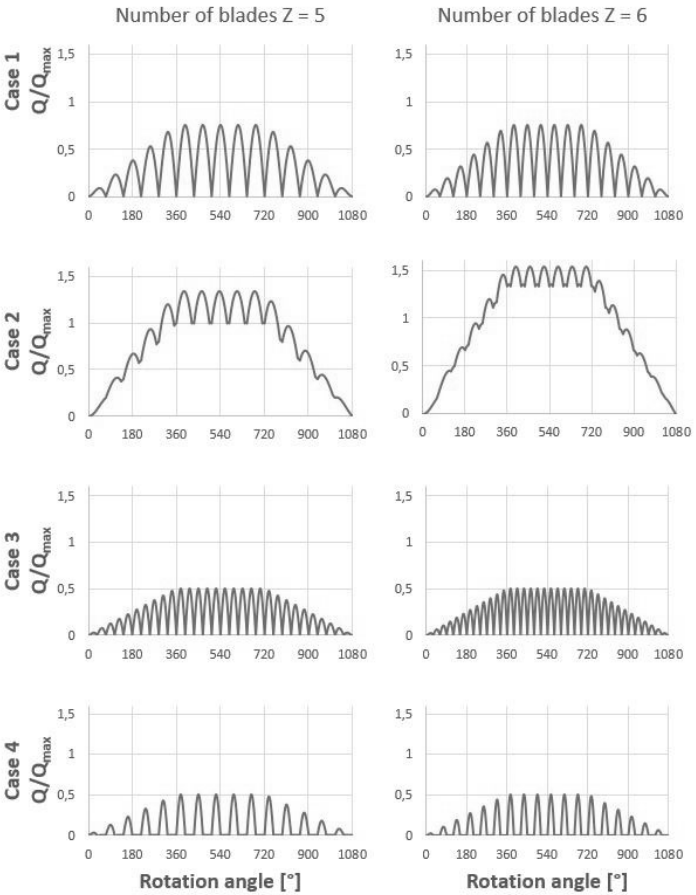
Table 9

Para. 5.6.3.1

Amended	Original	Remarks
increase C_q to its maximum value within one propeller revolution and vice versa to decrease it to zero. (4) The number of propeller revolutions and the number of impacts during milling sequences are to be obtained by the following formulae. (a) Number of propeller revolutions: $N_Q = 2H_{ice}$ (b) The number of impacts: ZN_Q where Z: Number of propeller blades Examples of all excitation cases for different numbers of blades are showing in **Fig. 4.4.13-1** and **Fig. 4.4.13-2**. (5) Dynamic simulation is to be performed for all excitation cases starting at MCR nominal, MCR bollard condition and just above all resonance speeds (1st engine and 1st blade harmonic), so that resonant vibration responses can be obtained. For fixed pitch propeller plants, such dynamic simulation is to also cover the bollard pull condition with a corresponding speed assuming maximum possible output of the engine. (6) If a speed drop occurs down to stand still of the main engine, it indicates that the engine may not be sufficiently powered for the intended service task. For consideration of loads, the maximum occurring torque during the speed drop process is to be applied. In such cases, excitation is to follow shaft speed.		

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
Fig. 4.4.13-1 Example of the Excitation Torque due to Torsional Load for Different Blade Numbers(Z=3 and Z=4) in Polar Class *PC*7 (H_{ice} = 1.5)  	(Newly added)	Appendix

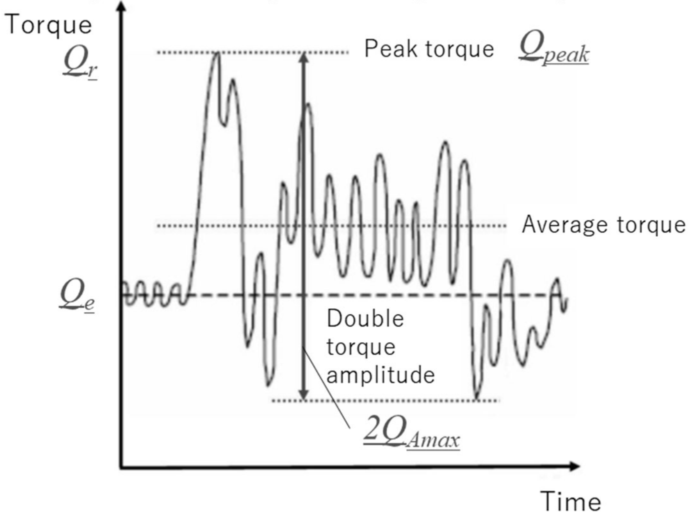
Amended	Original	Remarks
Fig. 4.4.13-2 Example of the Excitation Torque due to Torsional Load for Different Blade Numbers(*Z*=5 and *Z*=6) in Polar Class *PC*7 (H_{ice} = 1.5)  	(Newly added)	Appendix

Amended	Original	Remarks
3 Frequency domain excitations (1) For frequency domain calculations, the torque excitation is to be given by following formula. The excitation has been derived so that the time domain half sine impact sequences have been assumed to be continuous and the Fourier series components for blade order and twice the blade order components have been derived. The frequency domain analysis is generally considered as conservative compared to the time domain simulation provided there is a first blade order resonance in the considered speed range: $Q_F(\varphi) = Q_{max}(C_{q0} + C_{q1}\sin(ZE_0\varphi + \alpha_1) + C_{q2}\sin(2ZE_0\varphi + \alpha_2))$ (kNm) where C_{q0}: Mean torque parameter, as specified in **Table 4.4.13-2** C_{q1}: First blade order excitation parameter, as specified in **Table 4.4.13-2** C_{q2}: Second blade order excitation parameter, as specified in **Table 4.4.13-2** α_1, α_2 : Phase angles of the excitation component, as specified in **Table 4.4.13-2** φ : Angle of rotation E_0: Number of ice blocks in contact, as specified in **Table 4.4.13-2** Z : Number of propeller blades	(Newly added)	Para. 5.6.3.2

Amended	Original	Remarks
Table 4.4.13-2 Values of C_{q0}, C_{q1}, α_1, C_{q2}, α_2, and E_0	(Newly added)	Table 10

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

Amended	Original	Remarks
(2) Torsional vibration responses are to be calculated for all excitation cases.		Para. 5.6.3.2
(3) The results of the relevant excitation cases at the most critical rotational speeds are to be used in the following. The highest response torque (between the various lumped masses in the system) is in the following referred to as peak torque Q_{peak}. The highest torque amplitude during a sequence of impacts is to be determined as half of the range from max to min torque and is referred to as Q_{Amax}. An illustration of Q_{Amax} is given in **Fig. 4.4.13-3**. The highest torque amplitude is given by following formula:		

Amended	Original	Remarks
$$Q_{Amax} = \left(\frac{max(Q_r(time))-min(Q_r(time))}{2}\right) \ (kNm)$$ Fig.4.4.13-3 Interpretation of Q_{Amax} in Torque Curve  	(Newly added)	Figure 4
4.4.14 Design Torque on Propulsion Shafting System **1** If there is not a predominant 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. All the torques and the inertia moments are to be reduced to the rotation speed of the component being examined: Directly coupled two stroke diesel engines without flexible coupling	**4.2.7 Design Torque on Propulsion Shafting System** (Newly added)	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td>

$$Q_{peak} = Q_{emax} + Q_{vib} + Q_{max}\frac{I}{I_t}\ (kNm)$$

and other plants

$$Q_{peak} = Q_{emax} + Q_{max}\frac{\mathrm{I}}{I_t}\ (kNm)$$

where
Q_{peak}: Maximum response torque (kNm)
Q_{emax}: Maximum engine torque (kNm)
If the maximum torque, Q_{emax}, is not known, it is to be taken as specified in **Table 4.4.14-1**
Q_{vib}: Vibratory torque at considered component, taken from frequency domain open water torque vibration calculation (TVC)
I_e: Equivalent mass moment of inertia of all parts on the engine side of the component under consideration (kgm^2)
I_t: Equivalent mass moment of inertia of the entire propulsion system (kgm^2)

Table 4.4.14-1 Maximum Engine Torque Q_{emax}

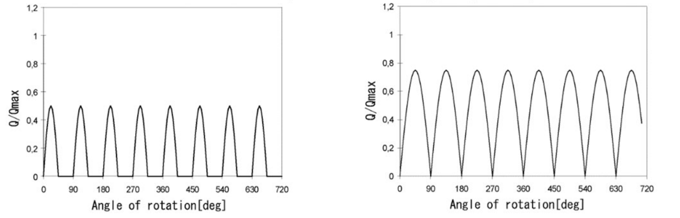
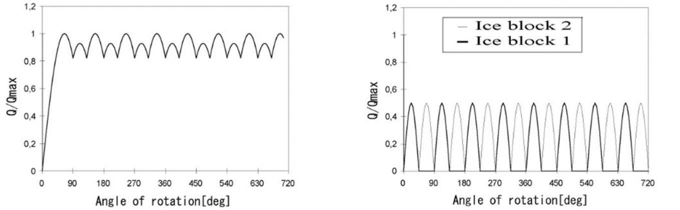
<table>
<tr><th>Propeller type</th><th>Q_{emax}</th></tr>
<tr><td>Propellers driven by electric motor</td><td>Q_{motor}</td></tr>
<tr><td>CP propellers not driven by electric motor</td><td>Q_n</td></tr>
<tr><td>FP propellers driven by turbine</td><td>Q_n</td></tr>
<tr><td>FP propellers driven by diesel engine</td><td>$0.75\ Q_n$</td></tr>
</table>

Notes:
Q_{motor}: Electric motor peak torque (kNm)
Q_n: Nominal torque at maximum continuous revolutions in free running condition (kNm)

</td><td>(Newly added)</td><td>Table 11</td></tr>
</table>

Amended	Original	Remarks
2 If there is a first blade order torsional resonance in the range 20 % above and 20 % below the maximum operating speed (bollard condition), the design torque of the shaft component is to be determined by means of a dynamic torsional vibration analysis of the entire propulsion line in the time domain or alternatively in the frequency domain. It is then assumed that the plant is sufficiently designed to avoid harmful operation in barred speed range.	(Newly added)	
(Deleted)	**1** The propeller ice excitation torque for shaft line dynamic analysis is to comply with the following requirements. (1) The excitation torque is to be described by a sequence of blade impacts which are of half sine shape and occur at the blade. The total ice torque is to be obtained by summing the torques of single ice blade ice impacts taking into account the phase shift. Single ice blade impact is to be given by the following formula. (See **Fig. 4.2.7-1**) (a) when $\varphi = 0\ to\ \alpha_i$ (*deg*) $Q(\varphi) = C_q Q_{\max} \sin(\varphi(180/\alpha_i))$ (b) when $\varphi = \alpha_i\ to\ 360$ (*deg*) $Q(\varphi) = 0$ where $Q_{\max}$: As specified in **4.2.5** C_q and α_i : As specified in **Table 4.2.7-1** (2) The number of propeller revolutions and the number of impacts during the milling sequence are to be given by the following formulae. For bow propellers, the number of propeller revolutions and the number of impacts during the milling sequence are subject to special consideration. (a) The number of propeller revolutions: $N_Q = 2H_{ice}$	

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td></td><td>(b) The number of impacts:
ZN_Q
Where
H_{ice} :As specified in Table 4.2.2-1
Z : Number of propeller blades</td><td></td></tr>
<tr><td>(Deleted)</td><td>2 The response torque at any shaft component is to be analyzed considering excitation torque at the propeller specified in -1, actual engine torque and mass elastic system.</td><td></td></tr>
<tr><td>(Deleted)</td><td>3 The design torque of the shaft component is to be determined by means of torsional vibration analysis of the propulsion line. Calculation is to be carried out for all excitation cases specified in Table 4.2.7-1 and the response is to be applied on top of the mean hydrodynamic torque in bollard condition at the considered propeller rotational speed.</td><td></td></tr>
<tr><td>(Deleted)</td><td>Table 4.2.7-1 Values of C_q and α_i
<table>
<tr><th>Torque excitation</th><th>Propeller-ice interaction</th><th>C_q</th><th>α_i</th></tr>
<tr><td>Case 1</td><td>Single ice block</td><td>0.5</td><td>45</td></tr>
<tr><td>Case 2</td><td>Single ice block</td><td>0.75</td><td>90</td></tr>
<tr><td>Case 3</td><td>Single ice block</td><td>1.0</td><td>135</td></tr>
<tr><td>Case 4</td><td>Two ice blocks with 45 degree phase in rotation angle</td><td>0.5</td><td>45</td></tr>
</table>
</td><td></td></tr>
</table>

Amended	Original	Remarks
(Deleted)	Fig.4.2.7-1 Example of the Shape of the Propeller Ice Torque Excitation (four bladed propeller)  (a) Case 1 Single blade impact (αi = 45[deg]) (b) Case 2 Single blade impact (αi = 90[deg])  (c) Case 3 Single blade impact (αi = 135[deg]) (d) Case 4 Double blade impact (αi = 45[deg])	
4.4.15 Torsional Vibration Calculations **1** The aim of Torsional vibration calculations are used to estimate the torsional loads for individual shaft line components over the life time of the ship in order to determine scantlings for safe operation. The model can be taken from the normal lumped mass elastic torsional vibration model (frequency domain) including the damping. Standard harmonics may be used to consider the gas forces. The engine torque speed curve of the actual plant is to be applied. **2** For time domain analysis, the model is to include the ice excitation at the propeller, the mean torques provided by the prime	**(Newly added)** (Newly added) (Newly added)	Para. 5.7

Amended	Original	Remarks
mover and the hydrodynamic mean torque produced by the propeller as well as any other relevant excitations. The calculations are to cover the variation of phase between the ice excitation and prime mover excitation. This is extremely relevant for propulsion lines with direct driven combustion engines. **3** For frequency domain calculations the load is to be estimated as a Fourier component analysis of the continuous sequence of half sine load peaks. The first and second order blade components is to be used for excitation, and calculations are to cover the entire relevant shaft speed range. The analysis of the responses at the relevant torsional vibration resonances may be performed for open water (without ice excitation) and ice excitation separately. The resulting maximum torque can be obtained for directly coupled plants by the following superposition: $Q_{peak} = Q_{emax} + Q_{opw} + Q_{ice} \frac{I}{I_t}$ (*kNm*) where Q_{emax}: Maximum engine torque at considered rotational speed (*kNm*) Q_{opw}: Maximum open water response of engine excitation at considered shaft speed and determined by frequency domain analysis (*kNm*) Q_{ice}: Calculated torque using frequency domain analysis for the relevant shaft speeds, ice excitation case 1 to case 4, resulting in the maximum response torque due to ice excitation (*kNm*)	(Newly added)	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
4.5 Design	**4.3 Design of Propulsion Shafting System**	Para .6
4.5.1 Design Principles	**(Newly added)**	Para. 6.1
1 Propulsion lines are to be designed according to the pyramid strength principle. This means that the loss of a propeller blade is not to cause any significant damage to other propeller shaft line components.	(Newly added)	
2 Propulsion line components are to withstand maximum and fatigue operational loads with the relevant safety margin. The loads do not need to be considered for shaft alignment or other calculations of normal operational conditions such the torsional vibration of shafting specified in **Chapter 8, Part D of the Rules**.	(Newly added)	
4.5.2 Fatigue Design in General	**(Newly added)**	Para. 6.2
1 Design loads are to be based on ice excitation and where necessary (shafting) dynamic analysis, and described as a sequence of blade impacts (**4.4.13-2**). Shaft response torque is to be determined according to **4.4.14**.	(Newly added)	
2 Propulsion line components are to be designed so as to prevent accumulated fatigue failure when considering the relevant loads using the linear elastic Miner's rule defined as follows: $D=\frac{n_1}{N_1}+\frac{n_2}{N_2}+\cdots+\frac{n_k}{N_k}\leq 1$ or $D=\sum_{j=1}^{j=k}\frac{n_j}{N_j}\leq 1$ where k: Number of stress level $N_{1..k}$: Number of load cycles to failure of the individual stress level class $n_{1..k}$: Accumulated number of load cycles of the case under	(Newly added)	

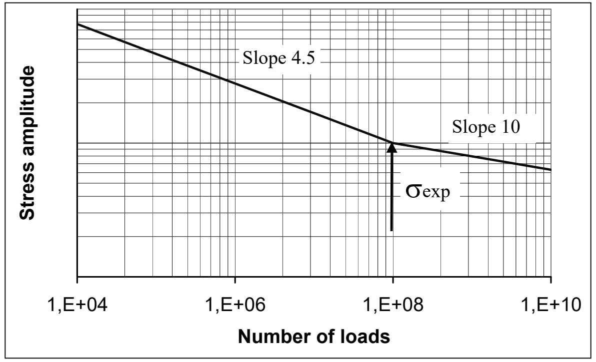
Amended	Original	Remarks
consideration, per class D: Sum of damage using Miner's rules		
3 Stress distributions are to be divided into frequency load spectrums having a minimum ten stress blocks (every 10 % of the load) because calculations having five stress blocks have been found to be too conservative. In addition, maximum allowable loads are limited by σ_{ref2} for propeller blades and yield strength for all other components, and load distributions (spectrums) are to be in accordance with the Weibull distribution.	(Newly added)	
4.5.3 Propeller Blades	**(Newly added)**	Para. 6.3
1 Calculations of blade stresses due to static loads are as follows: (1) Propeller blade stresses are to be calculated for the design loads given in **4.4.3** to **4.4.8** using Finite Element Analysis. (2) In the case of a relative radius $r/R < 0.5$, the blade stresses for all propellers at their root areas may be obtained by the following formula: $\sigma_{st} = C_1 \frac{M_{BL}}{100ct^2}$ (MPa) where C_1: $\frac{\text{stress obtained with FEM analysis result}}{\text{stress obtained with beam equation}}$ If the actual value is not available, C_1 is to be taken as 1.6. M_{BL}: Blade bending moment (kNm), but to be as follows in the case of a relative radius $r/R < 0.5$: $\mathrm{M_{BL}} = (0.75 - \mathrm{r/R})\mathrm{R}F$ where F: Force F_b or F_f, whichever has the greater absolute value.	(Newly added)	Para. 6.3.1
2 The calculated blade stress σ_{st} specified in **-1** above is to	(Newly added)	Para. 6.3.2

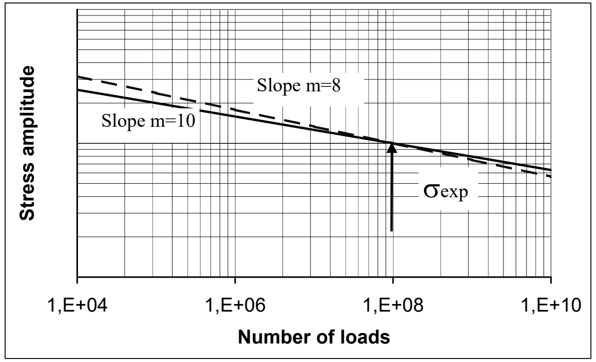
Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
comply with the following: $\frac{\sigma_{ref2}}{\sigma_{st}} \geq 1.3$ where σ_{st}: Maximum stress resulting from F_b or F_f(MPa). If Finite Element Analysis is used in estimating the stresses, von Mises stresses are to be used. σ_{ref2}: Reference strength (MPa), whichever is less, as obtained by the following formulae: $\sigma_{ref2} = 0.7\sigma_u$, or $\sigma_{ref2} = 0.6\sigma_{0.2} + 0.4\sigma_u$ **3** Fatigue design of propeller blades (1) General (a) For materials with two-slope S-N curves (See **Fig. 4.5.3-1**), the fatigue calculations specified in this section are not required if the following criterion is fulfilled. $\sigma_{\exp} \geq B_1 \sigma_{ref2}{}^{B_2} \log(N_{ice})^{B_3}$ where σ_{exp}: Mean fatigue strength of the blade material at 10^8 cycles to failure in seawater (MPa), as given in **Table 4.5.3-4** B_1, B_2 and B_3: Coefficients, as given in **Table 4.5.3-1**	(Newly added)	Para. 6.3.3 Para. 6.3.3.1

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td>
Table 4.5.3-1 The Coefficients B_1, B_2 and B_3
<table>
<tr><th>Coefficients</th><th>Open propeller</th><th>Ducted propeller</th></tr>
<tr><td>B_1</td><td>0.00328</td><td>0.00223</td></tr>
<tr><td>B_2</td><td>1.0076</td><td>1.0071</td></tr>
<tr><td>B_3</td><td>2.101</td><td>2.471</td></tr>
</table>
</td><td>(Newly added)</td><td>Table 12</td></tr>
<tr><td>
(b) Where the criterion in **(a)** above is not fulfilled the fatigue requirements defined below apply:
i) The fatigue design of a propeller blade is based on the estimated load distribution for the service life of the ship and the S-N curve for the blade material. An equivalent stress σ_{fat} that produces the same fatigue damage as the expected load distribution is to be calculated according to Miner’s rule. An equivalent stress that produces the same fatigue damage as the expected load distribution is to be calculated and the acceptability criterion for fatigue is to be fulfilled as given in this paragraph. The equivalent stress is normalised for 10^8 cycles.
ii) The blade stresses at various selected load levels for fatigue analysis are to be taken proportional to the stresses calculated for maximum loads given in **4.4.3** to **4.4.8**. The peak principal stresses σ_f and σ_b are determined from F_f and F_b using Finite Element analysis. The peak stress range $\Delta\sigma_{max}$ and the maximum stress amplitude σ_{Amax} are respectively determined on the basis of load cases 1 and 3, and cases 2 and 4.
$\Delta\sigma_{\max} = 2\sigma_{Amax} = \left|(\sigma_{ice})_{fmax}\right| +$
</td><td></td><td>Para. 6.3.3.1</td></tr>
</table>

Amended	Original	Remarks
$\lvert(\sigma_{ice})_{bmax}\rvert$ iii) The load spectrum for backward loads is normally expected to have a lower number of cycles than the load spectrum for forward loads. Since taking this into account in a fatigue analysis introduces complications that are not justified considering all uncertainties involved, two types of S-N curves are to be used for calculations of equivalent stress. 1) Two-slope S-N curve (slopes 4.5 and 10) (See **Fig. 4.5.3-1**) 2) One-slope S-N curve (the slope can be chosen) (See **Fig. 4.5.3-2**) iv) S-N curve type is to be selected to correspond with the material properties of the blade. If the S-N curve is unknown, a two-slope S-N curve is to be used. Fig.4.5.3-1 Two-slope S-N Curve  	(Newly added)	Figure 5

Amended	Original	Remarks
Fig.4.5.3-2 Constant-slope S-N Curve  	(Newly added)	Figure 6
(2) Equivalent fatigue stress (a) A more general method of determining the equivalent fatigue stress of propeller blades is described in **4.5.5**, where the principal stresses are considered in accordance with **4.4.3** to **4.4.8** using the Miner's rule. For the total number of load blocks nbl > 100, both methods deliver the same result, therefore, are regarded as equivalent. (b) The equivalent fatigue stress for 10^8 cycles which produces the same fatigue damage as the load distribution is to be obtained as follows: $\sigma_{fat} = \rho(\sigma_{ice})_{max}$ where $(\sigma_{ice})_{max} = 0.5\left((\sigma_{ice})_{fmax} - (\sigma_{ice})_{bmax}\right)$ $(\sigma_{ice})_{max}$: The mean value of the principal stress amplitudes resulting from design forward and backward blade forces at the location being studied.$(\sigma_{ice})_{fmax}$: The principal stress resulting he from forward load $(\sigma_{ice})_{bmax}$: The principal stress resulting from		Para. 6.3.3.2

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr>
<td>backward load
(c) In the calculation of $(\sigma_{ice})_{max}$, case 1 and case 3, or case 2 and case 4 are to be considered as pairs for $(\sigma_{ice})_{fmax}$ and $(\sigma_{ice})_{bmax}$ calculations. Case 5 is excluded from the fatigue analysis.
(d) The calculation of the parameter ρ for a two-slope S-N curve
i) The range of the number of load cycles N_{ice} is to be given as follows. In such cases the error of the method in **ii)** to determine the parameter ρ is sufficiently small.
$5 \times 10^6 \leq N_{ice} \leq 10^8$
ii) Parameter ρ relates the maximum ice load to the distribution of ice loads in accordance with the following regression formulae:
$\rho = C_1(\sigma_{ice})_{max}{}^{C2}\sigma_{fl}{}^{C3}\log(N_{ice})^{C4}$
where
σ_{fl}: Characteristic fatigue strength for blade material for 10^8 load cycles(MPa) (See **4.5.3-3(3)**)
C_1, C_2, C_3 and C_4: Coefficients, as given in **Table 4.5.3-2**

Table 4.5.3-2 The Coefficients C_1, C_2, C_3 and C_4
<table>
<tr><th>Coefficients</th><th>Open propeller</th><th>Ducted propeller</th></tr>
<tr><td>C_1</td><td>0.000747</td><td>0.000534</td></tr>
<tr><td>C_2</td><td>0.0645</td><td>0.0533</td></tr>
<tr><td>C_3</td><td>-0.0565</td><td>-0.0459</td></tr>
<tr><td>C_4</td><td>2.220</td><td>2.584</td></tr>
</table>
</td>
<td>(Newly added)</td>
<td>Table 13</td>
</tr>
</table>

Amended	Original	Remarks
(e) Calculations of the parameter ρ for constant-slope S-N curves i) In the case of materials with constant-slope S-N curves (See **Fig. 4.5.3-2**), the ρ parameter is to be obtained by the following formula: $\rho = \left(G\frac{N_{ice}}{N_R}\right)^{1/m}\left(\ln(N_{ice})\right)^{-1/k}$ where k: Shape parameter of the Weibull distribution to be taken as follows: Ducted propellers: 1.0 Open propellers: 0.75 N_R: The reference number of load cycles (= 10^8) m: slope for S-N curve in log/log scale G: Values corresponding to m/k given in **Table 4.5.3-3**. Linear interpolation may be used to calculate the G value of m/k ratios other than those given in **Table 4.5.3-3**.		

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td>

Table 4.5.3-3 Value for the *G* Parameter for Different *m/k* Ratios

<table>
<tr><th><i>m/k</i></th><th><i>G</i></th></tr>
<tr><td>3</td><td>6</td></tr>
<tr><td>3.5</td><td>11.6</td></tr>
<tr><td>4</td><td>24</td></tr>
<tr><td>4.5</td><td>52.3</td></tr>
<tr><td>5</td><td>120</td></tr>
<tr><td>5.5</td><td>287.9</td></tr>
<tr><td>6</td><td>720</td></tr>
<tr><td>6.5</td><td>1871</td></tr>
<tr><td>7</td><td>5040</td></tr>
<tr><td>7.5</td><td>14034</td></tr>
<tr><td>8</td><td>40320</td></tr>
<tr><td>8.5</td><td>119292</td></tr>
<tr><td>9</td><td>362880</td></tr>
<tr><td>9.5</td><td>1.133E6</td></tr>
<tr><td>10</td><td>3.629E6</td></tr>
<tr><td>10.5</td><td>11.899E6</td></tr>
<tr><td>11</td><td>39.917E6</td></tr>
<tr><td>11.5</td><td>136.843E6</td></tr>
<tr><td>12</td><td>479.002E6</td></tr>
</table>

(3) For the acceptance criterion for fatigue, the equivalent fatigue stresses at locations on blades area to satisfy the following acceptance criterion:

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

where

σ_{fat}: Equivalent fatigue ice load stress amplitude for 10^8 stress cycles

σ_{fl}: Characteristic given by following formula:

$$\sigma_{fl} = \gamma_{\varepsilon 1}\gamma_{\varepsilon 2}\gamma_{v}\gamma_{m}\sigma_{\exp}$$

</td><td>(Newly added)</td><td>Table 14

Para. 6.3.3.3</td></tr>
</table>

Amended	Original	Remarks
$\gamma_{\varepsilon 1}$: The reduction factor due to scatter (equal to one standard deviation) $\gamma_{\varepsilon 2}$: The reduction factor for test specimen size effect obtained by the following formula: $\gamma_{\varepsilon 2} = 1 - a \cdot \ln\left(\frac{t}{0.025}\right)$ where a: The values given in **Table 4.5.3-4** t: Maximum blade section thickness (m) γ_{v}: The reduction factor for variable amplitude loading γ_{m}: The reduction factor for mean stress obtained by the following formula: $\gamma_{m} = 1 - \left(\frac{1.4\sigma_{mean}}{\sigma_{u}}\right)^{0.75}$ σ_{exp}: The mean fatigue strength of the blade material at 10^{8} cycles to failure in seawater (MPa). The values in **Table 4.5.3-4** are to be used. The following values are to be used as reduction factors if actual values are unavailable: $\gamma_{\varepsilon 1} = 0.67$ $\gamma_{v} = 0.75$ $\gamma_{m} = 0.75$		

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td>Table 4.5.3-4 Mean Fatigue Strength σ_{exp} for Different Material Types
<table>
<tr><th colspan="2">Bronze and brass ($a=0.01$)</th><th colspan="2">Stainless steel ($a=0.05$)</th></tr>
<tr><td><i>Mn</i> Bronze (<i>KHBsC</i>1)</td><td>84 <i>MPa</i></td><td>Ferritic (12<i>Cr</i>-1<i>Ni</i>) (<i>KSCSP</i>1)</td><td>144 <i>MPa</i> (2)</td></tr>
<tr><td><i>Mn-Ni</i> Bronze (<i>KHBsC</i>2)</td><td>84 <i>MPa</i></td><td>Martensitic (13<i>Cr</i>-4<i>Ni</i>) (<i>KSCSP</i>2)</td><td>156 <i>MPa</i></td></tr>
<tr><td><i>Ni-Al</i> Bronze (<i>KAlBC</i>3)</td><td>120 <i>MPa</i></td><td>Martensitic(16<i>Cr</i>-5<i>Ni</i>) (<i>KSCSP</i>3)</td><td>168 <i>MPa</i></td></tr>
<tr><td><i>Mn-Al</i> Bronze (<i>KAlBC</i>4)</td><td>113 <i>MPa</i></td><td>Austenitic(19<i>Cr</i>-11<i>Ni</i>) (<i>KSCSP</i>3)</td><td>132 <i>MPa</i></td></tr>
</table></td><td>(Newly added)</td><td>Table 15</td></tr>
<tr><td>Notes:
(1) Values are defined from the results of constant amplitude loading fatigue tests at 10^7 load cycles and 50% survival probability and have been extended to 10^8 load cycles. Values other than those given in this table may be used, provided they are deemed appropriate by the Society. S-N curve characteristics are based on two slopes, the first slope 4.5 is from 1000 to 10^8 load cycles and the second slope 10 is above 10^8 load cycles. The maximum allowable stress for one or a low number of cycles is limited to the range in 4.5.3-2, fatigue strength σ_{fat} is the fatigue limit at 100 million load cycles.
(2) This value may be used, provided perfect galvanic protection is active, otherwise a reduction of about 30 MPa is to be applied.</td><td></td><td>Para. 6.3.3.3</td></tr>
<tr><td>4.5.4 Blade Bolts, Propeller Hubs and CP Mechanisms</td><td>(Newly added)</td><td>Para. 6.4</td></tr>
<tr><td>1 General</td><td>(Newly added)</td><td>Para. 6.4.1</td></tr>
<tr><td>(1) The blade bolts, CP mechanisms, propeller bosses, and fitting of propellers to propeller shafts are to be designed to withstand maximum static and fatigue design loads (as applicable) defined in 4.4.3 to 4.4.8 and 4.5.3, and safety factors are to be greater than follows unless otherwise stated.</td><td></td><td></td></tr>
</table>

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

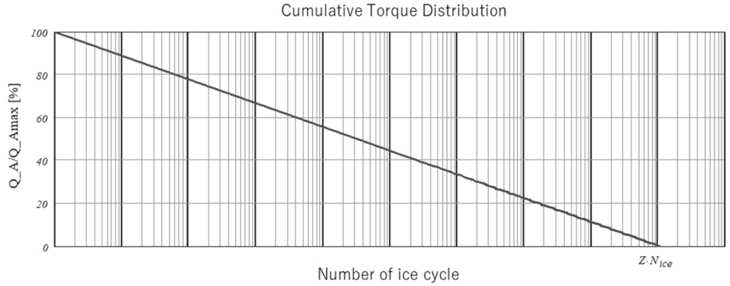
Amended	Original	Remarks
(a) Safety factor against yielding: 1.5 (b) Safety factor against fatigue: 1.5 (2) Safety factors for loads resulting from loss of propeller blades through plastic bending as defined in **4.4.9-1** are to be greater than 1.0 against yielding. (3) Provided that calculated stresses duly considering local stress concentrations are less than yield strength or a maximum of 70 % of σ_u of the respective materials, detailed fatigue analysis is not required. In other cases, however, components are to be analysed for cumulative fatigue, and an approach similar to that used for shafting assessment may be applied (See **4.5.5**). **2** Blade bolts (1) Blade bolts are to withstand the following bending moments considered around tangents on bolt pitch circles or other relevant axis for non-circular joints that are parallel to the root section considered: $M_{bolt} = SF_{ex}\left(0.8\frac{D}{2} - r_{bolt}\right)$ (kNm) where r_{bolt}: radius to the bolts plane S: Safety factor, taken as 1.0 (2) Blade bolt pre-tension is to be sufficient to avoid separation between mating surfaces when the maximum forward and backward ice loads defined in **4.4.3** to **4.4.8** (open and ducted propellers respectively) are applied. For conventional arrangements, the following formula is to be used: $d_{bb} = 41\sqrt[2]{\frac{F_{ex}\cdot(0.8D-d)\cdot S\cdot\alpha}{\sigma_{0.2}\cdot Z_{bb}\cdot PCD}}$ (mm) where	(Newly added)	Para. 6.4.2

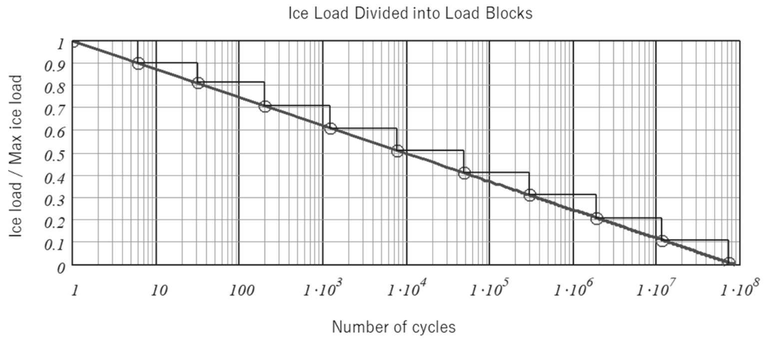
Amended	Original	Remarks
α: Factor based on the following bolt tightening methods. Other factors, however, may be used in cases where the Society deems it appropriate. Torque guided tightening: 1.6 Elongation guided: 1.3 Angle guided: 1.2 Other additional means: 1.1 d_{bb}: effective diameter of blade bolt in way of thread Z_{bb}: Number of blade bolts S: Safety factor, taken as 1.0 **3** CP mechanisms (1) Separate means (e.g. dowel pins) are to be provided in order to withstand the spindle torque resulting from blade failure Q_{sex} **(4.4.9)** or ice interaction Q_{smax} **(4.4.6)**, whichever is greater. In addition, other components of CP mechanisms are not to be damaged by the maximum spindle torques (Q_{sex} or Q_{smax}), and 1/3 of the spindle torque is to be assumed to be consumed by friction when not otherwise documented through further analysis. (2) Diameters of fitted pins d_{fp} between blades and blade carriers are to be obtained by the following formula: $$d_{fp} = 66\sqrt{\frac{(Q_s - Q_{fr})}{PCD \cdot z_{pin} \cdot \sigma_{0.2}}} \quad (mm)$$ where $Q_s = max(S \cdot Q_{smax}\ ;\ S \cdot Q_{sex})$ (kNm) S: Safety factor, taken as 1.3 for Q_{smax} and as 1.0 for Q_{sex} Q_{fr}: Friction between connected surfaces, taken as $0.33Q_s$. Alternative Q_{fr} calculations in accordance with reaction forces due to F_{ex} or F_f and F_b, whichever is relevant, may be used by utilising a friction coefficient = 0.15. In addition, stresses in actuating pins are to be obtained by	(Newly added)	Para. 6.4.3

Amended	Original	Remarks
the following formula: $\sigma_{vMises} = \sqrt{\left(\frac{\left(F\frac{h_{pin}}{2}\right)}{\frac{\pi}{32}}\right)^2 + 3\left(\frac{F}{\frac{\pi}{4}\cdot d_{pin}{}^2}\right)^2}$ (*MPa*) where $F = \frac{Q_s - Q_{fr}}{l_m}$ (*kN*) l_m: Distance pitching centre of blade to axis of pin (*m*) h_{pin}: Height of actuating pin (*mm*) d_{pin}: Diameter of actuating pin (*mm*) Q_{fr}: Friction torque in blade bearings acting on blade palms and caused by reaction forces due to F_{ex}, or F_f, F_b, whichever is relevant are to be taken as 1/3 of spindle torque Q_s (3) Blade failure spindle torque Q_{sex} is not to lead to any consequential damage, and fatigue strength is to be considered for parts transmitting the spindle torque from blades to servo systems in consideration of the ice spindle torque acting on one blade. In addition, maximum amplitude Q_{smax} is to be obtained by the following formula: $Q_{samax} = \frac{Q_{sb} + Q_{sf}}{2}$ (*kNm*) where Q_{sb}: Spindle torque due to $\lvert F_b \rvert$ (*mm*) Q_{sf}: Spindle torque due to $\lvert F_f \rvert$ (*mm*) **4** Servo pressures (1) Design pressures for servo systems are to be taken as the pressures caused by Q_{smax} or Q_{sex} when not protected by relief valves on the hydraulic actuator side or reduced by relevant friction losses in bearings caused by the respective ice loads.	(Newly added)	Para. 6.4.4

Amended	Original	Remarks
(2) Design pressures are not to be less than relief valve set pressure.		
4.5.5 Propulsion Line Components 1 General (1) The ultimate loads resulting from the total blade failure F_{ex} defined in **4.4.9** are to consist of combined axial and bending load components, wherever this is significant. In addition, the minimum safety factor against yielding is to be 1.0 for all shaft line components. (2) Shafts and shafting components (such as bearings, couplings and flanges) are to be designed to withstand operational propeller/ice interaction loads. (3) Obtained loads are not intended to be used for shaft alignment calculations, and cumulative fatigue calculations are to be conducted in accordance with Miner's rule. In addition, fatigue calculations are not necessary when maximum stress is below fatigue strength at 10^8 load cycles. (4) Torque and thrust amplitude distributions (spectrums) in propulsion lines are to be obtained by the following formula (Weibull exponent k = 1.0): $Q_A(N) = Q_{Amax}\left(1 - \frac{\log(N)}{\log(Z \cdot N_{ice})}\right)$ where ZN_{ice}: The number of load cycles in the load spectrum (5) The Weibull exponent to be considered is k = 1.0 for both open and ducted propeller torque and bending forces. Load distributions are accumulated load spectrums, and load spectrums are to be divided into a minimum of ten load blocks when using Miner's rules. The load spectrums used	**(Newly added)** (Newly added)	Para. 6.5

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
to count the number of cycles for 100 % load are to be the number of cycles above the next step (e.g. 90 % load) to ensure that calculations are on the conservative side, since calculated safety margins become more conservative as the number of stress blocks used decreases. An example of ice load distribution (spectrum) for shafting is shown in **Fig. 4.5.5-2.** (6) Load spectrums are to be divided into the number of load blocks (*nbl*) for the Miner's rules, the number of cycles for each load block is to be obtained by the following formula: $n_i = {N_{ice}}^{1-\left(1-\frac{i}{n_{bl}}\right)^k} - \sum_{i=1}^{i} n_{i-1}$ where *i*: Single load block *nbl*: Number of load blocks Fig. 4.5.5-1 Cumulative Torque Distribution 	(Newly added)	Figure 7

Amended	Original	Remarks
Fig. 4.5.5-2 Example of Ice Load Distribution (Spectrum) for the Shafting (k = 1.0)  	(Newly added)	Figure 7
2 Fitting propellers to shafts (1) Keyless cone mounting (a) Friction capacity at 0 °C is to be at least S = 2.0 times the highest peak torque Q_{peak} without exceeding the permissible hub stresses. (b) Necessary surface pressure $P_{0°C}$ is to be obtained by following formula: $P_{0°C} = \frac{2\cdot S\cdot Q_{peak}}{\pi\cdot\mu\cdot D_s{}^2\cdot L\cdot 10^3}$ (MPa) where μ : Coefficient of friction between metal materials applicable only to this requirement and obtained as follows. Coefficients are to be increased by 0.04 in cases where glycerin is used in wet mounting. Steel and steel: 0.15 Steel and bronze: 0.13 D_s: Shrinkage diameter at the mid-length of the taper (m) L: Effective length of taper (m) S: Safety factor, more than 2.0	(Newly added)	Para. 6.5.1 Para. 6.5.1.1

Amended	Original	Remarks
(2) Key mounting is not permitted. (3) Flange mounting (a) Flange thickness is to be at least 25 % of the required aft end shaft diameter (See **6.2.4-1** and **-2, Part K of the Rules**). (b) Additional stress raisers such as recesses for bolt heads are not to interfere with flange fillets unless flange thickness is increased correspondingly. (c) Flange fillet radii are to be at least 10 % of the required shaft diameter. (d) The diameter of shear pins is to be obtained by the following formula: $d_{pin} = 66\sqrt[2]{\frac{Q_{peak}\cdot S}{PCD\cdot z_{pin}\cdot \sigma_{0.2}}}\ (mm)$ where d_{pin}: Diameter of shear pins (mm) z_{pin}: Number of shear pins S: Safety factor, taken as 1.3 (e) Bolts are to be designed so that blade failure loads F_{ex} (**4.4.9**) in the backwards direction do not cause yielding of the bolts. The following formula is to be used: $d_b = 41\sqrt{\frac{F_{ex}\left(0.8\cdot\frac{D}{PCD}+1\right)\cdot\alpha}{\sigma_{0.2}\cdot z_b}}\ (mm)$ where α: Factor based on the following bolt tightening methods. Other factors, however, may be used in cases where the Society deems it appropriate. Torque guided tightening: 1.6 Elongation guided: 1.3		Para. 6.5.1.2 Para. 6.5.1.3

Amended	Original	Remarks
Angle guided: 1.2 Other additional means: 1.1 d_b: Diameter of flange bolt (*mm*) Z_b: Number of flange bolts		
3 Propeller shafts	(Newly added)	
(1) Blade failure loads F_{ex}		Para. 6.5.2
(a) Blade failure loads F_{ex} **(4.4.9)** applied parallel to shafts (forwards or backwards) are not to cause yielding, bending moments need not be combined with other loads. In addition, the diameter d_p in way of aft stern tube bearing are not to be less than the value of the following formula: $d_p = 160 \sqrt[3]{\frac{F_{ex} \cdot D}{\sigma_{0.2} \cdot \left(1 - \frac{{d_i}^4}{{d_p}^4}\right)}}$ (*mm*) where d_b: Propeller shaft diameter (*mm*) d_i: Propeller shaft inner diameter (*mm*)		Para. 6.5.2.1
(b) Forward of aft stern tube bearings, shaft diameters may be reduced based on direct calculation of the actual bending moment, or on the assumption that the bending moments caused by F_{ex} are linearly reduced to 25 % at the next bearing and in front of this linearly to zero at the third bearing.		
(c) Bending due to maximum blade forces F_b and F_f has been disregarded since the resulting stress levels are much lower than the stresses caused by the blade failure load.		
(2) Peak torque Q_{peak}		Para. 6.5.2.2
(a) Stresses due to the peak torque Q_{peak} are to have		

Amended	Original	Remarks
minimum safety factors of $S = 1.5$ against yielding in plain sections and $S = 1.0$ in way of stress concentrations in order to avoid bent shafts. (b) Minimum shaft diameters are to be obtained by the following formula. Notched shaft diameters, however, are not to be less than required plain shaft diameters. Plain shaft $d_p = 210 \sqrt[3]{\frac{Q_{peak} \cdot S}{\sigma_{0.2} \cdot \left(1 - \frac{{d_i}^4}{d^4}\right)}}$ (mm) Notched shaft $d_p = 210 \sqrt[3]{\frac{Q_{peak} \cdot S \cdot \alpha_t}{\sigma_{0.2} \cdot \left(1 - \frac{{d_i}^4}{d^4}\right)}}$ (mm) where α_t: Local stress concentration factor in torsion		
(3) Torque amplitudes (See **4.4.13**) of a corresponding number of load cycles are to be used in accumulated fatigue evaluation in which the safety factor is $S_{fat} = 1.5$. This is also to be considered when plants have high engine excited torsional vibrations (e.g. direct coupled 2-stroke engines).		Para. 6.5.2.3
(4) Fatigue strength (a) The fatigue strengths σ_F and τ_F (three million cycles) of shaft materials may be assessed by the following formula on the basis of the material's yield or 0.2 % proof strength: $\sigma_F = 0.436 \cdot \sigma_{0.2} + 77 = \tau_F \cdot \sqrt{3}$ (MPa) This is valid for small polished specimens (no notch) and reversed stresses. (See "**VDEH 1983 Bericht Nr. ABF11 Berechnung von Wöhlerlinien für Bauteile aus**		Para. 6.5.2.4

Amended	Original	Remarks
Stahl") (b) High cycle fatigue (*HCF*) is to be assessed based on the fatigue strengths in **(a)** above, notch factors (i.e. geometrical stress concentration factors and notch sensitivity), size factors, mean stress influence and at required safety factor of 1.6 at three million cycles increasing to 1.8 at 10^9 cycles. (c) Low cycle fatigue (*LCF*) representing 10^4 cycles is to be based on the smaller of yield or 0.7 of tensile strength $/\sqrt{3}$, and this criterion utilises a safety factor of 1.25. (d) The *LCF* and *HCF* given in **(b)** and **(c)** above represent the upper and lower knees in a stress-cycle diagram. Since the required safety factors are included in these values, a Miner's sum of unity is acceptable.		
4 Intermediate shafts are to be designed to satisfy **-3(2)** to **(4)** above.	(Newly added)	Para. 6.5.3
5 Shaft connections	(Newly added)	Para. 6.5.4
(1) Shrink fit couplings (keyless) are to be in accordance with **4.5.5-2(1)**. In such cases, a safety factor of 1.8 is to be used.		Para. 6.5.4.1
(2) Key mounting is not permitted.		Para. 6.5.4.2
(3) Flange mounting (a) Flange thickness is to be at least 20 % of the required shaft diameter (See **6.2.4-1** and **-2, Part D of the Rules**). (b) Additional stress raisers such as recesses for bolt heads are not to interfere with flange fillets unless flange thickness is increased correspondingly. (c) Flange fillet radii are to be at least 8 % of shaft diameters. (d) Diameters of ream fitted (i.e. light press fit) bolts are to		Para. 6.5.4.3

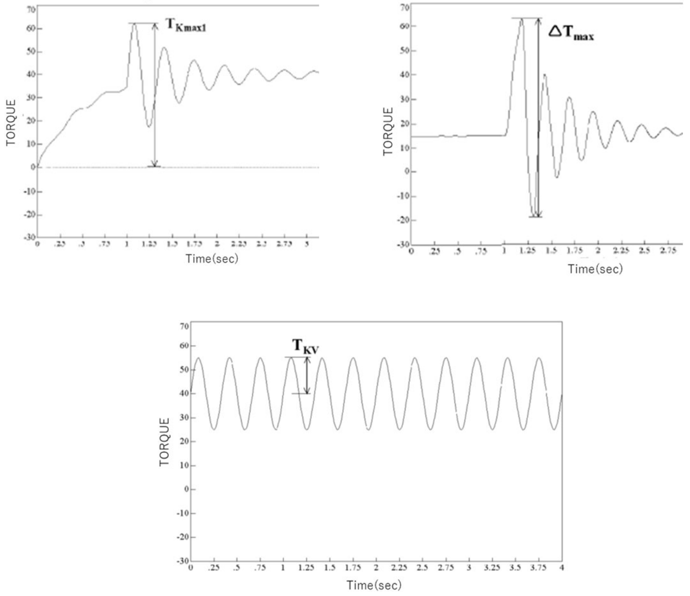
Amended	Original	Remarks
be chosen so that peak torque is transmitted with a safety factor of 1.9 in consideration of prestress. (e) Pins are to transmit the peak torque with a safety factor of 1.5 against yielding (See **-2(3)(d)**). (f) Bolts are to be designed so that blade failure loads F_{ex} (**4.4.9**) in the backwards direction do not cause yielding.		
(4) Splined shaft connections may be applied in cases where no axial or bending loads occur. In such cases, a safety factor of S = 1.5 against the allowable contact and shear stresses resulting from Q_{peak} is to be applied.		Para. 6.5.4.4
6 Gear transmissions	(Newly added)	Para. 6.5.4.5
(1) Shafts in gear transmissions are to satisfy the same safety levels as intermediate shafts, but bending stresses and torsional stresses are to be combined (e.g. by von Mises for static loads) where relevant. Maximum permissible deflection in order to maintain sufficient tooth contact pattern is to be considered for the relevant parts of the gear shafts.		Para. 6.5.4.6
(2) Gearing (a) The gearing is to satisfy the following three acceptance criteria: i) Tooth root stress ii) Pitting of tooth flanks iii) Scuffing (b) In addition to **(a)** above, criteria subsurface fatigue is to be considered, if necessary. (c) Common for all criteria is the influence of load distribution over face width. All relevant parameters such as elastic deflections (e.g. of mesh, shafts and gear		Para. 6.5.4.7

Amended	Original	Remarks
bodies), accuracy tolerances, helix modifications, and working positions in bearings (especially for multiple input single output gears) are to be considered. (d) Load spectrums (See **-1** above) are to be applied in such a way that the number of load cycles for output wheels are multiplied by a factor equaling the number of pinions on the wheel divided by number of propeller blades Z. For pinions and wheels operating at higher speeds, the number of load cycles is found by multiplication with the gear ratios. In addition, peak torque Q_{peak} is also to be considered during such calculations. (e) Cylindrical gears are to be assessed on the basis of the *ISO* 6336 series (i.e. *ISO* 6336-1:2019, *ISO* 6336-2:2019, *ISO* 6336-3:2019, *ISO* 6336-4:2019, *ISO* 6336-5:2016 and *ISO* 6336-6:2019), provided that “Method B” is used. **Annex 5.3.1, Part D of the Rules** may be applied provided that it is deemed equivalent by the Society. (f) The methods and standards applied to bevel gears are to be specially considered by the Society. (g) Tooth root safety is to be assessed against peak torque, torque amplitudes (with the pertinent average torque) and ordinary loads (open water free running) by means of accumulated fatigue analyses. The resulting safety factors are to be at least 1.5. (h) Safety against pitting is to be assessed in the same way as tooth root stresses but with a minimum resulting safety factor of 1.2. (i) Scuffing safety (flash temperature method – ref.		

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
ISO/TR 13989-1:2000 and *ISO/TR* 13989-2:2000) based on peak torque is to be at least 1.2 when the FZG class of oil is assumed one stage below specification.		
(j) Safety against subsurface fatigue of flanks for surface hardened gears (oblique fractures from active flank to opposite roots) is to be at the discretion of the Society. (It is, however, to be noted that high overloads can initiate subsurface fatigue cracks that may lead to a premature failure.)		
(3) Bearings are to be in accordance with **-10** below.		
(4) Torque capacity is to be at least 1.8 times the highest peak torque Q_{peak} (at the rotational speed) without exceeding the permissible hub stresses of 80 % yield.		Para. 6.5.4.8 Para. 6.5.4.9
7 Clutches	(Newly added)	
(1) Clutches are to have a static friction torque of at least 1.3 times the peak torque Q_{peak} and a dynamic friction torque 2/3 of the static friction torque.		Para. 6.5.5
(2) Emergency operation of clutches after failure of operating pressure is to be made possible within a reasonably short time. If this is arranged by bolts, it is to be on the engine side of the clutch in order to ensure access to all bolts by turning the engine.		
8 Elastic couplings	(Newly added)	
(1) There is to be a separation margin of at least 20 % between the peak torque and the torque where any twist limitation is reached. $Q_{peak} < 0.8T_{Kmax}(N = 1)$ (*kNm*)		Para. 6.5.6
(2) There is to be a separation margin of at least 20 % between the maximum response torque Q_{peak} (See **Fig. 4.4.13-3**) and the torque where any mechanical twist limitation or the		

Amended	Original	Remarks
permissible maximum torque of the elastic coupling, valid for at least a single load cycle ($N = 1$), is reached. (3) Sufficient fatigue strength is to be demonstrated at design torque level $Q_r(N=x)$ and $Q_A(N=x)$. This may be demonstrated by interpolation in a Weibull torque distribution (similar to **Fig. 4.5.5-1**) by the following formulae: $\frac{Q_r(N=x)}{Q_r(N=1)} = 1 - \frac{\log(x)}{\log(Z \cdot N_{ice})}$ $\frac{Q_A(N=x)}{Q_A(N=1)} = 1 - \frac{\log(x)}{\log(Z \cdot N_{ice})}$ where $Q_r(N=1)$ corresponds to Q_{peak} and $Q_A(N=1)$ corresponds to Q_{Amax}. $Q_r(N = 5E4) \cdot S < T_{Kmax}(N = 5E4)$ (kNm) $Q_r(N = 1E6) \cdot S < T_{KV}$ (kNm) $Q_A(N = 5E4) \cdot S < \Delta T_{max}(N = 5E4)$ (kNm) where S: General safety factor for fatigue, taken as 1.5 (4) Torque amplitude (or range Δ) is not to lead to fatigue cracking (i.e. not to exceed permissible vibratory torque). Permissible torque is to be determined by interpolation using a Weibull torque distribution in which T_{Kmax1} respectively ΔT_{max} refers to 50000 cycles and T_{KV} refers to 10^6 cycles (See **Fig.4.5.5-1**) $T_{Kmax1} \geq Q_r\ (5 \times 10^4\ load\ cycles)$ (kNm)		

Amended	Original	Remarks
Fig. 4.5.5-1 Example of T_{Kmax1}, ΔT_{max} and T_{KV} 	(Newly added)	Figure 9 - 11
9 Crankshafts Special consideration is to be given to plants with large inertia (e.g. flywheels, tuning wheels or PTO) in the non-driving end fronts of engines (opposite to main power take off).	(Newly added)	Para. 6.5.7
10 Bearings (1) Aft stern tube bearings and next shaft line bearings are to withstand the F_{ex} given in **4.4.9** in such a way that allows ships to maintain operational capability. (2) Rolling bearings are to have $L10a$ lifetimes of at least 40,000 *hours* according to *ISO* 281:2007. (3) Thrust bearings and their housings are to be designed to withstand with a safety factor S= 1.0 the maximum response thrusts in **4.4.11** and the axial forces resulting from the blade	(Newly added)	Para. 6.5.8

Amended	Original	Remarks
failure load F_{ex} in **4.4.9**. For the purpose of calculation, except for F_{ex}, shafts are assumed to rotate at rated speed. For pulling propellers, special consideration is to be given to loads from ice interaction on propeller hubs.		
11 Seals (1) Seals are to prevent egress of pollutants and be suitable for operating temperatures. In addition, contingency plans for preventing the egress of pollutants under failure conditions are to be documented. (2) Seals installed are to be suitable for the intended application. Manufacturers are to provide service experience in similar applications or testing results for consideration.	(Newly added)	Para. 6.5.9
(Deleted)	**4.3.1 General** In the design of the propulsion shafting system, the following are to be taken into account. (1) The propulsion shafting system is to have sufficient strength for the loads specified in **4.2**. (2) The blade failure load given in **4.2.9** is not to damage the propulsion shafting system other than the propeller blade itself. (3) The propulsion shafting system is to have sufficient fatigue strength.	
4.5.6 Azimuthing Main Propulsors In the design of the azimuthing main propulsors, the following are to be taken into account in addition to the requirements specified in **4.5.5**. (1) Loading cases which are extraordinary for propulsion units are to be taken into account. Estimation of the loading cases	**4.3.2 Azimuthing Main Propulsors** In the design of the azimuthing main propulsors, the following are to be taken into account in addition to the requirements specified in **4.3.1**. (1) Loading cases which are extraordinary for propulsion units are to be taken into account. Estimation of the loading cases	Para. 6.6

Amended	Original	Remarks
is to reflect the operational realities of the ship and the thrusters (for example, loads caused by the impact of ice blocks on propeller hubs of pulling propellers). Furthermore, loads resulting from thrusters operating at oblique angles to the flow are to be considered.	is to reflect the operational realities of the ship and the thrusters.	
(2) The steering mechanism, the fitting of the unit and body of the thruster are to be designed to withstand the loss of a blade without damage.	(2) The steering mechanism, the fitting of the unit and body of the thruster are to be designed to withstand the loss of a blade without damage.	
(3) The loss of a blade is to be considered in the propeller blade position, which causes the maximum load on the studied component. Typically, a top-down blade orientation leads to maximum bending loads acting on thruster bodies.	(3) The plastic bending of a blade is to be considered in the propeller blade position, which causes the maximum load on the studied component.	
(4) Azimuth thrusters are to be designed for estimated loads caused by thruster body/ice interaction, and the thruster bodies are to withstand the loads obtained when the maximum ice blocks given in **4.4.2** strike the thruster body when ships are at typical ice operating speed. In addition, the design situation in which ice sheets glide along ship hulls and presses against thruster bodies is to be considered in which sheet thicknesses are taken as the thickness of the maximum ice block entering the propeller, as defined in section **4.4.2**.	(4) Azimuth thrusters are to be designed for estimated loads specified in **3.4.10**.	
(Deleted)	**4.3.3 Propeller Blade** **1** Blade stresses are to be calculated using backward and forward loads given in **4.2.2** and **4.2.3**. The stresses are to be calculated with recognized and well documented FE-analysis or acceptable alternative methods. The backward load and the forward load are to be applied separately. **2** The calculated blade stress σ_{calc} for maximum ice load is	

Amended	Original	Remarks
(Deleted)	to comply with the following. $\sigma_{calc} < \frac{\sigma_{ref}}{S}$ Where $S = 1.5$ σ_{ref} : $0.7\sigma_u$ or $0.6\sigma_{0.2} + 0.4\sigma_u$, ($MPa$), whichever is less σ_u and $\sigma_{0.2}$: the stresses (MPa) as defined in **4.2.9-1** **4.3.4 Blade Edge Thickness** **1** The blade edge thickness and tip thickness are to be greater than the values obtained by the following formula. The requirement for edge thickness is to be applied for the leading edge and in case of reversible rotation open propellers, also the trailing edge. $S\ x\ S_{ice}\sqrt{\frac{3\ p_{ice}}{\sigma_{ref}}}$ (mm) x: Distance from the blade edge measured along the cylindrical sections from the edge and is to be 2.5% of chord length However not to be taken greater than 45mm. In the tip area (above 0.975R) the value is to be taken as 2.5% of 0.975R section length and is to be measured perpendicularly to the edge, however not to be taken greater than 45mm. S: Safety factor given below: S = 2.5 (for trailing edge) = 3.5 (for leading edge) = 5.0 (for tip) S_{ice}: Value specified in **Table 4.2.2-1** p_{ice}: Ice pressure =16 (MPa)	

Amended	Original	Remarks
	σ_{ref} : Value specified in **4.2.9-1** **2** Tip thickness is to be the maximum measured thickness in the tip area above 0.975R radius. The edge thickness in the area between the position of maximum tip thickness and edge thickness at 0.975R radius is to be interpolated between edge and tip thickness values and smoothly distributed.	
(Deleted)	**4.3.5 Controllable Pitch Propeller and Built-up Propeller** The strengths of the pitch control gear of the controllable pitch propeller and the blade bolts of the controllable pitch propeller, and the built-up propeller are to be evaluated in consideration of the stress generated when the loads in **4.2.4** and **4.2.9** act on the propeller blade. The safety factor is to be deemed appropriate by the Society.	
(Deleted)	**4.3.6 Shafting** **1** For evaluating the strength of shafting systems, twisting moment, bending moment and thrust which may be initiated by ice interaction with the propeller are to be taken into account. Safety factors for yielding and fatigue are to be deemed appropriate by the Society. **2** The strengths of the thrust shaft, intermediate shaft, propeller shaft and stern tube shaft are to be evaluated by calculating the maximum equivalent stresses (von Mises) on the shafts. **3** The strengths of the propeller shaft and connection parts of the propeller are also to be evaluated in consideration of the stress caused by the load given in **4.2.9** acting on the propeller blade.	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
4.6 Prime Movers	**4.4 Prime Movers**	Para. 7
4.6.1 Main Engines 1 Main engines are to be capable of being started and running propellers in the bollard condition. 2 Main engines are to be capable of being started and running the propeller with the controllable pitch in full pitch as limited by mechanical stoppers.	**4.4.1 Main Engines** (Newly added) The main engine is to be capable of being started and running the propeller with the controllable pitch in full pitch.	Para. 7.1
4.6.2 Starting Arrangements 1 The capacities of air receivers are to be sufficient to provide, without recharging, not less than 12 consecutive starts of propulsion engines, and not less than 6 consecutive starts when reversed for going astern in cases where propulsion engines do not need to be reversed for going astern. Air receivers serving other purposes in addition to starting propulsion engines are to have additional capacities sufficient for such purposes. 2 The capacities of air compressors are to be sufficient for charging air receivers from atmospheric to full pressure in 1 *hour*, except for ice class *PC*6 to *PC*1 ships for which propulsion engines need to be reversed for going astern. In such cases, compressors are to be able to charge receivers within 30 *minutes*.	**(Newly added)** (Newly added) (Newly added)	Para. 7.2
4.6.3 Emergency Generating Sets 1 Provisions are to be made for heating arrangements to ensure the ready starting of emergency power units from a cold state at an ambient temperature applicable to the polar class ship. 2 Emergency power units are to be equipped with starting devices with stored energy capabilities of at least three consecutive starts at the temperatures specified in -1 above, and sources of stored	**4.4.2 Starting Arrangement for Emergency Generating Sets** Provisions are to be made for heating arrangements to ensure that cold emergency power units are able to start at an ambient temperature applicable to the polar class ship. (Newly added)	Para. 7.3

Amended	Original	Remarks
energy are to be protected to preclude critical depletion by automatic starting systems, unless a second independent mean of starting is provided. In addition, a second source of energy is to be provided for an additional three starts within 30 *minutes*, unless manual starting can be demonstrated to be effective.		
4.7 Fastening Loading Accelerations	**4.5 Fastening Loading Accelerations**	Para. 8
4.7.1 Machinery Fastening Loading Accelerations Supports of essential equipment and main propulsion machinery are to be suitable for the accelerations given by the following formulae. Accelerations are to be considered as acting independently.	**4.5.1 Machinery Fastening Loading Accelerations** Supports of essential equipment and main propulsion machinery are to be suitable for the accelerations given by the following formulae. Accelerations are to be considered as acting independently.	Para. 8.1
(1) Maximum longitudinal impact acceleration at any point along the hull girder: $a_l = \left(\frac{F_{IB}}{\Delta}\right)\left\{[1.1\tan(\gamma + \phi)] + \left[\frac{7H}{L}\right]\right\}$ (*m/s*^{2})	(1) Maximum longitudinal impact acceleration at any point along the hull girder: $a_l = \left(\frac{F_{IB}}{\Delta}\right)\left\{[1.1\tan(\gamma + \phi)] + \left[\frac{7H}{L}\right]\right\}$ (*m/s*^{2})	Para. 8.2
(2) Combined vertical impact acceleration at any point along the hull girder: $a_v = 2.5\left(\frac{F_{IB}}{\Delta}\right)F_X$ (*m/s*^{2}) where F_X = 1.3 (at fore perpendicular) =0.2 (at midships) =0.4 (at aft perpendicular) =1.3 (at aft perpendicular for vessels conducting ice breaking astern) Intermediate values to be interpolated linearly.	(2) Combined vertical impact acceleration at any point along the hull girder: $a_v = 2.5\left(\frac{F_{IB}}{\Delta}\right)F_X$ (*m/s*^{2}) Where F_X = 1.3 (at fore perpendicular) =0.2 (at midships) =0.4 (at aft perpendicular) =1.3 (at aft perpendicular for vessels conducting ice breaking astern) Intermediate values to be interpolated linearly.	Para. 8.3
(3) Combined transverse impact acceleration at any point along hull girder: $a_t = 3F_i\frac{F_X}{\Delta}$ (*m/s*^{2})	(3) Combined transverse impact acceleration at any point along hull girder: $a_t = 3F_i\frac{F_X}{\Delta}$ (*m/s*^{2})	Para. 8.4

Amended	Original	Remarks
where F_X = 1.5 (at fore perpendicular) =0.25 (at midships) =0.5 (at aft perpendicular) =1.5 (at aft perpendicular for vessels conducting ice breaking astern) Intermediate values to be interpolated linearly. where ϕ: Maximum friction angle (*deg*) between steel and ice, normally taken as 10 *degrees* γ: Bow stem angle (*deg*) at the *UIWL* Δ: Displacement at the *UIWL* (*t*) *L*: Length of ship (*m*) defined in **2.1.2, Part A of the Rules** *H*: Distance (*m*) from the *UIWL* to the point being considered F_{IB}: Vertical impact force (*kN*) defined in **3.5.2** F_i: Force (*kN*) defined in **3.3.1-1(3)(b)**	Where F_X = 1.5 (at fore perpendicular) =0.25 (at midships) =0.5 (at aft perpendicular) =1.5 (at aft perpendicular for vessels conducting ice breaking astern) Intermediate values to be interpolated linearly. where ϕ: Maximum friction angle (*deg*) between steel and ice, normally taken as 10 *degrees* γ: Bow stem angle (*deg*) at the *UIWL* Δ: Displacement at the *UIWL* (*t*) *L*: Length of ship (*m*) defined in **2.1.2, Part A of the Rules** *H*: Distance (*m*) from the *UIWL* to the point being considered F_{IB}: Vertical impact force (*kN*) defined in **3.5.2** F_i: Force (*kN*) defined in **3.3.1-1(3)(b)**	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks
4.8 Auxiliary Systems and Piping Systems **4.8.1 Auxiliary Systems** **1** Machinery is to be protected from the harmful effects of ingestion or accumulation of ice or snow. Where continuous operation is necessary, means are to be provided to purge the system of accumulated ice or snow. **2** Means are to be provided to prevent tanks containing liquids to be damaged by freezing. **3** Vent pipes, intake and discharge pipes and associated systems are to be designed to prevent blockage due to freezing or ice and snow accumulation.	**4.6 Auxiliary Systems and Piping Systems** **4.6.1 Auxiliary Systems** **1** Machinery is to be protected from the harmful effects of ingestion or accumulation of ice or snow. Where continuous operation is necessary, means are to be provided to purge the system of accumulated ice or snow. **2** Means are to be provided to prevent tanks containing liquids to be damaged by freezing. **3** Vent pipes, intake and discharge pipes and associated systems are to be designed to prevent blockage due to freezing or ice and snow accumulation.	Para. 9
4.8.2 Sea Inlets and Cooling Water Systems **1** 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 polar class. **2** The construction of the sea chests is to comply with the following requirements: (1) At least two sea chests are to be arranged as ice boxes for *PC*1, *PC*2, *PC*3, *PC*4 and *PC*5 polar class ships. (2) At least one ice box is to be arranged preferably near the centerline for *PC*6 and *PC*7 polar class ships. (3) The calculated volume for each of the ice boxes is to be at least $1m^3$ for every $750kW$ of the engine output of the ship including the output of auxiliary engines. (4) Ice boxes are to be designed for an effective separation of ice and venting of air. (See example of **Fig. 4.8.2-1**) **3** Sea inlet valves are to be secured directly to the ice boxes or	**4.6.2 Sea Inlets and Cooling Water Systems** **1** 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 polar class. **2** The construction of the sea chests is to comply with the following requirements: (1) At least two sea chests are to be arranged as ice boxes for *PC*1, *PC*2, *PC*3, *PC*4 and *PC*5 polar class ships. (2) At least one ice box is to be arranged preferably near the centerline for *PC*6 and *PC*7 polar class ships. (3) The calculated volume for each of the ice boxes is to be at least $1m^3$ for every $750kW$ of the engine output of the ship including the output of auxiliary engines. (4) Ice boxes are to be designed for an effective separation of ice and venting of air. (See example of **Fig. 4.6.2-1**) **3** Sea inlet valves are to be secured directly to the ice boxes or	Para. 10

Amended	Original	Remarks
the sea bays. The valve is to be a full bore type.	the sea bays. The valve is to be a full bore type.	
4 Ice boxes and sea bays are to have vent pipes and to have shut off valves connected direct to the shell.	**4** Ice boxes and sea bays are to have vent pipes and to have shut off valves connected direct to the shell.	
5 Means are to be provided to prevent freezing of sea bays, ice boxes, ship side valves and fittings above the *LIWL*.	**5** Means are to be provided to prevent freezing of sea bays, ice boxes, ship side valves and fittings above the *LIWL*.	
6 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.	**6** 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.	
7 Detachable gratings or manholes are to be provided for ice boxes. Manholes are to be located above the *UIWL*.	**7** Detachable gratings or manholes are to be provided for ice boxes. Manholes are to be located above the *UIWL*.	
8 Openings in ship sides for ice boxes are to be fitted with gratings, or holes or slots in shell plates. The net area through these openings is to be not less than 5 times the area of the inlet pipe. The diameter of holes and width of the slot in shell plating is to be not less than 20*mm*.	**8** Openings in ship sides for ice boxes are to be fitted with gratings, or holes or slots in shell plates. The net area through these openings is to be not less than 5 times the area of the inlet pipe. The diameter of holes and width of the slot in shell plating is to be not less than 20*mm*.	
9 Gratings of the ice boxes are to be provided with a means of cleaning with a low pressure steam connection. Cleaning pipes are to be provided with screw-down type non return valves.	**9** Gratings of the ice boxes are to be provided with a means of cleaning with a low pressure steam connection. Cleaning pipes are to be provided with screw-down type non return valves.	
4.8.3 Ballast Tanks Efficient means are to be provided to prevent freezing in fore and after peak tanks and wing tanks located above the *LIWL* and where otherwise found necessary.	**4.6.3 Ballast Tanks** Efficient means are to be provided to prevent freezing in fore and after peak tanks and wing tanks located above the *LIWL* and where otherwise found necessary.	Para. 11
4.9 Ventilation System	**4.7 Ventilation System**	Para. 12
4.9.1 Ventilation System **1** The air intakes for machinery and accommodation ventilation are to be located on both sides of the ship at locations where manual de-icing is possible.	**4.7.1 Ventilation System** **1** The air intakes for machinery and accommodation ventilation are to be located on both sides of the ship.	

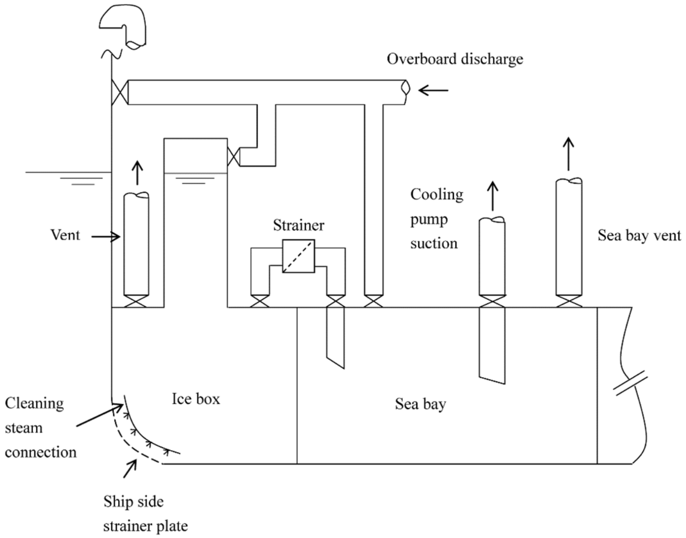
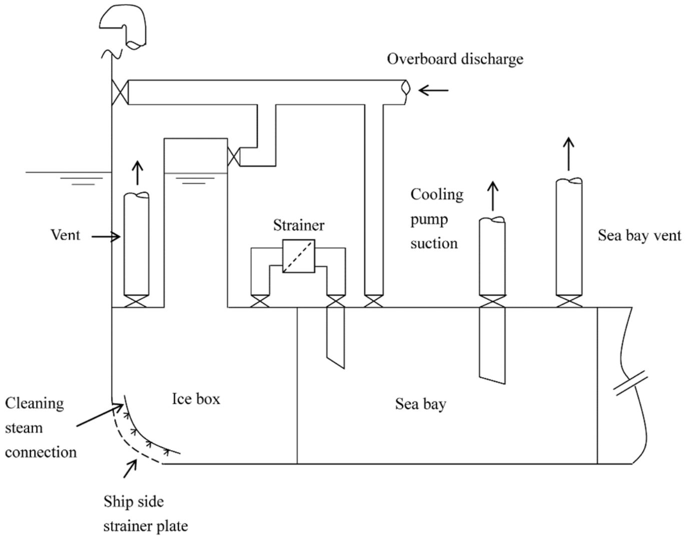
Amended	Original	Remarks
2 The air intakes specified in **-1** above may be provided with a means equivalent to the manual de-icing required in **-1** above when deemed appropriate by the Society.	**2** The air intakes specified in **-1** are to be provided with a means of heating.	
3 Multiple air intakes are to be provided for emergency generating sets, and such intakes are to be as far apart as possible.	**3** The temperature of inlet air provided to machinery from the air intakes is to be suitable for the safe operation of the machinery.	
4 Temperature of inlet air is to be suitable for the following purposes. In addition, accommodation and ventilation air inlets are, if necessary, to be provided with means of heating. (1) Safe operation of machinery (2) Thermal comfort in accommodation spaces	(Newly added)	
4.10 Rudders and Steering Arrangements	**4.8 Rudders and Steering Arrangements**	
4.10.1 Rudders and Steering Arrangements	**4.8.1 Rudders and Steering Arrangements**	
1 An ice knife is to be fitted to protect the rudder in the centre position against ice pressure. The ice knife is to be extended below the *LIWL*.	**1** An ice knife is to be fitted to protect the rudder against ice pressure. The ice knife is to be extended below the *LIWL*.	
2 Rudder stops to protect steering arrangements are to be provided, and the design ice force acting on rudders is to be transmitted to said rudder stops without damaging steering systems.	**2** Rudder stoppers to protect the steering arrangements are to be effective.	
3 The components of the steering gear are to be dimensioned to stand the yield torque of the rudder stock.	**3** The components of the steering gear are to be dimensioned to stand the yield torque of the rudder stock.	
4 Relief valves for hydraulic pressure of the steering arrangements are to be effective.	**4** Relief valves for hydraulic pressure of the steering arrangements are to be effective.	
4.10.2 Rudder Actuators	**(Newly added)**	Para. 13.2
1 Rudder actuators are to be designed for holding torque obtained by multiplying the open water torque specified in **15.2.2(1), Part D of the Rules** (in consideration of a maximum speed of 18	(Newly added)	

Amended	Original	Remarks
knots) by the factors specified in **Table 4.10.2-1**.		
2 Design pressures for calculations to determine the scantlings of rudder actuators are to be at least 1.25 times the maximum working pressure corresponding to the holding torque defined in **-1** above.	(Newly added)	
3 Rudder actuators are to be protected by torque relief arrangements, assuming the turning speeds (*deg/s*) specified in **Table 4.10.2-2** without undue pressure rise. If, however, rudder and actuator designs can withstand such rapid loads, such special relief arrangements are not necessary and conventional ones may be used instead.	(Newly added)	
4 For ship affixed with the additional notation "*Icebreaker*" (abbreviated to *ICB*), fast-acting torque relief arrangements are to be fitted in order to provide effective protection of rudder actuators in case where rudders are rapidly forced hard over against the stops.	(Newly added)	
5 For hydraulically operated steering gear, fast-acting torque relief arrangements are to be so designed that pressures cannot exceed 115 % of the set pressures of safety valves when rudders are forced to move at the speeds indicated in **Table 4.10.2-3**, and when taking into account oil viscosity at the lowest expected ambient temperatures in steering gear compartments.	(Newly added)	
6 For alternative steering systems, fast-acting torque relief arrangements are to demonstrate degrees of protection equivalent to that required for hydraulically operated arrangements.	(Newly added)	
7 Arrangements are to be designed such that steering capacity can be speedily regained.	(Newly added)	
Table 4.10.2-1 Factors for Holding Torque	(Newly added)	

Table 4.10.2-1 Factors for Holding Torque

	*PC*1 and *PC*2	*PC*3 to *PC*5	*PC*6 and *PC*7
Factor	5	3	1.5

<table>
<tr><th>Amended</th><th>Original</th><th>Remarks</th></tr>
<tr><td>Table 4.10.2-2 Turning Speeds of Steering Gear (Torque relief arrangements)
<table>
<tr><th></th><th>PC1 and PC2</th><th>PC3 to PC5</th><th>PC6 and PC7</th></tr>
<tr><td>Turning speeds (deg/s)</td><td>10</td><td>7.5</td><td>6</td></tr>
</table></td><td>(Newly added)</td><td>Table 17</td></tr>
<tr><td>Table 4.10.2-3 Turning Speeds of Steering Gear (Fast-acting torque relief arrangement)
<table>
<tr><th></th><th>PC1 and PC2</th><th>PC3 to PC5</th><th>PC6 and PC7</th></tr>
<tr><td>Turning speeds (deg/s)</td><td>40</td><td>20</td><td>15</td></tr>
</table></td><td>(Newly added)</td><td>Table 18</td></tr>
<tr><td>**4.11 Alternative Design**</td><td>**(Newly added)**</td><td>Para. 14</td></tr>
<tr><td>**4.11.1 Alternative Design**
As an alternative to this chapter, a comprehensive design study may be submitted and may be requested to be validated by an agreed test programme.</td><td>**(Newly added)**</td><td>Para. 14.1</td></tr>
</table>

Amended	Original	Remarks
Fig.4.8.2-1 Example of the Sea Inlets and Cooling Water Systems  	Fig.4.6.2-1 Example of the Sea Inlets and Cooling Water Systems  	

Amended-Original Requirements Comparison Table (Machinery of Polar Class Ships)

Amended	Original	Remarks

EFFECTIVE DATE AND APPLICATION

1. The effective date of the amendments is 1 July 2024.
2. Notwithstanding the amendments to the Rules, the current requirements apply to ships for which the date of contract for construction* is before the effective date.
   * "contract for construction" is defined in the latest version of IACS Procedural Requirement (PR) No.29.

IACS PR No.29 (Rev.0, July 2009)

1. The date of "contract for construction" of a vessel is the date on which the contract to build the vessel is signed between the prospective owner and the shipbuilder. This date and the construction numbers (i.e. hull numbers) of all the vessels included in the contract are to be declared to the classification society by the party applying for the assignment of class to a newbuilding.

2. The date of "contract for construction" of a series of vessels, including specified optional vessels for which the option is ultimately exercised, is the date on which the contract to build the series is signed between the prospective owner and the shipbuilder.
For the purpose of this Procedural Requirement, vessels built under a single contract for construction are considered a "series of vessels" if they are built to the same approved plans for classification purposes. However, vessels within a series may have design alterations from the original design provided:
(1) such alterations do not affect matters related to classification, or
(2) If the alterations are subject to classification requirements, these alterations are to comply with the classification requirements in effect on the date on which the alterations are contracted between the prospective owner and the shipbuilder or, in the absence of the alteration contract, comply with the classification requirements in effect on the date on which the alterations are submitted to the Society for approval.
The optional vessels will be considered part of the same series of vessels if the option is exercised not later than 1 year after the contract to build the series was signed.

3. If a contract for construction is later amended to include additional vessels or additional options, the date of "contract for construction" for such vessels is the date on which the amendment to the contract, is signed between the prospective owner and the shipbuilder. The amendment to the contract is to be considered as a "new contract" to which **1.** and **2.** above apply.

4. If a contract for construction is amended to change the ship type, the date of "contract for construction" of this modified vessel, or vessels, is the date on which revised contract or new contract is signed between the Owner, or Owners, and the shipbuilder.

Note:
This Procedural Requirement applies from 1 July 2009.