BUREAU VERITAS MARINE & OFFSHORE

RULE NOTE

AZIMUTH THRUSTERS IN ICE
FOR POLAR CLASS SHIPS AND ICEBREAKERS

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Rule Note
NR584

AZIMUTH THRUSTERS IN ICE FOR POLAR CLASS SHIPS AND ICEBREAKERS

April 2026

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Section 1 General

1 Application

1.1 General

1.1.1 Polar Class ships and icebreakers

This Rule Note applies to ships fitted with an azimuth propulsion system and assigned a service notation **icebreaker** or an additional class notation **POLAR CLASS**.

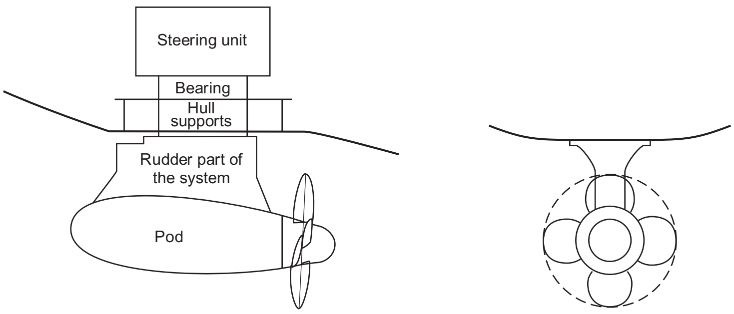
1.1.2 Types of azimuth propulsion systems

The azimuth propulsion system may be either a podded electrical thruster or a Z/L geared thruster constituted by the following sub-systems (see Fig 1):

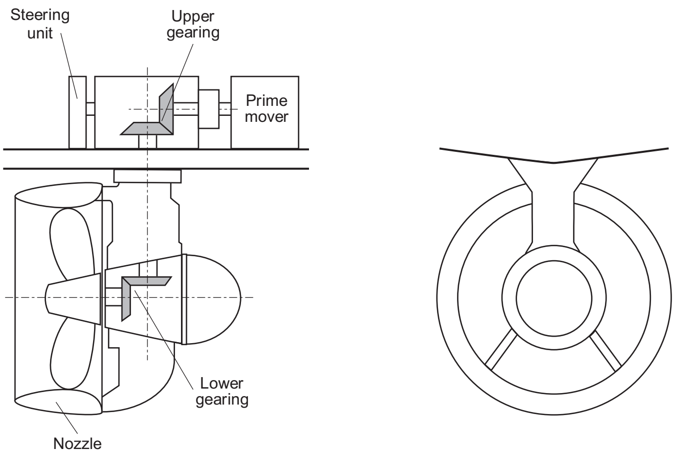
- for podded electrical thruster
  - the steering unit
  - the slewing bearing
  - the rudder part of the system
  - the pod which contains the electric motor
- for Z/L geared thruster
  - the steering unit
  - the upper gearing (for Z gear azimuth propulsion system)
  - the lower gearing
  - the nozzle
  - the prime mover, which may be either a diesel engine or electric motor.

Figure 1 : Podded electrical thrusters

Azimuth propulsion system arrangement



Z/L geared thrusters



1.2 Rule Note Structure

1.2.1 Rule Sections

This Rule Note contains 8 sections as summarised in Tab 1.

General requirements regarding the Rule Note application, material, mechanical, electrical and steering unit, for the azimuth propulsion systems intended for operation in both open sea and ice are provided in Sec 1, Sec 3, Sec 4 and Sec 5.

For the body structure the requirements applicable in open sea are provided in Sec 2 and those applicable for operation in ice are provided in Sec 6.

The tests and certification requirements are detailed in Sec 7 and Sec 8.

Table 1 : Overview of sections

<table>
<tr><th>Application and Materials</th><th>Body structure</th><th>Mechanical installation</th><th>Electrical installation</th><th>Steering unit</th><th>Tests</th><th>Certification</th></tr>
<tr><td rowspan="2">see Sec 1</td><td>Operations in open sea: see [1.2.3] and Sec 2</td><td rowspan="2">see [1.2.4], Sec 3 and Sec 7</td><td rowspan="2">see [1.2.5], Sec 4 and Sec 7</td><td rowspan="2">see [1.2.6], Sec 5 and Sec 7</td><td rowspan="2">see Sec 7</td><td rowspan="2">see Sec 8</td></tr>
<tr><td>Operations in ice: see Sec 6</td></tr>
</table>

1.2.2 SOLAS regulations

SOLAS regulations in the text are printed in Italic font. In reproducing such regulations in the present Rule Note applicable for the purpose of classification, the word “Administration”, wherever mentioned, has been replaced by the word “Society”.

1.2.3 Body structure

The requirements of the Sec 2 apply to the scantlings of the hull supports and the azimuth propulsion system body.

1.2.4 Mechanical installation

The Sec 3 and Sec 7 apply respectively:

- to the design, construction and installation,
- to the tests and trials,

of main propulsion and essential auxiliary machinery systems and associated equipment, piping systems as indicated in this Rule Note ans and as far as class is concerned only.

1.2.5 Electrical installation

The Sec 4 applies to the electrical installations and equipment of azimuth propulsion systems. In particular, it applies to the electrical propulsion motors, electrical slip rings and all electrical components of auxiliary services such as steering unit, cooling and lubrication systems. All auxiliary services vital to the operation of the azimuth thruster are to be considered as essential services for the application of this Rule Note.

1.2.6 Steering unit

Unless otherwise specified, the requirements of the Sec 5 apply to the steering units intended for thrusters used as means of propulsion.

1.3 Documentation to be submitted

1.3.1 General

The documentation listed in Tab 2 is to be submitted.

All listed plans are to be constructional plans complete with all dimensions and are to contain full indication of types of materials employed and where applicable by the welding details and welding procedures.

The Society reserves the right to request the submission of additional documents in the case of non-conventional design or if it is deemed necessary for the evaluation of the system, equipment or components.

1.3.2 Propellers

The documentation listed in Tab 3 is to be submitted for solid propellers intended for propulsion.

The documentation listed in Tab 4, as applicable, is to be submitted for built-up and controllable pitch propellers intended for propulsion.

For very highly skewed propellers and propellers of unusual design, in addition to the documentation listed in Tab 3 and Tab 4, as applicable, a detailed hydrodynamic load and stress analysis is to be submitted as per NR467, Pt C, Ch 1, Sec 8, |2.4.3].

1.3.3 Additional data to be submitted

The data and documentation listed in Tab 5 are to be submitted by the manufacturer in addition to the documents listed in Tab 3 and Tab 4.

1.3.4 Shipboard tests

A comprehensive list of the shipboard tests intended to be carried out by the shipyard is to be submitted to the Society.

For each test, the following information is to be provided:

- scope of the test
- parameters to be recorded.

Table 2 : Documentation to be submitted

No.	A/I **(1)**	Documentation	Particulars
1	I	General arrangement of the thruster	Including cross sections and sealing arrangements exposed to sea
2	I	General layout of the thruster	
3	A	Azimuth thruster general specification	Including rating, operation modes, limitations and maximum design loads (Torque, thrust, hydrodynamic loads, loads transferred to hull, steering loads and cycles) on the steering-propulsion unit, as well as material specifications for principal parts/assemblies and welding procedures
4	I	Environmental conditions	
5	A	Failure Mode and Effects Analysis of the propulsion	For the cooling, lubrication, ventilation systems, sealing system, steering functions of the thruster, thruster control system and fire safety
6	A	Process and Instrumentation of azimuth thruster	Including all the auxiliary services
7	A	Specification and data sheet of the auxiliary services	
8	A	Arrangements and details of the auxiliary services	
9	A	Diagrams of cooling, ventilation, draining/bilge, lubricating and hydraulic systems	
10	I	Description of the operational modes intended for steering in normal and emergency conditions	
11	I	Operation manual of the thruster in ice conditions	
12	A	Structural drawings of the thruster	
13	A	Structural connection to hull	
14	A	Arrangement of the azimuth propulsion system supports	
15	A	Ice load calculation of the thruster body	
16	A	Stress calculation for the thruster body	
17	A	Details of structural items (nozzle, bracing, etc.)	Including information on clearance between propeller and nozzle
18	I	Maximum propulsive power in ice conditions	As function of thruster steering angle and shaft rotational speed
19	A	Propeller details	including the applicable details mentioned in [1.3.2]
20	A	Detailed drawing of the propeller shaft	Including the brake
21	A	Propeller and intermediate shafts	
22	I	Installation drawing of the bearings	
23	A	Bearing details	
24	I	Bearing data	Including maximum allowable static and dynamic load capacity, permissible rotational speed range, axial displacement, operating conditions...
25	I	Dynamic load calculation of the bearings	
26	I	Lifetime calculation of the bearings	
27	A	Seals drawings	
28	A	Gears details	Including the applicable details required by the Society
29	A	Piping systems connected to thruster	
30	I	Assembly longitudinal and transverse sectional drawings of hydraulic pumps of power units, and their characteristic curves	
31	A	General arrangement of electrical installations	
32	A	Specification and data sheet of main electrical components	
(1) A: To be submitted for approval, I: To be submitted for information.			

No.	A/I **(1)**	Documentation	Particulars
33	A	Schematic electrical diagrams of main electrical components	
34	A	General arrangement diagram of main electrical components	
35	A	Thruster control system	
36	A	Diagram of the supply for monitoring and control systems of propulsion motors	
37	A	Diagram of the supply, monitoring and control systems of controllable pitch propellers	
38	A	Principles of control system and its power supply	
39	A	Alarm and monitoring system details	Including: • list of alarms and monitoring points • power supply diagram.
40	A	Safety system details	Including: • list of monitored parameters for safety system • power supply diagram.
41	A	Test program for electrical propulsion motors and electrical slip rings	
42	A	Diagrams of the electric power circuits	
43	A	Functional diagram of control, monitoring and safety systems	Including the remote control from the navigating bridge, with indication of the location of control, monitoring and safety devices
44	I	Operating manuals of the steering unit and of its main components	
45	I	General arrangement of the steering system	Including the information about the main components of the steering system (steering bearing and steering drive), protection and locking devices
46	I	Assembly drawing of the steering unit	Including sliding blocks, guides, stops and other similar components
47	I	General description of the steering unit installation and of its functioning principle	
48	A	Detailed drawings of the main components of the steering system	Including locking device and of all load carrying components
49	A	Rotating mechanism of the thruster	
50	A	For hydraulic steering unit, the schematic layout of the hydraulic piping of power actuating systems	Including the hydraulic fluid refilling system, with indication of: • the design pressure • the maximum working pressure expected in service • the diameter, thickness, material specification and connection details of the pipes • the hydraulic fluid tank capacity • the flashpoint of the hydraulic fluid
51	A	Assembly drawings of the actuators and constructional drawings of their components	Including, for hydraulic actuators, indication of: • the design torque • the maximum working pressure • the relief valve setting pressure
52	I	Constructional drawings of the relief valves for protection of the hydraulic actuators	Including indication of: • the setting pressure • the relieving capacity
53	A	Diagram of the supply, monitoring and control systems of the steering unit	
54	A	Risk analysis of single failure where the main steering unit comprises two or more identical power units	
55	A	Risk analysis of hydraulic locking, where more than one steering unit can be simultaneously operated	
(1) A: To be submitted for approval, I: To be submitted for information.			

Table 3 : Additional documentation to be submitted for solid propellers

No.	A/I **(1)**	Documentation
1	A	Sectional assembly
2	A	Blade and hub details
3	I	Rating (power, rpm, etc.)
4	A	Data and procedures for fitting propeller to the shaft
(1) A: To be submitted for approval, I: To be submitted for information		

Table 4 : Additional documentation to be submitted for built-up and controllable pitch propellers

No.	A/I **(1)**	Documentation
1	A	Sectional assembly
2	A	Blade and hub details
3	I	Rating (power, rpm, etc.)
4	A	Data and procedures for fitting propeller to the shaft
5	A	Blade bolts and pre-tensioning procedures
6	I	Pitch corresponding to maximum propeller thrust and to normal service condition
7	A	Pitch control mechanism
8	A	Pitch control hydraulic system
(1) A: To be submitted for approval, I: To be submitted for information		

Table 5 : Additional data and documentation to be submitted by manufacturer

No.	A/I **(1)**	Documentation	Particulars
1	I	Onboard installation procedure	
2	I	Rated power and revolutions	
3	I	Rated thrust	
4	A	Material specifications of the major parts	Including their physical, chemical and mechanical properties
5	A	Particulars on the design of welded joints, welding procedures, heat treatments and non-destructive examinations after welding	Where parts of thrusters are of welded construction
6	I	Background information on previous operating experience in similar applications	Where applicable
(1) A: To be submitted for approval, I: To be submitted for information			

1.4 Definitions

1.4.1 Auxiliary steering gear

Auxiliary steering gear is the equipment other than any part of the main steering gear necessary to steer the ship in the event of failure of the main steering gear but not including the tiller, quadrant or components serving the same purpose.

Steering gear control system is also understood to cover the equipment required to control the steering gear power actuating system.

1.4.2 Azimuth thruster

An azimuth thruster is a thruster which has the capability to rotate through 360° around a vertical axis in order to develop thrust in any direction.

1.4.3 Declared steering angle limits

Operational limits in terms of maximum steering angle, or equivalent, according to manufacturers' guidelines for safe operation, also taking into account the ship's speed or propeller torque/speed or other limitation.

The “declared steering angle limits” are to be declared by the directional control system manufacturer for each ship specific non-traditional steering mean; ship manoeuvrability tests, such as those in the Standards for ship manoeuvrability (IMO Resolution MSC.137(76)) are to be carried out with steering angles not exceeding the declared steering angle limits.

1.4.4 Earthing

The earth connection to the general mass of the hull of the ship in such a manner as will ensure at all times an immediate discharge of electrical energy without danger.

1.4.5 Frequency transient

Sudden change in frequency which goes outside the frequency tolerance limits and returns to and remains inside these limits within a specified recovery time after initiation of the disturbance (time range: seconds).

1.4.6 High-voltage systems

Alternating current systems with rated voltages greater than 1 000 V r.m.s. and direct current systems with a maximum instantaneous value of the voltage under rated operating conditions greater than 1 500 V.

1.4.7 Hydraulic locking

Hydraulic locking means all situations where two hydraulic systems (usually identical) oppose each other in such a way that it may lead to loss of steering. It can either be caused by pressure in the two hydraulic systems working against each other or by hydraulic “bypass” meaning that the systems puncture each other and cause pressure drop on both sides or make it impossible to build up pressure.

1.4.8 Local control station

Place of control where a system is installed which creates a reference value for the convertors independent from the remote control system and any external limitations.

1.4.9 Low-voltage systems

Alternating current systems with rated voltages greater than 50 V r.m.s. up to 1 000 V r.m.s. inclusive and direct current systems with a maximum instantaneous value of the voltage under rated operating conditions greater than 50 V up to 1 500 V inclusive.

1.4.10 Main control station

Place of control of the main propulsion system which is manned under seagoing conditions.

1.4.11 Main steering gear

Main steering gear is the machinery, actuators, steering gear power units, if any, and ancillary equipment and the means of applying torque to the thruster necessary for effecting movement of the thruster for the purpose of steering the ship under normal service conditions.

1.4.12 Maximum ahead service speed

Maximum ahead service speed is the greatest speed which the ship is designed to maintain in service at sea at the deepest seagoing draught.

1.4.13 Maximum astern speed

Maximum astern speed is the speed which it is estimated the ship can attain at the designed maximum astern power at the deepest seagoing draught.

1.4.14 Maximum working pressure

Maximum working pressure is the maximum expected pressure in the system when the steering gear is operated to comply with the provisions of Sec 5, [2.4.1].

1.4.15 Peak steering torque

Peak steering torque is the maximum expected torque corresponding to a large course change. The peak steering torque is to be specified by the thruster manufacturer.

1.4.16 Podded thruster

Propulsion system in which the motor is located in a dedicated, submerged unit (pod housing) of the ship.

1.4.17 Power actuating system

Power actuating system is the hydraulic equipment provided for supplying power to turn the thruster, comprising a steering gear power unit or units, together with the associated pipes and fittings, and an actuator. The power actuating systems may share common mechanical components, or components serving the same purpose.

1.4.18 Remote control system

System which comprises all equipment necessary to operate units from a control position where the operator cannot directly observe the effect of his actions.

1.4.19 Safety voltage

A voltage which does not exceed 50 V a.c. r.m.s. between conductors, or between any conductor and earth, in a circuit isolated from the supply by means such as a safety isolating transformer.

1.4.20 Steering gear

The steering gear is the machinery, actuators, power units, and ancillary equipment applied to turn the thruster about its axis of rotation in both directions for the purpose of steering the ship.

1.4.21 Steering actuating system

The steering actuating system consists of a steering gear power unit, a steering actuator and, for hydraulic or electro-hydraulic steering gears, the hydraulic piping.

1.4.22 Steering actuator

A steering actuator is a steering gear component which converts power into mechanical action to control the rotation of the thruster. It includes the following equipment:

- in case of electric steering: electric motor and driving pinion, worm gear, screw or similar components intended to transmit a force to the slewing ring
- in case of electro hydraulic steering: hydraulic motor and driving pinion, or similar components intended to transmit a force to the slewing ring.

1.4.23 Steering gear control system

Steering gear control system is the equipment by which orders are transmitted from the navigation bridge to the steering gear power units. Steering gear control systems comprise transmitters, receivers, hydraulic control pumps and their associated motors, motor controllers, piping and cables.

Steering gear control system is also understood to cover the equipment required to control the steering gear power actuating system.

1.4.24 Steering gear power unit

Steering gear power unit is:

- *in the case of electric steering gear, an electric motor and its associated electrical equipment*
- *in the case of electrohydraulic steering gear, an electric motor and its associated electrical equipment and connected pump*
- *in the case of other hydraulic steering gear, a driving engine and connected pump.*

For the purposes of alternative steering arrangements, the above definition is to be considered. The electric steering motors are to be considered as part of the power unit and actuator.

1.4.25 Steering system

Steering system means ship's directional control system, including main steering gear, auxiliary steering gear and steering gear control system.

1.4.26 Thruster

A thruster is a propeller installed in a revolving nozzle. A thruster may be intended for propulsion, manoeuvring and steering or any combination thereof. Propulsion propellers in fixed nozzles are not considered thrusters.

1.4.27 Voltage transient

Sudden change in voltage (excluding spikes) which goes outside the nominal voltage tolerance limits and returns to and remains inside these limits within a specified recovery time after the initiation of the disturbance (time range: seconds).

1.4.28 Hull Supports

Hull supports mean the steel structures of the ship hull adjacent to the azimuth thruster, including but not limited to the following: primary supporting members, stiffeners, plating, brackets, etc. The purpose of hull supports is to sustain the loads transmitted from azimuth thruster to the ship hull.

1.5 Environmental conditions

1.5.1 Ambient air temperatures

Machinery and electrical installations covered by this Rule Note are to be designed to operate properly under the ambient air conditions specified in Tab 6, unless otherwise specified.

For ships intended for long-term operation under low air temperature conditions, the requirements of NR467, Pt F, Ch 8, Sec 4 are to be applied additionally.

Table 6 : Ambient air temperatures

Location, arrangement	Temperature range, in °C	
	Unrestricted navigation **(1)**	Navigation outside the tropical zone
In enclosed spaces	0.... +45 **(2)**	0.... +40 **(2)**
On machinery components In spaces subject to higher or lower temperatures	According to specific local conditions	
On exposed decks	–25.... +45 **(2)**	–25.... +40 **(2)**

(1) For ships not intended for unrestricted navigation the Society may approve other temperatures.
(2) Electronic elements and devices designed for mounting in the switchboards, panels or casings are to be capable of reliable performance at an ambient air temperature up to 55°C.

1.5.2 Humidity

For ships classed for unrestricted service, the humidity ranges shown in Tab 7 are applicable in relation to the various locations of installation.

1.5.3 Sea water temperatures

The sea water temperatures for the machinery installations are those specified in Tab 8.

1.5.4 Inclinations

Electrical equipment and components are to be capable of reliable performance at angles of inclination specified in Tab 9.

1.5.5 Vibrations

Electrical equipment and components are to be designed to operate at vibrations levels specified in Tab 10.

1.5.6 Shocks

Electrical equipment and components are to be also capable of reliable performance at shocks having an acceleration of +5.0 g and at a frequency of 40 to 80 shocks per minute.

Table 7 : Humidity

Location	Humidity
General	95% up to 45°C 70% above 45°C
Air conditioned areas	Different values may be considered on a case-by-case basis

Table 8 : Sea water temperature

Temperature range, in °C	
Unrestricted navigation	Navigation outside the tropical zone
+32	+25

Table 9 : Inclination of ship

Type of machinery, equipment or component	Angles of inclination, in degrees **(1)**			
	Athwartship		Fore-and-aft	
	Static	Dynamic **(2)**	Static	Dynamic **(3)**
Machinery and electrical equipment relative to propulsion system	15	22,5	5	7,5
Switchgear and associated electrical and electronic components and remote control systems **(4)**	22,5	22,5	10	10

(1) Athwartship and fore-and-aft angles may occur simultaneously in their most unfavourable combination.
(2) The period of dynamic inclination may be assumed equal to 10 s.
(3) The period of dynamic inclination may be assumed equal to 5 s.
(4) No undesired switching operations or functional changes may occur up to an angle of inclination of 45°.

Table 10 : Vibration levels

Location	Frequency range, in Hz	Displacement amplitude, in mm	Acceleration amplitude g
In machinery spaces, command and control stations	from 2,0 to 13,2	1,0	–
	from 13,2 to 100	–	0,7
In podded thrusters	from 2,0 to 25,0	1,6	–
	from 25,0 to 100	–	4,0

1.6 Thruster installation

1.6.1 Thruster compartment

The compartment containing the thrusters is to be separate from machinery spaces of category A and equipped with appropriate ventilating (see Sec 4, [2.3], fire extinguishing, drainage (see Sec 4, [2.4.3] heating (see Sec 4, [2.4.2]) and lighting arrangements.

1.6.2 Access

An easy access is to be provided to component parts of the thrusters to allow their maintenance within the scope stipulated by the Service Manual.

See also Sec 4, [1.5.2], Sec 4, [2.4.3], Sec 4, [4.1.5], Sec 4, [4.2.1] and Sec 5, [2.2.1]

2 Materials

2.1 Materials for azimuth thrusters

2.1.1 General

Materials of azimuth thruster are to be made of approved steel or another ductile material.

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

2.1.2 Materials exposed to sea water temperature

The materials of components exposed to sea water temperature, such as propeller blades, propeller hubs, and thruster body, are to have an elongation of no less than 15% in a test specimen, the gauge length of which is five times the diameter.

A Charpy V impact test is to be carried out for materials other than bronze and austenitic steel, according to NR527, Sec 3, [2.1.1].

For the propeller shaft, blade bolts, CP mechanisms, shaft bolts, strut-pod connecting bolts, etc., an average impact energy value of 20 J, based on three tests, must be obtained at minus 10°C. It does not apply to surface-hardened components, such as bearings and gear teeth or sea water cooling lines (heat exchangers, pipes, valves, fittings, etc.).

The nodular cast iron of a ferrite structure type may be used for relevant parts other than bolts. The average impact energy for nodular cast iron is to be a minimum of 10 J at minus 10°C.

Note 1: The materials of the thruster components are also to comply with the relevant requirements of the Society for ships with **POLAR CLASS** notations.

2.1.3 Additional requirements for the body thruster

When the body thruster is made of steel plates, the material grade is not to be less than the one defined in Tab 11.

Table 11 : Material grade requirement for the body thruster steel platings

Thickness, in mm	Mild steel	High tensile steel
$t \leq 15$	A	AH
$15 < t \leq 20$	A	AH
$20 < t \leq 25$	B	AH
$25 < t \leq 30$	D	DH
$30 < t \leq 35$	D	DH
$35 < t \leq 40$	D	DH
$40 < t \leq 50$	E	EH

3 Ship requirements

3.1 Steering propulsion units

3.1.1 Number of steering propulsion units

Ships intended for unrestricted service are to be provided with at least two independent steering propulsion units when these are the sole means of propulsion and steering.

3.1.2 Use of single steering propulsion unit

Where a single steering-propulsion unit installation is considered, a detailed risk analysis is to be carried out in a form of FMEA (Failure Modes and Effects Analysis) or other effective methodology in order to ascertain that a single failure in the steering gear, control system and power supply does not result in the complete loss of the manoeuvring capability of the ship.

4 Azimuth thruster design principle

4.1 General

4.1.1 Ice loads

Azimuth thrusters are to be designed for estimated loads caused by thruster body/ice interaction in addition to loads applicable in open water as stated in Sec 3, [1.1.2].

The thruster body is to withstand the loads obtained when the maximum ice blocks, given in Sec 3, [2.2.1], strike the thruster body when the ship is at a typical ice operating speed. In addition, the design situation in which an ice sheet glides along the ship's hull and presses against the thruster body is to be considered. The thickness of the ice sheet is to be taken as the thickness of the maximum ice block entering the propeller, as defined in Sec 3, [2.2.1].

4.1.2 Load cases

The estimation of load cases is to reflect the way the thrusters are intended to operate on the specific ship. In this respect, for example, the loads caused by the impacts of ice blocks on the propeller hub of a pulling propeller are to be considered.

Furthermore, loads resulting from the thrusters operating at an oblique angle to the flow are to be considered.

4.1.3 Loss of a blade

The steering mechanism, the fitting of the unit, and the body of the thruster are to be designed to withstand the loss of a blade without damage. The loss of a blade is to be considered for the propeller blade orientation which causes the maximum load on the component being studied. Typically, top-down blade orientation places the maximum bending loads on the thruster body.

Section 2 Structure of the Thruster Body

Symbols

k : Material factor given as follows:

- for R_{eH} = 235 MPa, k = 1,00
- for R_{eH} = 315 MPa, k = 0,78
- for R_{eH} = 355 MPa, k = 0,72
- for R_{eH} = 390 MPa, k = 0,68

For intermediate values of R_{eH}, k is obtained by linear interpolation.

Steels with a specified minimum yield stress R_{eH}, greater than 390 MPa are considered by the Society on a case-by-case basis.

R_{eH} : Specified minimum yield stress of the material, in MPa.

1 Azimuth propulsion system

1.1 General

1.1.1 Application

The requirements of this Section apply to the scantlings of the hull supports and the azimuth thruster body.

The steering unit and the bearing are to comply with the requirements in Sec 5.

1.1.2 Operating conditions

The maximum angle at which the azimuth propulsion system can be oriented on each side when the ship navigates at its maximum speed is to be specified by the Designer. Such maximum angle is generally to be less than 35° on each side.

In general, orientations greater than this maximum angle may be considered by the Society for azimuth propulsion systems during manoeuvres, provided that the orientation values together with the relevant speed values are submitted to the Society for approval.

1.2 Locking device

1.2.1 General

The azimuth propulsion system is to be mechanically lockable in a fixed position in any angular positions, in order to avoid rotations of the system and propulsion in undesirable directions in the event of damage.

1.3 Design loads

1.3.1 Lateral pressure

The lateral pressure to be considered for scantling verification of plating and stiffeners of the azimuth propulsion system is to be determined for all orientations of the system up to the maximum angle at which the azimuth propulsion system can be oriented on each side. The design lateral pressure P, in kN/m², is the greatest pressure obtained for all configurations combining the orientation of the azimuth propulsion system in each side and the corresponding maximum speed the ship can be operated with.

The total force which acts on the azimuth propulsion system is to be obtained by integrating the lateral pressure on the external surface of the system.

The calculations of lateral pressure and total force are to be submitted to the Society for information.

1.4 Plating

1.4.1 Plating of the azimuth body part

The thickness of plating of the body part of the azimuth propulsion system, in mm, is to be not less than the values obtained from the following formulae rounded to the nearest half millimetre:

- for plating of body part considered as flat:

$$t = 5{,}5 s\beta \sqrt{k\left(T + \frac{P_b}{10}\right)} + 2{,}5$$

where:

s : Length, in m, of the shorter side of the plate panel

β : Coefficient equal to:

$$\beta = \sqrt{1{,}1 - 0{,}5\left(\frac{s}{b}\right)^2}$$

to be taken not greater than 1,0 if b/s > 2,5

T : Draught of the ship, in m, intended to receive the thruster

b : Length, in m, of the longer side of the plate panel

P_b : Design pressure, in kN/m², applied at half height of the considered element of the body part, determined as per [1.3.1]

- for plating of body part considered as rounded:

$$t = 6{,}45(P_p \cdot s_p)^{0{,}4} R_p^{0{,}6} 10^{-1} + 3$$

where:

P_p : Design pressure, in kN/m², applied at half height of the considered element of the body part, determined as per [1.3.1]

s_p : Spacing, in m, of the stiffeners

R_p : Radius, in m, of the shell

1.4.2 Webs

The web thickness of the body part supporting the propulsion module is to be at least 70% of that required in [1.4.1] in the same body part area and in no case to be less than 8 mm.

1.5 Stiffeners of the body part supporting the propulsion module

1.5.1 Section modulus

The section modulus Z, in cm³, of stiffeners of the body part supporting the propulsion module is not to be less than the following value:

$$Z = 1{,}2 \cdot \frac{P \cdot s \cdot \ell^2}{f_{bdg} \cdot 0{,}9 \cdot R_{eH}} 10^3$$

where:

P : Design pressure, in kN/m², acting on the propulsion module, calculated at the mean elevation of the propulsion shaft determined as per [1.3.1]

s : Stiffener spacing, in m

ℓ : Span of the stiffener, in m

f_{bdg} : Bending moment factor taken as:

- for vertical stiffener with fixed ends:

  $f_{bdg} = 12$

- for vertical stiffener with simple support ends:

  $f_{bdg} = 8$

1.5.2 Web thickness

The web thickness

t_w, in mm, of stiffeners of the body part supporting the propulsion module is not to be less than the following value:

$$t_w = \sqrt{3} \cdot \frac{f_{shr} \cdot P \cdot s \cdot \ell}{h_w \cdot 0{,}9 \cdot R_{eH}} 10^3 + 2$$

without being less than 6,5 mm, but not need exceed 9,5 mm

where:

P, s, ℓ : Parameters defined in [1.5.1]

f_{shr} : Shear force distribution factor taken as:

- for vertical stiffener with fixed ends

  $f_{shr} = 0{,}7$

- for vertical stiffener with simple support ends

  $f_{shr} = 0{,}5$

h_w : Web height of the stiffener, in mm, measured between the attached plating and the lower side of the stiffener flange, if any

1.6 Primary supporting members

1.6.1 Analysis criteria

The scantlings of primary supporting members of the azimuth propulsion system are to be provided by the Designer based on direct calculations. The direct calculations are to be carried out according to the following requirements:

- the structural model is to include the body part of the azimuth propulsion system, the bearing and the hull supports
- the boundary conditions are to represent the connections of the azimuth propulsion system to the hull structures
- the loads to be applied are those defined in [1.6.2].

The direct calculation analyses (structural model, load and stress calculation, strength checks) carried out by the Designer are to be submitted to the Society for information.

1.6.2 Loads

The following loads are to be considered by the Designer in the direct calculation of the primary supporting members of the azimuth propulsion system:

- gravity loads
- buoyancy
- maximum loads calculated for an orientation of the system equal to the maximum angle at which the azimuth propulsion system can be oriented on each side when the ship navigates at its maximum speed
- maximum loads calculated for the possible orientations of the system greater than the maximum angle at the relevant speed (see [1.3.1])
- maximum loads calculated for the crash stop of the ship obtained through inversion of the propeller rotation
- maximum loads calculated for the crash stop of the ship obtained through a 180° rotation of the pod.

1.6.3 Strength check

The Von Mises equivalent stress σ_E, in MPa, in primary supporting members, calculated for the load cases defined in [1.6.2], is to comply with the following formula:

$\sigma_E \leq \sigma_{ALL}$

where:

σ_{ALL} : Allowable stress, in MPa, to be taken equal to 0,55 R_{eH}

R_{eH} : Minimum yield stress, in MPa, of the specified steel. R_{eH} is not to exceed the lower of 0,7 R_m and 450 MPa

R_m : Minimum ultimate tensile strength, in MPa, of the steel used.

When the loads are calculated for crash stop of the ship, the criteria given in [1.6.4] are to be complied with.

When fine mesh finite element analysis (typically 50 mm x 50 mm) is used for the calculation of stresses, then the following criteria may be applied:

- for elements not adjacent to the weld:

  $\sigma_E \leq 1{,}53\ \sigma_{ALL}$

- for elements adjacent to the weld:

  $\sigma_E \leq 1{,}34\ \sigma_{ALL}$

1.6.4 Strength check for crash stop of the ship

When the loads are calculated for crash stop of the ship, the Von Mises equivalent stress σ_E, in MPa, in primary supporting members is to comply with the following formula:

$\sigma_E \leq \sigma_{CRASH}$

where:

$\sigma_{CRASH} = 1{,}25\ \sigma_{ALL}$

σ_{ALL} : Allowable stress as defined in [1.6.3].

When fine mesh finite element analysis (typically 50 mm x 50 mm) is used for the calculation of stresses, then the following criteria may be applied:

- for elements not adjacent to the weld:

  $\sigma_E \leq 1{,}53\ \sigma_{CRASH}$

- for elements adjacent to the weld:

  $\sigma_E \leq 1{,}34\ \sigma_{CRASH}$

1.7 Hull supports of the azimuth propulsion system

1.7.1 Analysis criteria

The ship is to be provided with supports appropriate to sustain the loads that azimuth thruster applies to it.

Although the hull supports verification of the azimuth propulsion system is part of the ship design review process, the scantlings of hull supports are to be provided by the designer based on direct calculations to be carried out in accordance with the Society requirements.

Subject to the agreement of the Society, the following loads, strength check for yielding failure and buckling assessment are to be applied.

1.7.2 FEM model

The FEM model is to include the whole thruster room structure, the adjacent bulkhead and the primary structure at least up to the first deck above the thruster room. The extent of the FEM model should be sufficient to avoid the stress concentrations due to the boundary conditions.

1.7.3 Loads

The loads to be considered in the direct calculation of the hull supports of the azimuth propulsion system are specified in [1.6.2].

1.7.4 Strength check

The Von Mises equivalent stress σ_E, in MPa, in hull supports calculated for the load cases defined in [1.6.2], is to comply with the following formula:

$\sigma_E \leq \sigma_{ALL}$

where:

σ_{ALL} : Allowable stress, in MPa, equal to:

$\sigma_{ALL} = 65 / k$

When the loads are calculated for crash stop of the ship, the criteria given in [1.7.5] is to be complied with.

When fine mesh finite element analysis (typically 50 mm x 50 mm) is used for the calculation of stresses, then the following criteria may be applied:

- for elements not adjacent to the weld

  $\sigma_E \leq 1{,}53\ \sigma_{ALL}$

- for elements adjacent to the weld

  $\sigma_E \leq 1{,}34\ \sigma_{ALL}$

Values of σ_E greater than σ_{ALL} may be accepted by the Society on a case-by-case basis, depending on the localisation of those σ_E values and on the type of direct calculation analysis.

1.7.5 Strength check for crash stop of the ship

When the loads are calculated for crash stop of the ship, the Von Mises equivalent stress σ_E, in MPa, in primary supporting members is to comply with the following formula:

$\sigma_E \leq \sigma_{CRASH}$

where:

$\sigma_{CRASH} = 1{,}25\ \sigma_{ALL}$

σ_{ALL} : Allowable stress as defined in [1.7.4].

When fine mesh finite element analysis (typically 50 mm x 50 mm) is used for the calculation of stresses, then the following criteria may be applied:

- for elements not adjacent to the weld:

  $\sigma_E \leq 1{,}53\ \sigma_{CRASH}$

- for elements adjacent to the weld:

  $\sigma_E \leq 1{,}34\ \sigma_{CRASH}$

1.7.6 Buckling

The buckling check of the structural elements of the hull supports of the azimuth propulsion system, is to be carried out according to the buckling assessment criteria of NR467, Pt B, Ch 9, Sec 1, in consideration with the applied stresses in the plate panels according to [1.7.1] and [1.7.3].

Section 3 Machinery Assessment

Symbols

a_l : Longitudinal impact acceleration, in m/s^2
a_t : Transverse impact acceleration, in m/s^2
a_v : Vertical impact acceleration, in m/s^2
$c_{0,7}$: Chord length of the blade section at 0,7 R, in m
CP : Controllable pitch propeller
C_q, α_i : Parameters for ice torque excitation of shaft line
d : Propeller hub external diameter, in m, at propeller plane
d_{pin} : Diameter of shear pin, in mm
D : Propeller diameter, in m
D_{limit} : Limit value for the propeller diameter, in m
EAR : Expanded blade area ratio
F_b : Maximum backward blade force, in kN, for the ship's service life, See Tab 1
F_{ex} : Ultimate blade load resulting from blade failure through plastic bending, in kN, See Tab 1
F_f : Maximum forward blade force, in kN, for the ship's service life, See Tab 1
F_{ice} : Ice load, in kN
$(F_{ice})_{max}$: Maximum ice load for the ship's service life, in kN
FP : Fixed pitch propeller
H : Distance from the waterline to the point being considered, in m
h_0 : Depth of the propeller centreline from lower ice waterline (LIWL), in m
H_{ice} : Ice block dimension for propeller load definition, in m
I : Equivalent mass moment of inertia of all parts on engine side of component under consideration, in kg/m^2
I_t : Equivalent mass moment of inertia of the whole propulsion system, in kg/m^2
k : Shape parameter for Weibull distribution
LIWL : Lower ice waterline
m : Slope for S-N curve in log/log scale
M_{BL} : Blade bending moment, in kNm
MCR : Maximum continuous rating, in kW
n_n : Nominal rotational propeller speed at MCR, in free running open water conditions, in rps
N : Number of ice load cycles
N_{class} : Reference number of ice impacts per propeller revolution per ice class
N_{ice} : Total number of ice loads on propeller blade for the ship's service life
N_R : Reference number of loads for equivalent fatigue stress (10^8 cycles)
N_Q : Number of propeller revolution during a milling sequence
$P_{0,7}$: Propeller pitch at 0,7 R radius, in m
$P_{0,7n}$: Propeller pitch at 0,7 R radius at MCR in free running condition, in m
PCD : Pitch circle diameter, in m
$Q(\varphi)$: Torque, in kNm
Q_{Amax} : Maximum response torque amplitude as a simulation result, in kNm
Q_{emax} : Maximum engine torque, in kNm
$Q_F(\varphi)$: Ice torque excitation for frequency domain calculations, in kNm
Q_{fr} : Friction torque in pitching mechanism, reduction of spindle torque, in kNm
Q_{max} : Maximum torque on a propeller due to ice-propeller interaction, in kNm, See Tab 1
Q_{motor} : Electric motor peak torque, in kNm
Q_n : Nominal torque at MCR in free running condition, in kNm
$Q_r(t)$: Response torque along the propeller shaft line, in kNm, See Tab 1
Q_{peak} : Maximum of the response torque $Q_r(t)$, in kNm
Q_{sex} : Extreme spindle torque corresponding to the blade failure load F_{ex}, in kNm, See Tab 1
Q_{smax} : Maximum blade spindle torque for the ship's service life, in kNm, See Tab 1

Q_{vib} : Vibratory torque at considered component, taken from frequency domain open water TVC, in kNm
r : Blade section radius, in m
R : Propeller radius, in m
S : Safety factor
S_{fat} : Safety factor for fatigue
S_{ice} : Ice strength index for blade ice force
t : Maximum blade section thickness, in m
T : Hydrodynamic propeller thrust in bollard condition, in kN
T_b : Maximum backward propeller ice thrust for the ship's service life, in kN, See Tab 1
T_f : Maximum forward propeller ice thrust for the ship's service life, in kN, See Tab 1
T_{kmax} : Maximum torque capacity of flexible coupling, in kNm
T_{kmax1} : T_{kmax} at N = 5 10^4 load cycles, in kNm
T_{kmax2} : T_{kmax} at N = 1 load cycle, in kNm
T_{kv} : Vibratory torque amplitude at N = 10^6 load cycles, in kNm
T_n : Nominal propeller thrust at MCR in free running open water condition, in kN
T_r : Maximum response thrust along the shaft line, in kN, See Tab 1
TVC : Torsional vibration calculation
ΔT_{kmax} : Maximum range of T_{kmax} at N = 5 10^4 load cycles, in kNm
Z : Number of propeller blades
z_{pin} : Number of shear pins
α_i : Duration of propeller blade/ice interaction expressed in rotation angle, in deg
γ_ε : Reduction factor for fatigue; scatter and test specimen size effect
γ_v : Reduction factor for fatigue; variable amplitude loading effect
γ_m : Reduction factor for fatigue; mean stress effect
ρ : Reduction factor for fatigue correlating the maximum stress amplitude to the equivalent fatigue stress for 10^8 stress cycles
$\sigma_{0,2}$: Proof yield strength of material at 0,2% plastic strain, in MPa
σ_{exp} : Mean fatigue strength of blade material at 10^8 cycles to failure in sea water, in MPa
σ_{fat} : Equivalent fatigue ice load stress amplitude for 10^8 stress cycles, in MPa
σ_{fl} : Characteristic fatigue strength for blade material, in MPa
σ_{ref1} : Reference stress, in MPa, equal to:

$$\sigma_{ref1} = 0{,}6\sigma_{0,2} + 0{,}4\sigma_u$$

σ_{ref2} : Reference stress, in MPa, equal to:

$$\sigma_{ref2} = \mathrm{Min}(0{,}7\sigma_u, 0{,}6\sigma_{0,2} + 0{,}4\sigma_u)$$

σ_{st} : Maximum stress, in MPa, resulting from F_b or F_f
σ_u : Ultimate tensile strength for blade material, in MPa
$(\sigma_{ice})_A(N)$: Blade stress amplitude distribution, in MPa
$(\sigma_{ice})_{Amax}$: Maximum ice load stress amplitude at the considered location on the blade, in MPa
$(\sigma_{ice})_{bmax}$: Principal stress caused by the maximum backward propeller ice load, in MPa
$(\sigma_{ice})_{fmax}$: Principal stress caused by the maximum forward propeller ice load, in MPa
σ_{mean} : Mean stress, in MPa

1 General

1.1 Application

1.1.1 Polar class and icebreaker notations

This Section applies to ships having one of the additional class notations **POLAR CLASS** or service notation **icebreaker** and gives requirements for main propulsion, emergency and auxiliary systems essential for the safety of the ship and the crew.

1.1.2 Open water

The requirements of this Section are additional to those applicable for the basic open water class of the ship.

1.2 General design principle

1.2.1 Drainability

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

1.2.2 Ship operation in case of propeller damage

Ships assigned one of the additional class notations **POLAR CLASS 1** to **POLAR CLASS 5** or one of the service notations **icebreaker** are to have means provided to ensure sufficient ship operation in the case of propeller damage, including the Controllable Pitch (CP) mechanism. Sufficient ship operation means that the ship should be able to reach safe haven (safe location) where repairs can be undertaken. This may be achieved either by a temporary repair at sea, or by towing, assuming assistance is available. This would lead however to a condition of approval.

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

1.2.3 Propeller immersion

The propeller is to be fully submerged at the ships lowest ice water line (LIWL).

2 Design ice loads

2.1 General

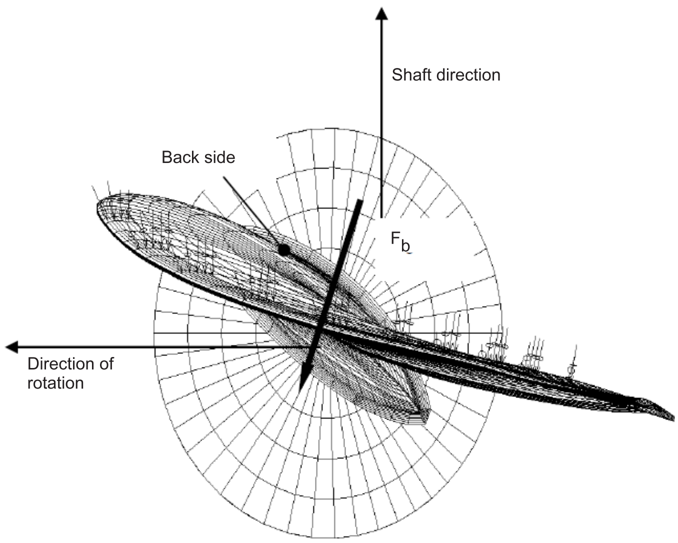
2.1.1 Ice load definition

The definitions of the different ice loads are given in Tab 1.

Table 1 : Definition of ice loads

	Definition	Use of the load in design process
F_b	The maximum lifetime backward force on a propeller blade resulting from propeller / ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to 0,7 R chord line. See Fig 1	Design force for strength calculation of the propeller blade
F_{ex}	Ultimate blade load resulting from blade loss through plastic bending. The force that is needed to cause total failure of the blade so that plastic hinge appear in the root area. The force is acting on 0,8 R. The spindle arm is 2/3 of the distance between the axis of blade and leading / trailing edge (whichever is the greater) at the 0,8 R	Blade failure load is used to dimension the blade bolts, pitch control mechanism, propeller shaft, propeller shaft bearing and trust bearing. The objective is to guarantee that total propeller blade failure does not lead to damage to other components
F_f	The maximum lifetime forward force on a propeller blade resulting from propeller / ice interaction, including hydrodynamic loads on that blade. The direction of the force is perpendicular to 0,7 R chord line	Design force for strength calculation of the propeller blade
F_{ti}	Maximum response force caused by ice block impacts on the thruster body or the propeller hub	Design load for thruster body and slewing bearings
F_{tr}	Maximum response force on the thruster body caused by ice ridge/ thruster body interaction	Design load for thruster body and slewing bearings
Q_{max}	The maximum ice-induced torque resulting from propeller / ice interaction on one propeller blade, including hydrodynamic loads on that blade	Is used for estimating of the response torque (Q_r) along the propulsion shaft line and as excitation for torsional vibration calculations
Q_r	Maximum response torque along the propeller shaft line, taking into account the dynamic behaviour of the shaft line for ice excitation (torsional vibration) and hydrodynamic mean torque on propeller	Design torque for propeller shaft line components
Q_{smax}	The maximum lifetime spindle torque on a propeller blade resulting from propeller / ice interaction, including hydrodynamic loads on that blade	When designing the propeller strength, the spindle torque is automatically taken into account because the propeller load is acting on the blade as distributed pressure on the leading edge or tip area
T_b	The maximum lifetime thrust on propeller (all blades) resulting from propeller / ice interaction. The direction of the thrust is the propeller shaft direction and the force is opposite to the hydrodynamic thrust	Is used for estimating of the response thrust T_r T_b can be used as an estimate of excitation in axial vibration calculations. However, axial vibration calculations are not required for classification purpose
T_f	The maximum lifetime thrust on propeller (all blades) resulting from propeller / ice interaction. The direction of the thrust is the propeller shaft direction acting in the direction of hydrodynamic thrust	Is used for estimating of the response thrust T_r T_f can be used as an estimate of excitation in axial vibration calculations. However, axial vibration calculations are not required for classification purpose
T_r	Maximum response thrust along shaft line, taking into account the dynamic behaviour of the shaft line for ice excitation (axial vibration) and hydrodynamic mean thrust on propeller	Design thrust for propeller shaft line components

Figure 1 : Direction of the backward blade force resultant taken perpendicular to chord line at radius 0,7 R. Ice contact pressure at leading edge is shown with small arrows



2.1.2 Ice load application

This Article covers open and ducted type propellers situated at the stern of a ship having controllable pitch or fixed pitch blades. Ice loads on bow-mounted propellers are to receive special consideration. The given loads are expected, single occurrence, maximum values for the whole ship's service life for normal operational conditions, including loads resulting from directional change of rotation where applicable. These loads do not cover off-design operational conditions, for example when a stopped propeller is dragged through ice.

This Article also covers loads due to propeller ice interaction for azimuthing and fixed thrusters with geared transmission or an integrated electric motor (“geared and podded propulsors”). However, the load models of this requirement do not include propeller/ice interaction loads when ice enters the propeller of a turned azimuthing thruster from the side (radially). Ice loads resulting from ice impacts on the body of thrusters are to be estimated on a basis of Sec 6.

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

The following different loads are to be applied separately:

- Force F_b maximum force experienced during the lifetime of the ship that bends a propeller blade backwards when the propeller mills an ice block while rotating ahead
- Force F_f maximum force experienced during the lifetime of the ship that bends a propeller blade forwards when the propeller mills an ice block while rotating ahead.

2.2 Ice class factors

2.2.1 Design ice block and ice strength index

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

Table 2 : Design Ice class factors

	POLAR CLASS 1 or **icebreaker 1**	**POLAR CLASS 2** or **icebreaker 2**	**POLAR CLASS 3** or **icebreaker 3**	**POLAR CLASS 4** or **icebreaker 4**	**POLAR CLASS 5** or **icebreaker 5**	**POLAR CLASS 6** or **icebreaker 6**	**POLAR CLASS 7** or **icebreaker 7**
H_{ice}	4,00 m	3,50 m	3,00 m	2,50 m	2,00 m	1,75 m	1,50 m
S_{ice}	1,20	1,10	1,10	1,10	1,10	1,00	1,00

2.3 Propeller ice interaction loads

2.3.1 Maximum backward blade force F_b for open propellers

The maximum backward blade force F_b, in kN, for open propellers is equal to:

- when $D < D_{limit}$:

$$F_b = 27\, S_{ice} (nD)^{0,7} \left(\frac{EAR}{Z}\right)^{0,3} D^2$$

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

$$F_b = 23\ S_{ice}\ (nD)^{0,7} \cdot \left(\frac{EAR}{Z}\right)^{0,3} (H_{ice})^{1,4}\ D$$

where:

$D_{limit} = 0,85\ (H_{ice})^{1,4}$, in m

n : Rotational propeller speed, in rps, taken as follows:
  - for CP propellers:
    $n = n_n$
  - for FP propellers:
    $n = 0,85\ n_n$

For ships with the service notation **icebreaker**, the F_b value is to be multiplied by a factor of 1,1.

2.3.2 Maximum forward blade force F_f for open propellers

The maximum forward blade force F_f, in kN, for open propellers is equal to:

- when $D < D_{limit}$:

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

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

$$F_f = 500\ \frac{1}{1 - \frac{d}{D}}\ H_{ice}\ \frac{EAR}{Z}\ D$$

where:

$$D_{limit} = \frac{2}{1 - \frac{d}{D}}\ H_{ice}$$

2.3.3 Loaded area on the blade for open propellers

Load cases 1 to 4 given in Tab 3 are to be verified for CP and FP propellers. In order to obtain blade ice loads for a reversing propeller, load case 5 is also to be checked for propellers where reversing is possible.

2.3.4 Maximum backward blade force for ducted propellers

The maximum backward blade force F_b, in kN, for ducted propellers is equal to:

- when $D < D_{limit}$:

$$F_b = 9,5\ S_{ice}\ (nD)^{0,7} \left(\frac{EAR}{Z}\right)^{0,3} D^2$$

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

$$F_b = 66\ S_{ice}\ (nD)^{0,7} \left(\frac{EAR}{Z}\right)^{0,3} (H_{ice})^{1,4} D^{0,6}$$

where:

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

n : Rotational propeller speed, in rps, taken as follows:
  - for CP propellers:
    $n = n_n$
  - for FP propellers:
    $n = 0,85\ n_n$

For ships with the service notation **icebreaker**, the F_b value is to be multiplied by a factor of 1,1.

2.3.5 Maximum forward blade force F_f for ducted propellers

The maximum forward blade force F_f, in kN, for ducted propellers is equal to:

- when $D < D_{limit}$:

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

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

$$F_f = 500 \frac{1}{1-\frac{d}{D}} H_{ice} \frac{EAR}{Z} D$$

where:

$$D_{limit} = \frac{2}{1-\frac{d}{D}} H_{ice}$$

2.3.6 Loaded area on the blade for ducted propellers

Load cases 1 and 3 given in Tab 4 are to be verified for all propellers. In order to obtain blade ice loads for a reversing propeller, the load case 5 is also to be checked for propellers where reversing is possible.

2.3.7 Maximum blade spindle torque Q_{smax} for open and ducted propeller

The spindle torque Q_{smax} around the axis of the blade fitting is to be calculated both for the maximum backward blade force F_b and forward blade force F_f , which are applied as per Tab 3 and Tab 4.

If the above method gives a value which is less than the default value given by the formula below, in kNm, the default value is to be used.

$Q_{smax\text{-}Def} = 0{,}25\ F\ c_{0,7}$

where:

$F = Max[|F_b|,|F_f|]$

Table 3 : Loaded areas and load case definition for open propeller

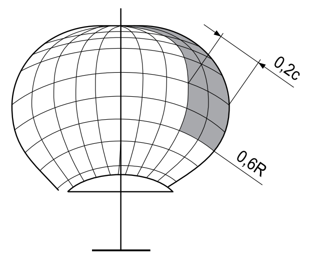
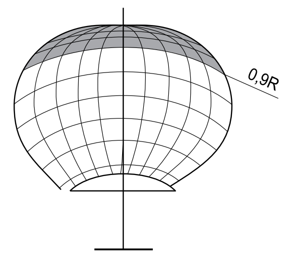
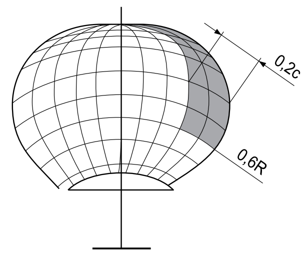
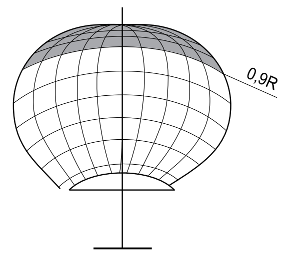
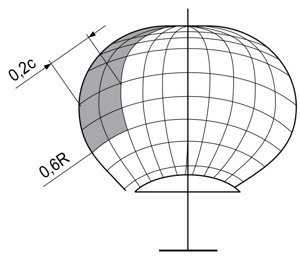
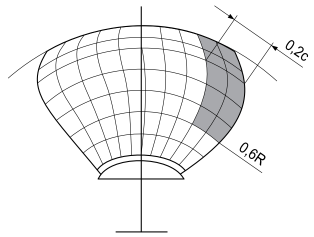
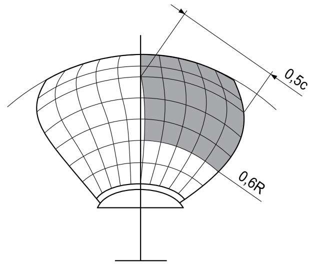
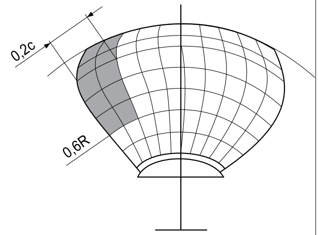
Load case	Force /Loaded area / Right handed propeller blade seen from behind	
1	F_b • Uniform pressure applied on the back of the blade (suction side) to an area from 0,6 R to the tip and from the leading edge to 0,2 times the chord length	
2	$0{,}5\ F_b$ • Uniform pressure applied on the back of the blade (suction side) on the propeller tip area outside of 0,9 R radius	
3	F_f • Uniform pressure applied on the blade face (pressure side) to an area from 0,6 R to the tip and from the leading edge to 0,2 times the chord length	
4	$0{,}5\ F_f$ • Uniform pressure applied on propeller face (pressure side) on the propeller tip area outside 0,9 R radius	
5	$0{,}6\ Max\ (F_b,\ F_f)$ • Uniform pressure applied on propeller face (pressure side) to an area from 0,6 R to the tip and from the trailing edge to 0,2 times the chord length	

Table 4 : Loaded areas and load case definition for ducted propeller

Load case	Force /Loaded area / Right handed propeller blade seen from behind	
1	F_b • Uniform pressure applied on the back of the blade (suction side) to an area from 0,6 R to the tip and from the leading edge to 0,2 times the chord length	
3	F_f • Uniform pressure applied on the blade face (pressure side) to an area from 0,6 R to the tip and from the leading edge to 0,5 times the chord length	

Load case	Force /Loaded area / Right handed propeller blade seen from behind	
5	0,6 Max (F_b, F_f) • Uniform pressure applied on propeller face (pressure side) to an area from 0,6 R to the tip and from the trailing edge to 0,2 times the chord length	

2.3.8 Load distributions (spectra) for blade loads

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

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

where:

k : Shape parameter of the spectrum

N_{ice} : Number of load cycles in the spectrum, as defined in [2.3.9]

F_{ice} : Random variable for ice loads on the blade, such as

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

The resulting blade stress amplitude distribution is given by the following formula:

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

where:

$(\sigma_{ice})_{Amax}$: Maximum ice load stress amplitude at the considered location on the blade, in MPa, as given by:

$$(\sigma_{ice})_{Amax} = \frac{(\sigma_{ice})_{fmax} - (\sigma_{ice})_{bmax}}{2}$$

k : Shape parameter for the ice force distribution to be taken as:

- for open propeller:
  k = 0,75
- for ducted propeller:
  k = 1,0

2.3.9 Number of ice loads

The number of load cycles, N_{ice}, used in the load spectrum per blade is given by the following formula:

$$N_{ice} = k_1 k_2 k_3 N_{class} n$$

where:

k_1 : Coefficient defined as:

- for centre propeller:
  $k_1 = 1$

- for wing propeller:
  $k_1 = 2$
- for pulling propeller, wing and centre:
  $k_1 = 3$

k_2 : Coefficient defined as:

- for $f < 0$:
  $k_2 = 0{,}8 - f$
- for $0 \le f \le 1$:
  $k_2 = 0{,}8 - 0{,}4f$
- for $1 < f \le 2{,}5$:
  $k_2 = 0{,}6 - 0{,}2f$
- for $f > 2{,}5$:
  $k_2 = 0{,}1$

f : Coefficient taken equal to:

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

When h_0 is unknown, $h_0 = D/2$

k_3 : Propulsion machinery type factor, to be taken equal to:

- for fixed propulsors:
  $k_3 = 1{,}0$
- for azimuthing propulsors:
  $k_3 = 1{,}2$

N_{class} : Reference number of impacts per propeller revolution for each ice class taken according to Tab 5

For ships with the service notation **icebreaker**, the N_{ice} value is to be multiplied by a factor of 3.

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

Figure 2 : Weibull-type distribution for fatigue design

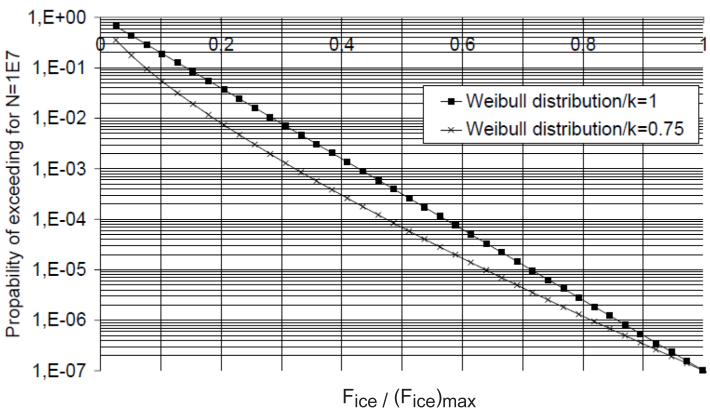


Table 5 : Reference number of impacts

	POLAR CLASS 1 or icebreaker 1	POLAR CLASS 2 or icebreaker 2	POLAR CLASS 3 or icebreaker 3	POLAR CLASS 4 or icebreaker 4	POLAR CLASS 5 or icebreaker 5	POLAR CLASS 6 or icebreaker 6	POLAR CLASS 7 or icebreaker 7
N_{class}	$21\ 10^6$	$17\ 10^6$	$15\ 10^6$	$13\ 10^6$	$11\ 10^6$	$9\ 10^6$	$6\ 10^6$

2.4 Blade failure load for both open and ducted propellers

2.4.1 Bending Force F_{ex}

The minimum load required resulting in blade failure through plastic bending is to be calculated iteratively along the radius of the blade from blade root to 0,5 R using the following formula with the ultimate load assumed to be acting at 0,8 R in the weakest direction.

$$F_{ex} = \frac{0{,}3\,c\,t^2}{0{,}8D - 2r}\sigma_{ref1}10^3$$

where:

F_{ex} : Blade failure load, in kN

c : Chord length of the considered blade section, in m

c, t, r : Values as defined in Symbols taken at the cylindrical root section of the blade, i.e. at the weakest section outside the root fillet located typically at the termination of the fillet into the blade profile

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

2.4.2 Spindle torque Q_{sex}

The force that causes blade failure typically reduces when moving from the propeller centre towards the leading and trailing edges. The maximum spindle torque occurs at a certain distance from the blade centre of rotation.

The maximum spindle torque, Q_{sex}, due to a blade failure load acting at 0,8 R is defined either by an appropriate stress analysis or with the following formula:

$$Q_{sex} = \mathrm{Max}(c_{LE0,8}\ ;\ 0,8 \cdot c_{TE0,8}) \cdot C_{spex} \cdot F_{ex}$$

where:

$c_{LE0,8}$: Leading edge portion of the chord length at 0,8 R

$c_{TE0,8}$: Trailing edge portion of the chord length at 0,8 R

C_{spex} : Coefficient defined by the following formula without being less than 0,3:

$$C_{spex} = C_{sp} \cdot C_{fex}$$

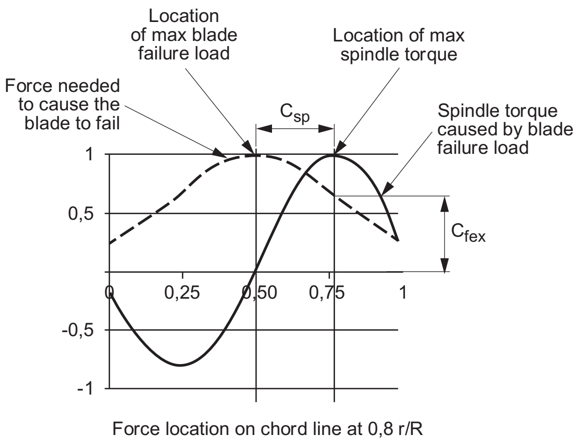
C_{sp} : Non-dimensional parameter taking into account the spindle arm equal to 0,7

C_{fex} : Non-dimensional parameter taking into account the reduction of blade failure force at the location of maximum spindle torque, taken as:

$$C_{fex} = 1 - \left(4 \cdot \frac{EAR}{Z}\right)^3$$

The Fig 3 illustrates the spindle torque values due to blade failure loads across the whole chord length.

Figure 3 : Force location on chord line at 0,8 r/R.Schematic figure showing the blade failure load and the related spindle torque when the force acts at different location on the chord line at radius 0,8



2.5 Axial design loads acting on open and ducted propellers

2.5.1 Maximum ice thrust acting on open and ducted propellers

The maximum forward and backward ice thrusts, T_f and T_b, in kN, acting on open or ducted propellers are given by the following formula:

$$T_f = 1,1 \cdot F_f$$

$$T_b = 1,1 \cdot F_b$$

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

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

The design thrust along the propeller shaft line, T_r, in kN, is to be calculated with the following formulae. The greater value of the forward and backward directional loads is to be taken as the design load for both directions. The factors 2,2 and 1,5 take into account the dynamic magnification resulting from axial vibration:

- In forward direction:

  $T_r = T + 2{,}2 \cdot T_f$

- In backward direction:

  $T_r = 1{,}5 \cdot T_b$

where

T : Hydrodynamic bollard thrust, in kN

When the hydrodynamic bollard thrust is unknown, T is to be taken from the Tab 6.

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

Table 6 : Guidance for hydrodynamic bollard thrust values

Propeller type		T
CP propellers	Open	$1{,}25\ T_n$
	Ducted	$1{,}1\ T_n$
FP propellers driven by turbine or electrical motor		T_n
FP propellers driven by diesel engine	Open	$0{,}85\ T_n$
	Ducted	$0{,}75\ T_n$

2.6 Torsional design loads acting on open and ducted propellers

2.6.1 Design ice torque on propellers

The design ice torque, Q_{max}, in kNm, acting on propellers is equal to:

- when $D < D_{limit}$:

$$Q_{max} = k_{prop} \cdot \left(1 - \frac{d}{D}\right) \cdot \left(\frac{P_{0,7}}{D}\right)^{0,16} \cdot (nD)^{0,17} \cdot D^3$$

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

$$Q_{max} = 1{,}9 \cdot k_{prop} \cdot \left(1 - \frac{d}{D}\right) \cdot H_{ice}^{1,1} \cdot \left(\frac{P_{0,7}}{D}\right)^{0,16} \cdot (nD)^{0,17} \cdot D^{1,9}$$

where:

$D_{limit} = 1{,}8\ H_{ice}$

k_{prop} : Coefficient depending on the propeller type and taken as:

- for open propeller:

  $k_{prop} = k_{open}$

- for ducted propeller:

  $k_{prop} = k_{ducted}$

k_{open} : Coefficient taken as:

- for **POLAR CLASS 1** to **POLAR CLASS 5** and **icebreaker 1** to **icebreaker 5**:

  $k_{open} = 14{,}7$

- for **POLAR CLASS 6**, **POLAR CLASS 7**, **icebreaker 6** and **icebreaker 7**:

  $k_{open} = 10{,}9$

k_{ducted} : Coefficient taken as:

- for **POLAR CLASS 1** to **POLAR CLASS 5** and **icebreaker 1** to **icebreaker 5**:

  $k_{ducted} = 10{,}4$

- for **POLAR CLASS 6**, **POLAR CLASS 7**, **icebreaker 6** and **icebreaker 7**:

  $k_{ducted} = 7{,}7$

n : Rotational propeller speed in bollard condition, in rps

If unknown, n is to be taken from Tab 7.

For CP propellers, the propeller pitch $P_{0,7}$ is to correspond to MCR in bollard condition. If not known, $P_{0,7}$ is to be taken as $0{,}7\ P_{0,7n}$ where $P_{0,7n}$ is the propeller pitch at MCR in free running condition.

Table 7 : Guidance for rotational propeller speed n

Propeller type	n
CP propellers	n_n
FP propellers driven by turbine or electrical motor	n_n
FP propellers driven by diesel engine	0,85 n_n

2.6.2 Ice torque excitation for open and ducted propellers

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

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

2.6.3 Excitation for the time domain calculation

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

- when φ rotates from 0 to α_i plus integer revolutions

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

- when φ rotates from α_i to 360 plus integer revolutions

$$Q(\varphi) = 0$$

where:

φ : Rotation angle, in deg, starting when the first ice impact occurs

C_q : Parameter given in Tab 8

α_ι : Duration of propeller blade/ice interaction expressed in propeller rotation angle, given in Tab 8, in deg.

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

At the beginning and at the end of the milling sequence (within calculated duration), linear ramp functions are to be used to increase

C_q to its maximum within one propeller revolution and vice versa to decrease it to zero.

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

$$N_Q = 2 \cdot H_{ice}$$

with:

$Z\ N_Q$: Number of impacts for blade order excitation

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

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

Table 8 : Ice impact magnification and duration factors for different blade numbers

Torque excitation	Propeller/ice interaction	C_q	α_i, in 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	1,00	135	135	135	135
Case 3	Two ice blocks (phase shift 360/(2·Z) deg.)	0,50	45	45	36	30
Case 4	Single ice block	0,50	45	45	36	30

2.6.4 Frequency domain excitation

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

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

where:

α_1, α_1 : Phase angles of excitation component given in Tab 9

φ : Rotation angle, in deg

C_{q0} : Mean torque component given in Tab 9

C_{q1} : First blade order excitation amplitude given in Tab 9

C_{q2} : Second blade order excitation amplitude given in Tab 9

E_0 : Number of ice blocks in contact

Z : Number of blades

Torsional vibration responses are to be calculated for all excitation cases.

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

- The highest response torque (between the various lumped masses in the system) is in the following referred to as peak torque Q_{peak}.
- 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},that can be determined, in kNm, by the following formula:

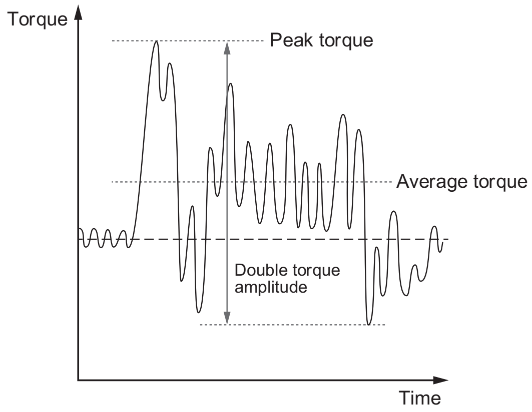
  $$Q_{Amax} = \frac{Max[Q_r(time)] - Min[Q_r(time)]}{2}$$

  An illustration of Q_{Amax} is given in Fig 4

Table 9 : Coefficients for simplified excitation torque estimation

Z	Excitation Case	C_{q0}	C_{q1}	α_1	C_{q2}	α_2	E_0
3	1	0,3750	0,375	−90	0,0000	0	1
	2	0,7000	0,330	−90	0,0500	−45	1
	3	0,2500	0,250	−90	0,0000	0	2
	4	0,2000	0,250	0	0,0500	−90	1
4	1	0,4500	0,360	−90	0,0600	−90	1
	2	0,9375	0,000	−90	0,0625	−90	1
	3	0,2500	0,251	−90	0,0000	0	2
	4	0,2000	0,250	0	0,0500	−90	1
5	1	0,4500	0,360	−90	0,0600	−90	1
	2	1,1900	0,170	−90	0,0200	−90	1
	3	0,3000	0,250	−90	0,0480	−90	2
	4	0,2000	0,250	0	0,0500	−90	1
6	1	0,4500	0,375	−90	0,0500	−90	1
	2	1,4350	0,100	−90	0,0000	0	1
	3	0,3000	0,250	−90	0,0480	−90	2
	4	0,2000	0,250	0	0,0500	−90	1

Figure 4 : Example of different torques in a measured curve



2.6.5 Design torque along shaft line

a) If there is no relevant first order propeller torsional resonance in the range 20% of n_n above and 20% below the maximum operating speed in bollard condition (see Tab 7), the following estimation of the maximum response torque Q_r, in kNm, can be used to calculate the design torque along the propeller shaft line:

- for directly coupled two stroke diesel engines without flexible coupling:

$$Q_r = Q_{emax} + Q_{vib} + Q_{max} \cdot \frac{I}{I_t}$$

- for all other plants:

$$Q_r = Q_{emax} + Q_{max} \cdot \frac{I}{I_t}$$

where:

I : Equivalent mass moment of inertia of all parts on engine side of component under consideration

I_t : Equivalent mass moment of inertia of the whole propulsion system

Q_{emax} : Maximum torque, in kNm. If Q_{emax}, is not known, it is to be taken as follows:

- for propellers driven by electric motor:
  $Q_{emax} = Q_{motor}$
- for CP propellers not driven by electric motor:
  $Q_{emax} = Q_n$
- for FP propellers driven by turbine:
  $Q_{emax} = Q_n$
- for FP propellers driven by diesel engine:
  $Q_{emax} = 0,75\ Q_n$

Q_{motor} : Electric motor peak torque, in kNm

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

b) If there is a first blade order torsional resonance in the range 20% of n_n above and 20% below the maximum operating speed (bollard condition), the design torque Q_r of the shaft component 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.

2.7 Design Ice Force at Bow

2.7.1 Bow forms

a) Bow with icebreaking form:

Design ice loads calculated according to [2.7.6] and [2.7.7] are applicable for bow form where:

- the buttock angle at the stem γ_{stem} is positive and less than 80 deg, and
- the normal frame angle θ at the centre of the foremost sub-region is greater than 10 deg.

(see Fig 5)

b) Bow with non-icebreaking form

Design ice loads calculated according to [2.7.6] and [2.7.7] are applicable to ships having the additional class notation **POLAR CLASS 6** or **POLAR CLASS 7**, or, the service notation **icebreaker 6** or **icebreaker 7**, and a bow with vertical sides or a bulbous bow. This includes bows where the normal frame angles θ at the considered sub-regions are between 0 and 10 deg. (see Fig 5)

For ships assigned the additional class notation **POLAR CLASS 6** or **POLAR CLASS 7** or, the service notation **icebreaker 6** or **icebreaker 7**, and equipped with bulbous bows, the design ice forces on the bow determined according to [2.7.6] and [2.7.7] are to be taken as the maximum between:

- ice loads calculated for non-icebreaking form,
- ice loads calculated for icebreaking form and assuming the shape coefficient $c_i = 0{,}6$

c) Other bow forms

For ships with bow forms other than those defined in a) and b), the design ice loads are to be specially considered by the Society.

2.7.2 Glancing impact load characteristics - Class factors

The parameters defining the glancing impact load characteristics are reflected in the class factors listed in Tab 10.

Table 10 : Glancing impact load characteristics - Class factors for icebreaking and non-icebreaking forms

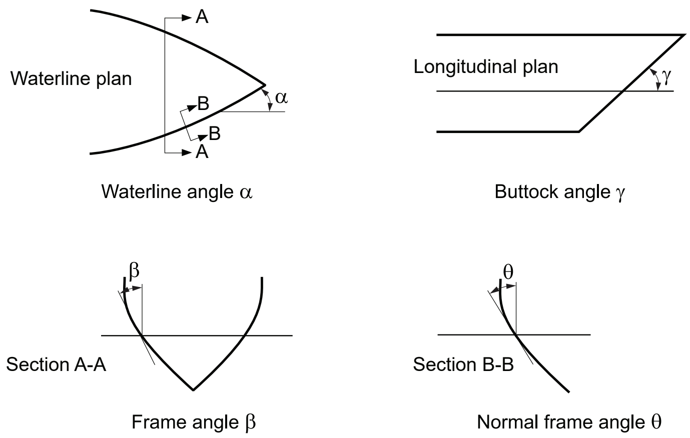
	C_C (crushing failure)	C_{CV}	C_F (flexural failure)		C_L (longitudinal strength)
			Brackish water **(1)**	Open sea	
POLAR CLASS 1	17,69	–	76,92	68,60	7,46
POLAR CLASS 2	9,890	–	54,45	46,80	5,46
POLAR CLASS 3	6,060	–	25,64	21,17	4,17
POLAR CLASS 4	4,500	–	17,05	13,48	3,15
POLAR CLASS 5	3,100	–	11,94	9,00	2,50
POLAR CLASS 6	2,400	3,43	8,70	5,49	2,37
POLAR CLASS 7	1,800	2,60	6,69	4,06	1,81

(1) Brackish water is applicable for ships sailing in water with salinity between sea water and fresh water salinity (typically less than 31ppt).

2.7.3 Calculation method

In the bow area, the design ice force F_{NB} and F_{VB} defined in [2.7.6] and [2.7.7], associated with the glancing impact load scenario are function of the hull angles measured at the upper ice waterline (UIWL). The influence of the hull angles is captured through calculation of a bow shape coefficient c_i . The hull angles are defined in Fig 5.

Figure 5 : Definition of hull angles



2.7.4 Sub-regions in bow area

The waterline length of the bow region is to be divided into four sub-regions "i" of equal length. Forces F_i and bow shape coefficients c_i are to be calculated with respect to the mid-length position x_i of each sub-region "i".

2.7.5 Shape coefficient

The shape coefficient c_i , in each sub-region i of the bow area, is to be obtained from the following formulae:

- for icebreaking form defined in [2.7.1], item a):
  - when $\theta_i > 0$:

    $c_i = \text{Min}\ (c_{i,1}\ ;\ c_{i,2}\ ;\ c_{i,3})$
  - when $\theta_i = 0$:

    $c_i = 0.60$

- for non-icebreaking form defined in [2.7.1], item b):

  $c_i = \alpha_i / 30$

where:

$$c_{i,1} = \left[0{,}097 - 0{,}68\left(0{,}85 - \frac{x_i}{L_{ui}}\right)^2\right]\frac{\alpha_i}{\sqrt{\theta_i}}$$

$$c_{i,2} = \frac{99{,}81\,C_F}{C_C\,\Delta_{ui}^{0{,}64}\,\sin\theta_i}$$

$c_{i,3} = 0{,}60$

Δ_{ui} : Displacement at the upper ice waterline (UIWL), in t, to be taken not less than 5 000 t

θ_i : Normal frame angle, in degree, in sub-region i of the bow area.

2.7.6 Design ice force normal to shell plating at the bow

The force F_{NB}, in kN, is to be obtained from the following formula:

$F_{NB} = \text{Max}\ (F_i)$

where:

F_i : Force in sub-region i of the bow area, in kN, taken equal to:

- for icebreaking form (see [2.7.1] item a)):

  $$F_i = 12.02 \cdot c_i \cdot C_C \cdot \Delta_{ui}^{0{,}64}$$

- for non-icebreaking form (see [2.7.1], item b)):

  $$F_i = 38.90 \cdot c_i \cdot C_{CV} \cdot \Delta_{ui}^{0{,}47}$$

Δ_{ui} : Displacement in t, as defined in [2.7.5]

2.7.7 Vertical design ice force at the bow

The vertical design ice force at the bow F_{VB}, in kN, is to be obtained from the following formula:

$F_{VB} = \text{Min}\ (F_{IB1}\ ;\ F_{IB2})$

where:

$$F_{IB1} = 1.505\ K_f^{0{,}15} \cdot K_h^{0{,}35}\ (\sin\gamma_{stem})^{0{,}2}\,\Delta_{ui}^{0{,}5}\,C_L$$

$$F_{IB2} = 1200 \cdot C_F$$

with:

K_h : Coefficient, in kN/m, taken equal to:

$K_h = 10\ A_{wp}$

K_f : Coefficient equal to:

- for a simple wedge bow form ($c_{eb} = 1$):

  $$K_f = \frac{\left[\dfrac{B_{ui}}{2\,L_B}\right]^{0{,}9}}{(\tan\gamma_{stem})^{1{,}8}}$$

- for a spoon bow form ($0 < c_{eb} < 1$):

  $$K_f = \frac{\left[\dfrac{B_{ui}}{L_B^{c_{eb}}(1 + c_{eb})}\right]^{0{,}9}}{(\tan\gamma_{stem})^{0{,}9\,(1 + c_{eb})}}$$

- for a landing craft bow form ($c_{eb} = 0$):

  $$K_f = \left[\frac{B_{ui}}{\tan\gamma_{stem}}\right]^{0{,}9}$$

Δ_{ui} : Displacement at the upper ice waterline (UIWL), in t, to be taken not less than 10 000 t

A_{wp} : Ship waterplane area, in m², at the upper ice waterline (UIWL)

γ_{stem} : Buttock angle at the stem, in degree, to be measured between the horizontal axis and the stem tangent at the upper ice waterline (UIWL)

c_{eb} : Bow shape exponent that describes the waterplane at the upper ice waterline (UIWL), see Fig 6 and Fig 7

L_B : Bow length, in m, at the upper ice waterline (UIWL) as defined in Fig 7.

When the general arrangement plan of the ship is available, the way to find c_{eb} and L_B is to select two points on the bow. If the coordinates of the two points are (x_1, y_1) and (x_2, y_2), the shape parameters c_{eb} and L_B are to be obtained from the following formulae:

$$c_{eb} = \frac{\ln\left(\frac{y_2}{y_1}\right)}{\ln\left(\frac{L_{ui} - x_2}{L_{ui} - x_1}\right)}$$

$$L_B = (L_{ui} - x_2)\left(\frac{2y_2}{B_{ui}}\right)^{-1/c_{eb}}$$

Figure 6 : Examples of c_{eb} for B_{ui} = 20 m and L_B = 16 m

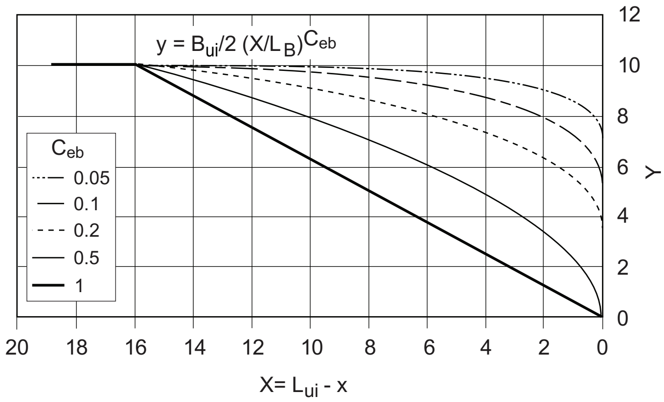
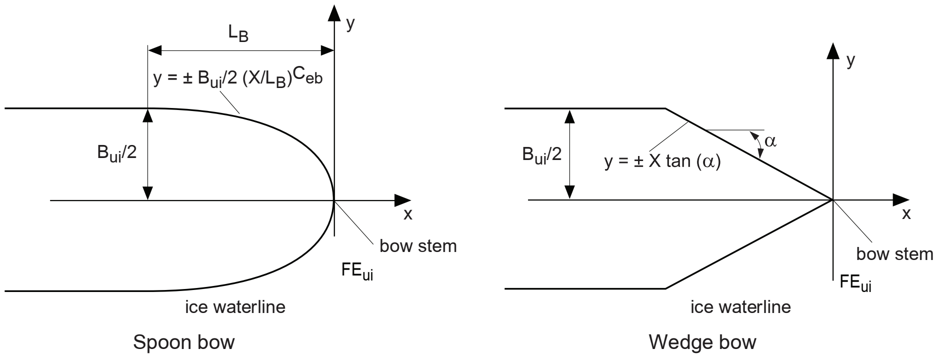


Figure 7 : Bow shape definition



3 Design of machinery components

3.1 Design principle

3.1.1 Pyramid strength principle

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

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

3.2 General fatigue criteria

3.2.1 General fatigue requirement

The design loads are to be based on the ice excitation and where necessary (shafting) dynamic analysis, described as a sequence of blade impacts (See [2.6.3]). The shaft response torque is to be determined according to [2.6.5].

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

$$D = \sum_{j=1}^{k} \frac{n_j}{N_j} \leq 1$$

where:

D : Miners damage sum

k : Number of stress levels

N_j : Number of load cycles to failure of the individual stress level class, j from 1 to k

n_j : Accumulated number of load cycles of the case under consideration, per class, j from 1 to k

Note 1: The stress distribution is to be divided into a frequency load spectrum having minimum 10 stress blocks (every 10% of the load). The maximum allowable load is limited to σ_{ref2} for propeller blades and yield strength for all other components. The load distribution (spectrum) is to be in accordance with the Weibull distribution.

3.3 Propeller blades

3.3.1 Calculation of blade stresses due to static loads

The blade stresses, i.e. equivalent and principal stresses, are to be calculated for the design loads given in [2.3]. Finite element analysis (FEA) is to be used for stress analysis as part of the final approval for all propeller blades.

The von Mises stresses, σ_{vm}, in MPa, are to comply with acceptability criterion given in [3.3.2].

Alternatively the following simplified equation can be used in estimating the blade stresses for all propellers in the root area (r/R < 0,5) for final approval:

$$\sigma_{vm} = C_1 \cdot \frac{M_{BL}}{100ct^2}$$

where:

C_1 : Ratio between the actual stress and the one obtained from beam theory
If C_1 is not available, $C_1 = 1,6$

M_{BL} : Bending moment applied on the blade taken equal to:

$$M_{BL} = \left(0,75 - \frac{r}{R}\right) \cdot R \cdot F \quad \text{for relative radius } r/R < 0,5$$

$$F = Max[F_b\ ;\ F_t]$$

3.3.2 Acceptability criterion for static loads

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

$$\sigma_{st} \le \frac{\sigma_{ref2}}{1,3}$$

where:

σ_{st} : Calculated stress for the design loads, in MPa

If finite element analysis is used in estimating the stresses, von Mises stresses are to be used.

3.3.3 Fatigue design of propeller blade

For materials with a two slope S-N curve (See Fig 8), the fatigue calculations defined in this requirement are not required if the following criterion is fulfilled:

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

where:

B_1, B_2, B_3 : Coefficients for open and ducted propellers:

- for open propeller:
  - $B_1 = 0,00328$
  - $B_2 = 1,0076$
  - $B_3 = 2,101$
- for ducted propeller:
  - $B_1 = 0,00223$
  - $B_2 = 1,0071$
  - $B_3 = 2,471$

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

- the fatigue design of the propeller blade is based on an estimated load distribution for the service life of the ship and the S-N curve for the blade material
- an equivalent stress σ_{fat} that produces the same fatigue damage as the expected load distribution is to be calculated according to Miner's rule
- the acceptability criterion for fatigue is to be fulfilled as given in this requirement
- the equivalent stress is normalised for 10^8 cycles.

The blade stresses at various selected load levels for fatigue analysis are to be taken proportional to the stresses calculated for maximum loads given in [2.3].

The peak principal stresses σ_f and σ_b are determined from F_f and F_b using FEA.

The peak stress range $\Delta\sigma_{max}$, in MPa, is given by the following formula:

$$\Delta\sigma_{max} = 2\sigma_{Amax} = |(\sigma_{ice})_{fmax}| + |(\sigma_{ice})_{bmax}|$$

where:

$\Delta\sigma_{max}$: Peak stress range, in MPa

σ_{Amax} : Maximum stress amplitude, in MPa, determined on the basis of load cases 1 and 3, 2 and 4.

For the calculation of equivalent stress, two types of S-N curves are available:

- Two slope S-N curve (slopes 4,5 and 10), see Fig 8
- One slope S-N curve (the slope can be chosen), see Fig 8

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

Figure 8 : Two-slope S-N curve

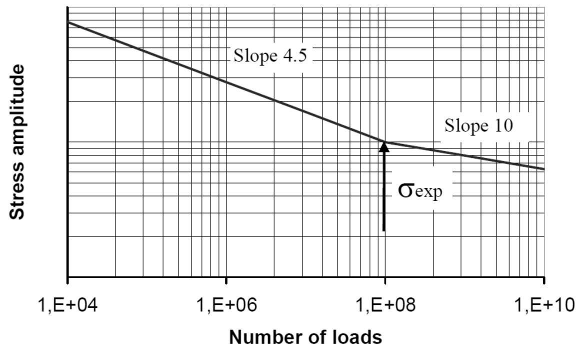
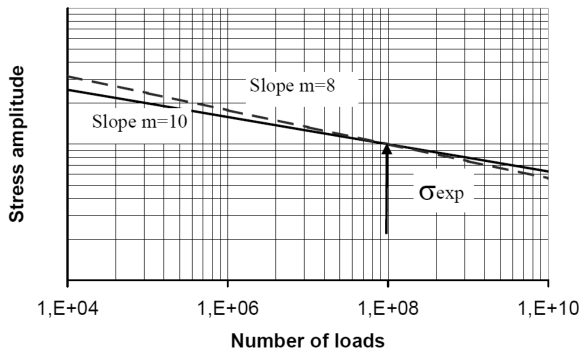


Figure 9 : Constant-slope S-N curve



3.3.4 Equivalent fatigue stress

The equivalent fatigue stress, in MPa, for 10^8 cycles which produces the same fatigue damage as the load distribution is given by the following formula:

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

where:

$(\sigma_{ice})_{max}$: Mean value of the principal stress amplitudes resulting from design forward and backward blade forces at the location being studied, in MPa, taken equal to:

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

In the calculation of $(\sigma_{ice})_{max}$, case 1 and case 3 or case 2 and case 4 are considered as pairs for $(\sigma_{ice})_{fmax}$ and $(\sigma_{ice})_{bmax}$ calculations. Case 5 is excluded from the fatigue analysis.

$(\sigma_{ice})_{fmax}$: Principal stress resulting from forward load, in MPa

$(\sigma_{ice})_{bmax}$: Principal stress resulting from backward load, in MPa

ρ : Parameter related to S-N curve, taken as follow:

- for two-slope S-N curve:

  The error of the following method to determine ρ is sufficiently small, if the number of load cycles N_{ice} is within the range $5 \cdot 106 \leq N_{ice} \leq 10^8$

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

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

  where:

σ_{fl} : Blade material fatigue strength at 10^8 load cycles, in MPa, given as:

$$\sigma_{fl} = \gamma_{\varepsilon 1} \cdot \gamma_{\varepsilon 1} \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp}$$

(See [3.3.5] for the parameters used in the formula)

C_1, C_2, C_3, C_4: Coefficients given in Tab 11

- for constant-slope S-N curve:

  For materials with a constant-slope S-N curve, (see Fig 9), ρ is to be calculated from the following formula:

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

where:

k : Shape parameter of the Weibull distribution:
- for ducted propellers: $k = 1,0$
- for open propellers: $k = 0,75$

N_{ice} : Number of load cycles to be taken between $5 \cdot 10^6$ and 10^8

N_R : Reference number of load cycles: $N_R = 10^8$

G : Parameter defined in Tab 12. Linear interpolation may be used to calculate the value of G for intermediate values of m/k ratios

Note 1: A more general method of determining the equivalent fatigue stress of propeller blades is described in [3.6], where the principal stresses are considered according to [2.3] using the Miner's rule. For a total number of load blocks $n_{bl} > 100$, both methods deliver the same result. Therefore, they are regarded as equivalent.

Table 11 : Coefficients C1, C2, C3 and C_4

	C_1	C_2	C_3	C_4
Open propeller	0,000747	0,0645	– 0,0565	2,220
Ducted propeller	0,000534	0,0533	– 0,0459	2,584

Table 12 : G parameter as a function of m/k

m/k	3	3,5	4	4,5	5	5,5	6	6,5
G	6	11,6	24	52,3	120	287,9	720	1 871

m/k	7	7,5	8	8,5	9	9,5	10
G	5 040	14 034	40 320	119 292	362 880	$1,133\ 10^6$	$3,623\ 10^6$

3.3.5 Acceptability criterion for fatigue

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

$$\sigma_{fat} \le \frac{\sigma_{fl}}{1,5}$$

where:

σ_{fat} : Fatigue strength, in MPa, corresponding to the fatigue limit at 10^8 load cycles

σ_{fl} : Blade material fatigue strength at 10^8 load cycles, in MPa, given as:

$$\sigma_{fl} = \gamma_{\varepsilon 1} \cdot \gamma_{\varepsilon 1} \cdot \gamma_v \cdot \gamma_m \cdot \sigma_{exp}$$

$\gamma_{\varepsilon 1}$: Reduction factor due to scatter, equal to one standard deviation

If the actual value is not known, $\gamma_{\varepsilon 1} = 0,85$

$\gamma_{\varepsilon 2}$: Geometrical size factor for test specimen size effect given as:

$$\gamma_{\varepsilon 2} = 1 - a \cdot \ln\left(\frac{t}{0,025}\right)$$

with:

$a = 0,10$ for bronze and brass

$a = 0,05$ for stainless steel

t : Maximum blade thickness at the considered point, in m

γ_v : Reduction factor for variable amplitude loading

If the actual value is not known, $\gamma_v = 0,75$

γ_m : Reduction factor for mean stress, given as:

$$\gamma_m = 1 - \left(1,4 \cdot \frac{\sigma_{mean}}{\sigma_u}\right)^{0,75}$$

If the actual value is not known, $\gamma_m = 0,75$

σ_{exp} : Mean fatigue strength of the blade material at 10^8 cycles to failure in seawater

σ_{exp} provided in Tab 13 has been defined from the results of constant amplitude loading fatigue tests at 10^7 load cycles and 50% survival probability and has been extended to 10^8 load cycles.

Fatigue strength values and correction factors other than those given in Tab 13 may be used, provided the values are determined under conditions approved by the Society.

The characteristics of the S-N curve are based on two slopes, the first slope is equal to 4,5 from 1 000 to 10^8 load cycles; the second slope is equal to 10 over 10^8 load cycles.

The maximum allowable stress for one or low number of cycles is limited to σ_{ref2}/S, with S = 1,3 for static loads.

Table 13 : Mean fatigue strength σ_{exp} for different material types

Mean fatigue stress for 10^8 load cycles and stress ration R = -1 with a survival probability of 50%

Bronze and brass	σ_{exp} , in MPa	Stainless steel	σ_{exp} , in MPa
Mn-Bronze, CU1 (high tensile brass)	84	Ferritic (12Cr 1Ni)	144 **(1)**
Mn-Ni-Bronze, CU2 (high tensile brass)	84	Martensitic (13Cr 4Ni/13Cr 6Ni)	156
Ni-Al-Bronze, CU3	120	Martensitic (16Cr 5Ni)	168
Mn-Al-Bronze, CU4	113	Austenitic (19Cr 10Ni)	132
(1) This value may be used provided a perfect galvanic protection is active. Otherwise a reduction of about 30 MPa is to be applied			

3.4 Blade bolts and pitch control mechanism

3.4.1 General

The blade bolts, pitch control mechanism, and the fitting of the propeller to the propeller shaft are to be designed to withstand the maximum static and fatigue design loads (as applicable), as defined in [2.3] and [3.3].

The safety factor S against yielding due to static loads and against fatigue is to be greater than 1,5, if not stated otherwise. The safety factor S for loads, resulting from propeller blade failure as defined in [2.4] is to be greater than 1,0 against yielding.

Provided that calculated stresses duly considering local stress concentrations are less than yield strength, or maximum of 0,7 σ_u of respective materials, detailed fatigue analysis is not required. In all other cases components are to be analysed for cumulative fatigue. An approach similar to that used for shafting assessment may be applied (See [3.6]).

3.4.2 Blade bolts design

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

$$M_{bolt} = S \cdot F_{ex}\left(0,8 \cdot \frac{D}{2} - r_{bolt}\right)$$

where:

r_{bolt} : Radius from the shaft centreline to the blade bolt plane, in m

S : Safety factor taken as:

S = 1

F_{ex} : Bending force, in kN, given in [2.4.1]

Blade bolt pre-tension is to be sufficient to avoid separation between mating surfaces when the maximum forward and backward ice loads defined in [2.3] (open and ducted propellers respectively) are applied. For conventional arrangements, the blade bolts effective diameter d_{bb}, in mm, may be obtained by the following formula:

$$d_{bb} = 41 \sqrt{\frac{F_{ex} \cdot (0,8D - d) \cdot S \cdot \alpha}{\sigma_{0,2} \cdot z_{bb} \cdot PCD}}$$

where:

d_{bb} : Effective diameter of blade bolt in way of thread, in mm

S : Safety factor taken as:

S = 1

z_{bb} : Number of blade bolt

α : Coefficient taken equal to:

- for torque guided tightening:
  $\alpha = 1,6$
- for elongation guided:
  $\alpha = 1,3$
- for angle guided:
  $\alpha = 1,2$
- for elongated by other additional means:
  $\alpha = 1,1$

Other value of α may be used, provided evidence is demonstrated to the Society.

3.4.3 Pitch control mechanism

Separate means, e.g. dowel pins, are to be provided in order to withstand the spindle torque resulting from blade failure Q_{sex} (See [2.4.2]) or ice interaction Q_{smax} (See [2.3.7]), whichever is greater. Other components of the pitch control mechanism are not to be damaged by the maximum spindle torques (Q_{smax}, Q_{sex}). One third of the spindle torque is assumed to be consumed by friction, if not otherwise documented through further analysis.

The diameter of fitted pins d_{fp}, in mm, between the blade and blade carrier can be determined as follow:

$$d_{fp} = 66\sqrt{\frac{Q_s - Q_{fr}}{PCD \cdot z_{pin} \cdot \sigma_{0,2}}}$$

where:

Q_s : Spindle torque, in kNm, equal to:

$$Q_s = Max(1{,}3\ Q_{smax};Q_{sex})$$

Q_{fr} : Friction between connected surfaces:

$Q_{fr} = 0{,}33\ Q_s$

Alternative values of Q_{fr} according to reaction forces due to F_{ex}, or F_f, F_b which ever is relevant, utilising a friction coefficient equal to 0,15 may be used, provided they are approved by the Society.

The stress in the actuating pin can be estimated by:

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

where

$$F = \frac{Q_s - Q_{fr}}{\ell_m}$$

ℓ_m : Distance from the pitching centre of the blade to the pin axis, in m

h_{pin} : Height of actuating pin, in mm

d_{pin} : Diameter of actuating pin, in mm

Q_{fr} : Friction torque in blade bearings acting on the blade palm and caused by the reaction forces due to F_{ex}, or F_f, F_b whichever is relevant, taken $Q_{fr} = 0{,}33\ Q_s$

The blade failure spindle torque

Q_{sex} is not to lead to any consequential damage.

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

$$Q_{samax} = \frac{Q_{sb} + Q_{sf}}{2}$$

where:

Q_{sb} : Spindle torque due to $|F_b|$, in kNm

Q_{sf} : Spindle torque due to $|F_f|$, in kNm

3.4.4 Servo pressure

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

3.5 Propeller fitting to the shaft

3.5.1 Keyless cone mounting

The friction capacity at 0° C is to correspond at least to 2 times the highest value of the peak torque, Q_{peak}, as determined in [2.6] without exceeding the permissible hub stresses. The permissible equivalent uniaxial stress in the hub at 0° C based on the Mises-Hencky criterion σ_E is not to exceed 70% of the yield point or 0,2% proof-stress (0,2% offset yield strength) for the propeller material based on the test piece value. For cast iron the value is not to exceed 30% of the nominal tensile strength.

The minimum surface pressure p_0 at 0°C is to be determined as:

$$p_0 = \frac{2 \cdot S \cdot Q_{peak}}{\pi \cdot \mu \cdot D_S^2 \cdot L \cdot 10^3}$$

where:

μ : Friction coefficient taken as:

- for steel-steel friction:
  $\mu = 0{,}15$
- for steel-bronze friction:
  $\mu = 0{,}13$

These friction coefficients may be increased by 0,04 if glycerine is used in wet mounting.

D_S : Shrinkage diameter at the mid-length of the taper, in m

L : Effective length of taper, in m

S : Safety factor, equal to 2,0

The contact pressure corresponding to the actual pull-up length and to the ambient temperature during the propeller fitting should guarantee that the required safety factor will be fulfilled in ice operating conditions.

3.5.2 Key mounting

Key mounting is not permitted.

3.5.3 Flange mounting

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

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

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

The diameter of shear pins, d_{pin}, in mm, is to be determined as follow:

$$d_{pin} = 66\sqrt{\frac{Q_{peak} \cdot S}{PCD \cdot z_{pin} \cdot \sigma_{0,2}}}$$

where:

z_{pin} : Number of shear pins

S : Safety factor taken as $S = 1{,}3$

The bolts are to be designed so that the blade failure load F_{ex} (See [2.4]) in backward direction does not cause yielding of the bolts. The flange bolt diameter is determined as:

$$d_b = 41\sqrt{\frac{F_{ex} \cdot \left(\frac{0{,}8 \cdot D}{PCD} + 1\right) \cdot \alpha}{\sigma_{0,2} \cdot z_b}}$$

where:

α : Coefficient taken equal to:

- for torque guided tightening:
  $\alpha = 1{,}6$
- for elongation guided:
  $\alpha = 1{,}3$
- for angle guided:
  $\alpha = 1{,}2$
- for elongated by other additional means:
  $\alpha = 1{,}1$

Other α values may be used, if evidence is demonstrated to the Society.

d_b : Flange bolt diameter, in mm

z_b : Number of flange bolts

3.6 Propulsion line components

3.6.1 General

The ultimate load resulting from total blade failure F_{ex} as defined in [2.4] consists of combined axial and bending load components, wherever this is significant. The minimum safety factor against yielding is to be 1,0 for all shaft line components.

The shafts and shafting components, such as bearings, couplings and flanges are to be designed to withstand the operational propeller/ice interaction loads as given in [2].

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

3.6.2 Cumulative fatigue calculation

Cumulative fatigue calculations are to be conducted according to the Miner's rule. A fatigue calculation is not necessary, if the maximum stress is below fatigue strength at 10^8 load cycles.

The torque, Q_{Amax}, and thrust amplitude distribution (spectrum), $Q_A(N)$, in the propulsion line is to be taken as:

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

$Z\ N_{ice}$: Number of load cycles in the load spectrum

The ratio $Q_{Amax}/Q_A(N)$ is illustrated by the example in Fig 10

The load spectrum is divided into NBL load blocks for the Miner summation method. The following formula can be used for calculation of the number of cycles for each load block (See Fig 11).

$$n_i = N_{ice}^{1-\left(1-\frac{i}{NBL}\right)^k} - \sum_{j=2}^{i} n_{j-1}$$

where:

i : Single load block i

NBL : Number of load blocks

Figure 10 : Cumulative torque distribution

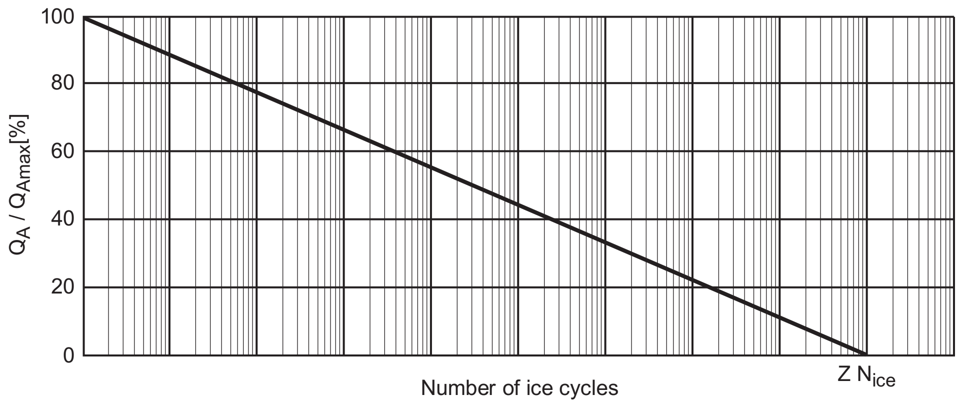
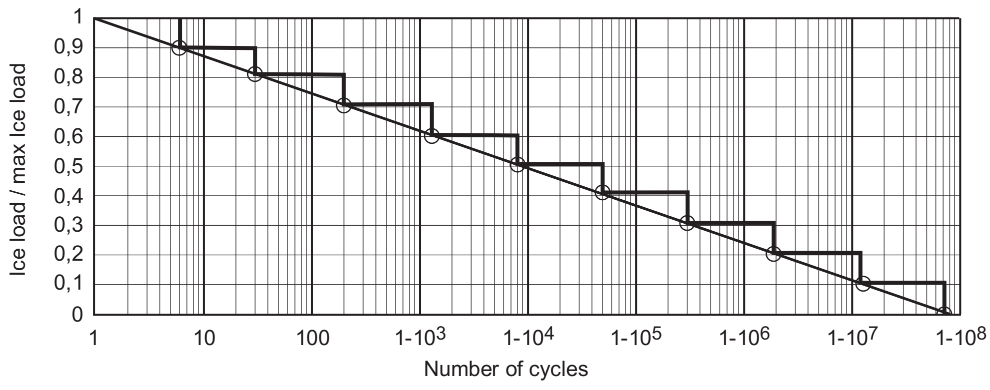


Figure 11 : Example of ice load distribution (spectrum) for the shafting with Weibull exponent k = 1



3.6.3 Propeller shaft

The design of the propeller shaft is to fulfil the following:

a) The blade failure load F_{ex} (See [2.4]) applied parallel to the shaft (forward or backwards) is to not cause yielding. The bending moment need not to be combined with any other loads. The diameter in way of the aft stern tube bearing, d_{pastb}, in mm, is to not be less than:

$$d_{pastb} = 160 \sqrt[3]{\frac{F_{ex} \cdot D}{\sigma_{0,2} \cdot \left[1 - \left(\frac{d_i}{d_p}\right)^4\right]}}$$

where:

d_p : Propeller shaft diameter, in mm

d_i : Propeller shaft inner diameter, in mm

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

b) The stresses due to the peak torque Q_{peak} are to have a minimum safety factor equal to 1,5 against yielding in plain sections and to 1,0 in way of stress concentrations in order to avoid bent shafts.

Minimum diameter of the plain and notched shafts are defined as follow, in mm:

- plain shaft:

$$d_p = 210 \sqrt[3]{\frac{Q_{peak} \cdot S}{\sigma_{0,2} \cdot \left[1 - \left(\frac{d_i}{d}\right)^4\right]}}$$

- notched shaft:

$$d_p = 210 \sqrt[3]{\frac{Q_{peak} \cdot S \cdot \alpha_t}{\sigma_{0,2} \cdot \left[1 - \left(\frac{d_i}{d}\right)^4\right]}}$$

where:

α_t : Local stress concentration factor in torsion

d_i : Actual inner diameter of the propeller shaft, in mm

d : Actual outer diameter of the propeller shaft, in mm

d_p : Minimum required diameter of the propeller shaft, in mm

In any case, the actual notched shaft diameter is not to be less than the required plain shaft diameter.

c) The torque amplitudes (See [2.6.5]) with the corresponding number of load cycles are to be used in an accumulated fatigue evaluation where the fatigue safety factor is S_{fat} = 1,5.

If the plant has high engine excited torsional vibrations (e.g. direct coupled 2-stroke engines), this is also to be considered.

d) The fatigue strengths σ_F and τ_F ($3\ 10^6$ cycles) of shaft materials may be assessed on the basis of the material's yield or 0,2% proof strength as:

$$\sigma_F = 0{,}436 \cdot \sigma_{0,2} + 77 = \sqrt{3} \cdot \tau_F$$

This formula is valid for small polished specimens (no notch) and reversed stresses.

The high cycle fatigue (HCF) is to be assessed based on the above fatigue strengths, notch factors (i.e. geometrical stress concentration factors and notch sensitivity), size factors, mean stress influence and the required safety factor of 1,6 at $3\ 10^6$ cycles increasing to 1,8 at 10^9 cycles.

The low cycle fatigue (LCF) representing 10^4 cycles is to be based on the smaller value of yield or 0,7 of tensile strength/$3^{0,5}$. The criterion utilises a safety factor of 1,25.

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

3.6.4 Intermediate shafts

The design of intermediate shafts are to fulfil the requirements [3.6.3] item b) to item d).

3.6.5 Shaft connections

a) Shrink fit couplings (keyless)

The requirements in [3.5.1] apply with a safety factor S equal to 1,8. The cone angle value should be agreed with the Society.

b) Key mounting

Key mounting is not permitted.

c) Flange mounting

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

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

The flange fillet radius is to be at least 8% of the shaft diameter.

The diameter of ream fitted (light press fit) bolts is to be chosen so that the peak torque is transmitted with a safety factor of 1,9. This accounts for a pre-stress. Pins are to transmit the peak torque with a safety factor of 1,5 against yielding (See d_{pin} in [3.5.3]).

The bolts are to be designed so that the blade failure load (See [2.4]) in backward direction does not cause yielding.

d) Splined shaft connections

Splined shaft connections can be applied where no axial or bending loads occur. A safety factor of S = 1,5 against allowable contact and shear stress resulting from Q_{peak} is to be applied.

e) Gear transmissions

1) Shafts

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

2) Gearing

The gearing are to fulfil following three acceptance criteria:

- Tooth root stresses
- Pitting of flanks
- Scuffing

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

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

The load spectrum (see [3.6]) is to be applied in such a way that the numbers of load cycles for the output wheel are multiplied by a factor equal to the number of pinions on the wheel divided by the number of propeller blades, Z. For pinions and wheels operating at higher speeds the numbers of load cycles are found by multiplication with the gear ratios. The peak torque (Q_{peak}) is also to be considered during calculations.

Cylindrical gears can be assessed on the basis of the international standard ISO 6336 series (i.e. ISO 6336-1:2019, ISO 6336-2:2019, ISO 6336-3:2019, ISO 6336-4:2019, ISO 6336-5:2016 and ISO 6336-6:2019), provided that “method B” is used. Bevel gears can be assessed on the basis of the international standard ISO 10300 series (i.e. ISO 10300-1:2014, ISO 10300-2:2014 and ISO 10300-3:2014).

The ice application factor K which is referenced in ISO 6336 should be updated to take into account ice operation. Calculations should be performed using $K_{A\text{-}Ice}$ calculated according to the following formula:

$$K_{A-Ice} = K_A + \frac{Q_{eq}}{Q_n} \cdot \frac{I}{I_t}$$

where

K_A : Application factor considered for the gearing assuming no ice operation

Q_{eq} : Equivalent ice torque calculated in accordance with ISO 6336 Pt. 6 A.3

The load spectrum is to be divided into 10 load blocks minimum, and the effective number of cycles for each block is calculated with the following formula:

$$n_i = (ZN_{ice})^{\frac{1}{imax}} - \sum_{j=2}^{i} n_{j-1}$$

where

i : Index of each load block (starting at 1 for the highest load value)

n_j : Number of cycles for the load Q_i defined by

$$Q_i = Q_{max}\left[1 - \frac{i-1}{i_{max}}\right]$$

with i_{max} equal to the total number of blocks taken not less than 10.

Tooth root safety is to be assessed against the peak torque, torque amplitudes (with the pertinent average torque) as well as the ordinary loads (open water free running) by means of accumulated fatigue analyses. The resulting factor of safety is to be at least 1,5.

The safety against pitting is to be assessed in the same way as tooth root stresses, but with a minimum resulting safety factor of 1,2.

The scuffing safety (flash temperature method - ref. ISO/TR 13989-1:2000 and ISO/TR 13989-2:2000) based on the peak torque is to be at least 1.2 when the FZG class of the oil, as defined in ISO 14635-1:2000, is assumed one stage below specification.

The safety against subsurface fatigue of flanks for surface hardened gears (oblique fracture from active flank to opposite root) is to be assessed in accordance with the relevant provisions of ISO 6336-4:2019.

3) Gear wheel shaft connections

The torque capacity is to be at least 1,8 times the highest peak torque Q_{peak} (at considered rotational speed) as determined in [3.6] without exceeding the permissible hub stresses of 80% yield.

3.6.6 Clutches

Clutches are to have a static friction torque of at least 1,3 times the peak torque Q_{peak} and dynamic friction torque of 2/3 of the static.

Emergency operation of clutch after failure of e.g. operating pressure is to be made possible within reasonably short time. If this is arranged by bolts, it is to be on the engine side of the clutch in order to ensure access to all bolts by turning the engine.

3.6.7 Flexible couplings

A separation margin of at least 20% is required between the peak torque Q_{peak} and the torque T_{Kmax}, in kNm, where any twist limitation is reached.

$$Q_{peak} < 0{,}8 \cdot T_{Kmax}(N = 1)$$

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

A sufficient fatigue strength 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 10):

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

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

respectively

where:

$Q_r(N=1)$: Torque level corresponding to Q_{peak}, in kNm

$Q_A(N=1)$: Torque level corresponding to Q_{Amax}, in kNm

$$Q_r(N = 5\times10^4) \cdot S < T_{Kmax}(N = 5\times10^4)$$

$$Q_r(N = 1\times10^6) \cdot S < T_{KV}\ldots$$

$$Q_A(N = 5\times10^4) \cdot S < \Delta T_{max}(N = 5\times10^4)$$

S : General safety factor for fatigue, equal to 1,5

The torque amplitude (or range, Δ) is not to lead to fatigue cracking, i.e. exceeding the permissible vibratory torque. The permissible torque is obtained by interpolation in a Weibull torque distribution where T_{Kmax1} and ΔT_{Kmax} refer to 5 10^4 cycles and T_{KV} to 10^6 cycles respectively.

$$T_{Kmax1} \geq Q_r \text{ at } 5\times10^4 \text{ load cycles}$$

See illustration of T_{Kmax1}, ΔT_{Kmax} and T_{KV} = f (time) in Fig 12, Fig 13 and Fig 14 respectively.

Figure 12 : T_{Kmax1}

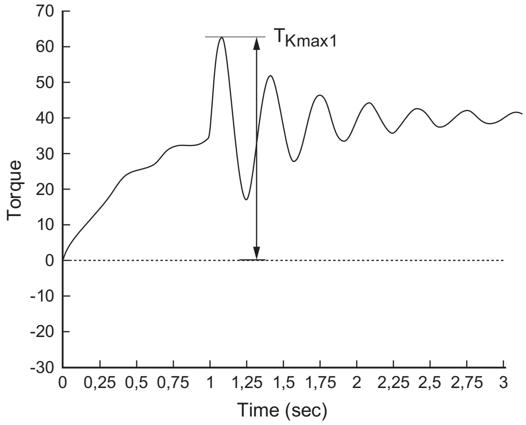


Figure 13 : ΔT_{Kmax}

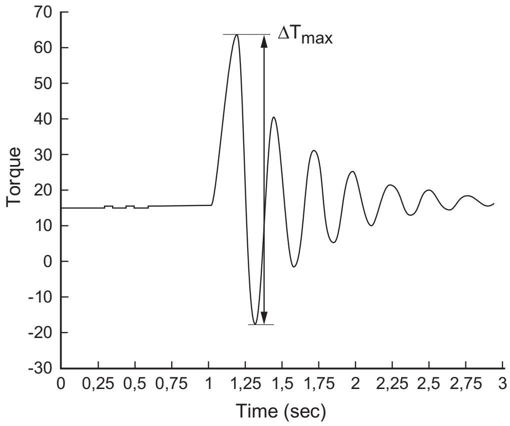
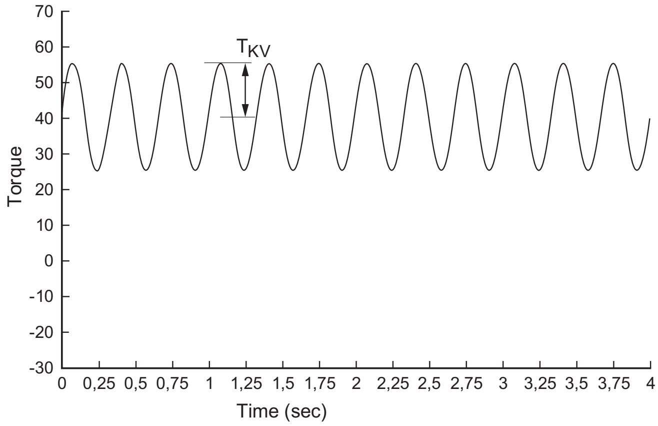


Figure 14 : T_{KV} = f (time)



3.6.8 Braking device

Where the design of the main thrusters does not insure against free rotation of the propeller and shafting in case of failure of the prime mover, provision is to be made for a braking device which ensure the retention of the failed shafting at a speed of the ship not less than 7 knots.

3.6.9 Seals

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

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

3.7 Equipment fastening loading accelerations

3.7.1 General

Essential equipment and supports are to be suitable for the accelerations as indicated in [3.7.2] to [3.7.4] or for those in open water whichever are the greater (See [1.1.2]). Accelerations are to be considered as acting independently.

3.7.2 Longitudinal impact accelerations, a_l

The maximum longitudinal impact acceleration at any point along the hull girder, in m/s², is defined as:

$$a_l = \frac{F_{VB}}{\Delta} \cdot \left(1,1 \cdot \tan(\gamma + \varphi) + 7 \cdot \frac{H}{L}\right)$$

where:

F_{VB}	:	Vertical design ice force, in kN, defined in [2.7.7]
Δ	:	Displacement, in t
γ	:	Bow stem angle at waterline, in deg
φ	:	Maximum friction angle between steel and ice, normally taken as 10 deg, φ = 10
H	:	Distance from the water line to the point being considered, in m
L	:	Ship length between perpendiculars, in m

3.7.3 Vertical impact acceleration, a_v

The combined vertical impact acceleration at any point along the hull girder, in m/s², is defined as:

$$a_v = 2{,}5 \cdot \frac{F_{VB} \cdot F_x}{\Delta}$$

F_x : Coefficient taken as:

- at FP:
  $F_x = 1{,}3$
- at midship perpendicular:
  $F_x = 0{,}2$
- at AP:
  $F_x = 0{,}4$
- at AP for vessels conducting ice breaking astern:
  $F_x = 1{,}3$

Intermediate values are obtained by linear interpolation.

F_{VB}, Δ : Parameters defined in [3.7.2]

3.7.4 Transverse impact acceleration, a_t

The combined transverse impact acceleration at any point along hull girder, in m/s², is defined as:

$$a_t = 3 \cdot \frac{F_{NB} \cdot F_x}{\Delta}$$

F_{NB} : Total design ice force normal to shell plating in the bow area due to oblique ice impact, in kN, defined in [2.7.6]

F_x : Coefficient taken as:

- at FP:
  $F_x = 1{,}5$
- at midship perpendicular:
  $F_x = 0{,}25$
- at AP:
  $F_x = 0{,}5$
- at AP for vessels conducting ice breaking astern:
  $F_x = 1{,}5$

Intermediate values are obtained by linear interpolation.

3.8 Alternative design

3.8.1 General

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

Section 4 Electrical Installations

1 General design requirements

1.1 Arrangement

1.1.1 All electrical equipment not covered by the present Rule Note are to comply with the requirements defined in NR467.

1.1.2 The electrical components of installations are to be designed and constructed to operate satisfactorily under the environmental conditions on board.

In particular, the environmental conditions specified in Sec 1 and in this Section are to be taken into account.

Note 1: The environmental conditions are characterised by:

- one set of variables including climatic conditions (e.g. ambient air temperature and humidity), conditions dependent upon chemically active substances (e.g. salt mist) or mechanically active substances (e.g. dust or oil), mechanical conditions (e.g. vibrations or inclinations) and conditions dependent upon electromagnetic noise and interference, and
- another set of variables dependent mainly upon location on vessels, operational patterns and transient conditions.

1.2 Quality of power supply

1.2.1 All electrical components supplied from the main and emergency systems are to be so designed and manufactured that they are capable of operating satisfactorily under the normally occurring variations in voltage and frequency specified in [1.2.2] and [1.2.3].

1.2.2 For alternating current components the voltage and frequency variations of power supply shown in Tab 1 are to be assumed.

1.2.3 For direct current components the power supply variations shown in Tab 2 are to be assumed.

For direct current components supplied by electrical battery the voltage variations shown in Tab 3 are to be assumed.

Any special system, e.g. electronic circuits, whose function cannot operate satisfactorily within the limits shown in the tables should not be supplied directly from the system but by alternative means, e.g. through stabilized supply.

1.2.4 All electrical equipment are to be designed and constructed to operate satisfactorily with total voltage harmonic distortion defined for the electrical distribution system.

Table 1 : Voltage and frequency variations of power supply in a.c.

Parameter	Variations		
	Continuous		Transient
Voltage	+ 6%	− 10%	± 20% (recovery time: 1,5 s)
Frequency	± 5%		± 10% (recovery time: 5 s)
Note 1: For alternating current components supplied by emergency generating sets, different variations may be considered.			

Table 2 : Voltage variations in d.c.

Parameters	Variations
Voltage tolerance (continuous)	± 10%
Voltage cyclic variation	5%
Voltage ripple (a.c. r.m.s. over steady d.c. voltage)	10%

Table 3 : Voltage variations for battery systems

Systems	Variations
Components connected to the battery during charging (see Note)	+30%, −25%
Components not connected to the battery during charging	+20%, −25%
Note 1: Different voltage variations as determined by the charging/discharging characteristics, including ripple voltage from the charging device, may be considered.	

1.3 Electromagnetic susceptibility

1.3.1 Provisions are to be made to ensure the electromagnetic compatibility of electrical and electronic equipment, in order to prevent malfunction of equipment exposed to electromagnetic disturbances and to prevent undesired electromagnetic emissions which may affect the performance of other equipment. Selection and installation of equipment are to satisfy with the requirements given in IEC 60533.

1.4 Material

1.4.1 In general, and unless it is adequately protected, all electrical equipment is to be constructed of durable, flame-retardant, moisture-resistant materials which are not subject to deterioration in the atmosphere and at the temperatures to which they are likely to be exposed. Particular consideration is to be given to sea air and oil vapour contamination.

Note 1: The flame-retardant and moisture-resistant characteristics may be verified by means of the tests cited in IEC Publication 60092-101 or in other recognised standards.

1.4.2 Insulating materials of live parts are not to contain asbestos and are to have adequate dielectric strength and resistance to creepage currents, moisture, sea air and oil vapour, as well as sufficient mechanical strength, unless suitably protected.

1.4.3 The insulating materials used for winding insulation in machines, apparatus and other equipment for essential services are to comply with the agreed standards. The insulation classes given in Tab 4 may be used in accordance with IEC Publication 60085.

Table 4 : Insulation Classes

Class	Maximum continuous operating temperature, in °C
A	105
E	120
B	130
F	155
H	180

1.5 Construction

1.5.1 All electrical apparatus is to be so constructed as not to cause injury when handled or touched in the normal manner.

1.5.2 The design of electrical equipment is to allow accessibility to each part that needs inspection or adjustment, also taking into account its arrangement on board. Such parts as require repair or replacement while in service are to be easily dismountable.

1.5.3 Enclosures are to be of adequate mechanical strength and rigidity.

1.5.4 Cable entrance are not to impair the degree of protection of the relevant enclosure specified in [1.6].

1.5.5 Where screw fastenings are employed, including those in connection with current-carrying parts, provision is to be made of exclude self-loosening of screws and nuts or, where dismantling and opening are a frequent occurrence, loss of same.

1.5.6 All equipment is generally to be provided with suitable, fixed terminal connectors in an accessible position for convenient connection of the external cables.

1.6 Degree of protection of enclosures

1.6.1 Electrical equipment is to be protected against the ingress of foreign bodies and water.

If not specified within this Rule Note, the minimum degree of protection of electrical equipment located in machinery spaces is to be in compliance with Society's requirements, depending of its place of installation.

1.6.2 The degrees of protection are to be in accordance with:

- IEC Publication No. 60529 for equipment in general
- IEC Publication No. 60034-5 for rotating machines.

2 Electrical propulsion motors

2.1 General requirements

2.1.1 The requirements of the IEC 60034 series apply to the electrical propulsion motors.

2.1.2 The electrical propulsion motors are to be either synchronous or asynchronous. DC motors will be considered on case-by-case basis.

2.1.3 The normal torque available on the electric propulsion motors for manoeuvring is to be such as to enable the vessel to be stopped or reversed when sailing at its maximum service speed.

2.1.4 Adequate torque margin is to be provided for three-phase electric motors to avoid the motor pulling out during rough weather and when turning.

2.1.5 For electrical propulsion plants having only one propulsion motor, the propulsion motor is to be equipped with two stator winding systems which can be disconnected from their respective converter.

2.1.6 The motor is to be capable of withstanding overspeed up to the limit reached in accordance with the characteristics of the overspeed protection device at its specified operational setting.

2.1.7 Motors operating with semiconductor converters are to be designed for the expected harmonics of the system. A sufficient reserve is to be considered for the temperature rise, compared with sinusoidal load.

2.1.8 The winding insulation of motors is to be capable of withstanding the overvoltage which may occur in manoeuvring conditions.

2.1.9 The motor is to be capable to withstand a sudden short-circuit currents at its terminals under nominal load without damage. Steady state short-circuit current of a permanent excited motor is not to cause thermal damages of the motor.

2.1.10 Evaluation based on calculations and/or documented measures is to be submitted to the Society to demonstrate that the electric propulsion motors are capable to withstand the vibrations and shocks expected on board and specified in Sec 1, [1.5.5] and Sec 1, [1.5.6]

2.2 Construction

2.2.1 Materials and construction of electrical machines are to conform to the requirements of NR467, Pt C, Ch 2, Sec 4, [1.1.1].

2.2.2 Shafts are to be made of material complying with the provisions of NR467, Pt C, Ch 2, Sec 4, [1.1.2].

Shaft material for electric propulsion where the shaft is part of the propulsion shafting is to be certified in accordance with Sec 8.

2.2.3 Motors and shaft-lines are to be capable of withstanding all loading conditions. For the torsional vibration characteristics of the electric propulsion plant, the provisions of Sec 3, [2.6] apply.

2.2.4 Where welded parts are foreseen on shafts and rotors, the requirements of NR467, Pt C, Ch 2, Sect 4, [1.1.3] are to be applied.

2.2.5 The electric motors are to be constructed so that, at any speed reached in service, all the moving components are suitably balanced.

2.2.6 Electrical propulsion motor having a rated power of 1 MW and over is to be protected against internal fault due to short-circuit of windings. If differential protection is used for the purpose of internal fault protection, sufficient space for three current transformers and leads of neutral winding ends is to be provided.

2.2.7 Embedded temperature sensors are to be provided in stator windings of a.c. motors rated above 500kW.

2.2.8 The electrical propulsion motors are to be provided with overspeed protection in the event of the propeller breaking down or working clear of water. The speed sensor of the overspeed protection device is to be independent and separate of those used for control and indication. Where a speed measuring system is used for control and indication, the system is to be duplicated with separate sensor circuits and separate power supply.

2.2.9 Insulating materials for windings and other current carrying parts are to comply with the requirements of [1.4.3].

2.2.10 Suitable, fixed terminal connectors are to be provided in an accessible position for connection of the external cables.

2.2.11 All terminal connectors are to be clearly identified with reference to a diagram.

2.3 Cooling system

2.3.1 The cooling system is to ensure sufficient cooling under all load and speed conditions in order that the design temperature rise is not exceeded.

2.3.2 Propulsion motors fitted with forced air ventilation are to be provided with temperature sensor for monitoring the temperature of the cooling air. An alarm is to be given with the excess of the permissible temperature.

2.3.3 Where propulsion motors are forced ventilated, at least two fans, or other suitable arrangements, are to be provided so that limited operation is possible in the event of one fan failing. A visual signal indicating fans operation and an alarm on their failure are to provided.

2.3.4 When liquid cooling is used, the heat exchanger is to be installed so that water leakages and condensed moisture are kept away from the windings, and provision is to be made to alarm leakage.

2.3.5 For machines with a closed circuit cooling system and a heat exchanger, the flow of primary and secondary coolants are to be monitored.

2.3.6 If the cooling system of the propulsion motor fails, an emergency operation mode permitting to ensure restricted service (manoeuvrability) is to be possible after interventions by an operator, for example opening of emergency air flaps. This requirement is applicable to on-board electrical propulsion motors only.

2.4 Protection against moisture and condensate

2.4.1 Provisions are to be made for arrangements preventing the generation and accumulation of moisture and condensate in casing of propulsion motor, in particular, while being idle for a long time. These arrangements may be electric heaters, air dryers, etc.

2.4.2 Propulsion motors is to be equipped with an electric heating designed to maintain the temperature inside the machine at about 3°C above ambient temperature.

2.4.3 A draining arrangement readily accessible for maintenance is to be provided in the lower part of a machine casing for removal of condensate.

2.4.4 Vertically-designed motors are to be fitted on their top with a rigidly secured canopy preventing the ingress of water and foreign objects inside the machine. A lower end shield is to be shaped so as to prevent accumulation of water in way of a bearing.

2.5 Bearings

2.5.1 Bearings are to be efficiently and automatically lubricated at all running speeds and ship's inclinations. Provision is to be made to avoid the possibility of oil splashing or leaking along the shaft and coming into contact with the machine windings or live parts.

2.5.2 The casing of the sleeve bearing is to be fitted with a hole for excessive lubricating oil drain and with an inspection lid in the upper part of the casing. Means for visual indication of oil level or use of an oil gauge is to be provided.

2.5.3 Bearings with forced lubrication are to be provided with redundant pump and an alarm device which will operate in the event of oil pressure loss. Refer to control, alarm and monitoring functions specified in [5.5.1].

2.5.4 All bearings are to be provided with a temperature indicator. For bearings with forced lubrication system an alarm is to be given before the maximum permissible temperature value stated by the manufacturer is exceeded.

2.5.5 Means are to be provided to prevent bearings from being damaged by the flow of currents circulating between them and the shaft. According to the manufacturer’s requirements, electrical insulation of at least one bearing is to be considered.

2.5.6 To support operating capacity of propulsor till special survey the service life of the rolling bearings is to be at least 30 000 hours for the main propulsors and 5 000 hours for auxiliary ones.

3 Electrical slip rings

3.1 General requirements

3.1.1 The purpose of the electrical slip ring is to form a continuous electrical connection between a fixed part (the POD room) and a rotating part (the POD). The electrical slip rings may transfer power or control/automation signals.

3.1.2 High voltage electrical slip rings are to be segregated from slip ring operating at different voltage ratings.

3.1.3 Slip rings fitted with forced cooling system are to be capable of restricted operation in case of loss the cooling system. The cooling system failure is to be alarmed.

3.1.4 Where data transmission is carried out via a bus system, transmission paths are to be duplicated. Failure of each single system is to be alarmed.

3.2 Construction

3.2.1 It is to be taken into account that the mechanical and electrical characteristics of the slip rings can be degraded by contamination or by oxidation.

3.2.2 Enclosures for slip ring assemblies is to ensure at least a degree of protection IP23 according to IEC 60529.

3.2.3 The suitability of used materials at maximum permitted temperature values is to be proven. The permitted conductor temperature values of the connected cables is not to be exceeded.

3.2.4 The maximum temperature rise of external surface parts which can easily be touched in service is not to exceed 15°C.

3.2.5 The enclosure is to be protected against internal condensation.

3.2.6 Clearance and creepage distances for low voltage slip rings

The minimum clearance and creepage distances between non-insulated parts in low voltage slip rings are given in Tab 5. The clearances and creepage distances apply to phase to phase, phase to neutral, phase to earth and neutral to earth.

Table 5 : Clearance and creepage distance for low voltage slip rings

Rated insulation voltage a.c. r.m.s. or d.c. (V)	Minimum clearance (mm)	Minimum creepage distance (mm)
< 250	15	20
> 250 to < 690	20	25
> 690	25	35

Reduced values as specified in IEC Publication 60092-302 may be accepted for type tested slip rings.

3.2.7 Clearance and creepage distances for high voltage slip rings

In general, for non type tested equipment phase-to-phase air clearances and phase-to-earth air clearances between non-insulated parts are to be not less than those specified in Tab 6.

Intermediate values may be accepted for nominal voltages, provided that the next higher air clearance is observed.

In the case of smaller distances, an appropriate voltage impulse test is to be applied.

Creepage distances between live parts and between live parts and earthed metal parts are to be in accordance

with IEC 60092-503 for the nominal voltage of the system, the nature of the insulation material and the transient over-voltage developed by switch and fault conditions.

Table 6 : Minimum clearances

Nominal voltage (kV)	Highest voltage for equipment (kV)	Minimum air clearance (mm)
3 - 3,3	3,6	55
6 - 6,6	7,2	90
10 - 11	12	120
15	17,5	160

3.2.8 Power slip rings are to be capable to withstand without damage the maximum prospective value of the short-circuit current which can occur at its terminals.

3.2.9 High voltage slip rings are to be able to withstand an internal short circuit arcing failure with the maximum duration and magnitude which can occur at this particular point of the installation, without harmful effect to operators. They are to be internal arc classified (IAC). Where they are accessible by authorized personnel only, Accessibility Type A is sufficient (IEC 62271-200 Annex AA 2.2). Installation and location of the slip rings is to correspond with its internal arc classification and classified sides.

4 Specific requirements for podded azimuth propulsion units

4.1 General

4.1.1 The thermal losses are dissipated by the liquid cooling of the bulb and by the internal ventilation of the podded azimuth thrusters. The justification for the evaluation of the heating balance between the sea water and air cooling is to be submitted to the Society.

Note 1: The calculation method used for the evaluation of the cooling system (mainly based on computer programs) is to be documented. The calculation method is to be justified based on the experience of the designer of the system. The results of scale model tests or other methods may be taken into consideration.

4.1.2 The water level in pod bilge is to be monitored with level sensors operating an alarm. In addition, independent sensors are to be provided to detect a high emergency level which prevent false operations and automatically stop the propulsion.

4.1.3 Means for removal of bilge water from the pod housing is to be provided.

4.1.4 An effective fire detection system with the adequate number of sensors of the relevant type is to be provided. This system is to comply with the relevant requirements of NR467.

4.1.5 For pods providing direct access to their interior space for maintenance and inspection, the arrangement is to ensure protection to both equipment and personnel and sufficient illumination and temporary ventilation are to be provided. Entries to these areas are to be locked in such a way that access is only possible, if the personnel cannot be endangered by the drives.

4.2 Equipment and components

4.2.1 If the space, where an electrical machine and other equipment are located, is inaccessible during operation and associated with special environmental conditions (high temperature, humidity, etc.), special measures are to be taken like use of highly reliable materials and components, adequate number of sensors, as well as special means for protection of components against flooding and damages.

4.2.2 The degree of protection of electrical equipment located within a POD housing is not to be less than IP44.

4.2.3 The components, e.g. controls, sensors, slip rings, cable connections and auxiliary drives are to withstand undamaged the strength of vibration specified in Sec 1, [1.5.5] for podded thrusters.

4.2.4 Sensors which can be changed only during dry docking are to be fitted with two sensor elements in the same housing with separate data transmission channels.

4.3 Electric propulsion motor

4.3.1 The electric propulsion motor installed in POD housing may have a degree of protection equal to IP00, except for all terminals, cable glands and busbar connections which are to be at least IP 44. Precautions are to be taken to prevent objects to enter the air gap.

4.3.2 The temperature of bearings is to be monitored by an alarm and protection system. The alarm is to be carried out in two steps: alarm and motor stop. The protection system is to be independent from the temperature indication and temperature alarm.

4.3.3 Oil filling levels in bearing housings are to be monitored during operation and standstill. Any oil leakage is to activate an alarm.

This applies to circulated lubrication systems as well. These systems are additionally to be equipped with lubricating oil flow monitoring. A flow level monitoring is to be independent from the electric propulsion motor control system.

4.3.4 Vibrations of the electric motor are to be monitored. The alarm set point is to be defined in accordance with the manufacturer recommendation.

4.3.5 Humidity is to be monitored for motors with closed air cooling systems. The excess of the permissible humidity level is to be alarmed.

5 Control, monitoring, alarms and safeties

5.1 General requirements

5.1.1 The control, monitoring and safety systems are to be arranged to ensure safe and reliable operation of the azimuth propulsion units and is to include the following main functions:

a) Monitoring of the alarms: any event critical for the proper operation of an essential auxiliary or a main element of the installation requiring immediate action to avoid a breakdown is to activate an alarm
b) Shutdown or slow down when necessary
c) Automatic and manual starting of auxiliaries
d) Indication and control of the propeller speed
e) Indication and control of the direction of thrust and/or direction of rotation for fixed propeller plants
f) Indication and control of the propeller pitch for CP propellers
g) Indication and control of the angular position of the steering propulsion unit (See Sec 5, [2.11])

5.1.2 The loss of power or malfunctioning of any other control and monitoring systems is not to result in loss of propulsion and electric propulsion plant control, ship's steering or azimuth drive.

5.1.3 Safety functions are to be independent of control and monitoring functions.

5.1.4 Control, alarm and safety systems are to be based on the fail-to-safety principle.

5.1.5 Control, monitoring and safety systems are to have self-check facilities. In the event of failure, an alarm is to be activated.

5.1.6 The control and monitoring systems, including computer based systems, are to be type approved by the Society as specified in Section 8.

5.1.7 Detailed indication, alarm and safety requirements regarding automation systems are to be found in Tab 7 and Tab 8. Each row of these tables corresponds to one independent sensor.

Table 7 : Control, monitoring, alarms and safeties for electric propulsion motors

Monitored parameter	Monitoring			Automatic control		
	Limiting value	Local measuring instrument	Alarm, display at main control station **(1)**	Slow-down	Shut-down	Auxiliary stand-by start
ELECTRIC PROPULSION MOTOR						
Stator winding temperature on phase 1, 2, 3	H		X	X		
	HH		X		X	
Cooling air temperature **(2)**	H	Thermometer	X			
Ventilation fans **(2)**	In operation		X			
Ventilation fans **(2)**	Malfunction		X	X		
Secondary coolant temperature **(3)**	H	Thermometer	X			
Secondary cooling system **(3)**	In operation		X			
Secondary cooling system **(3)**	Malfunction		X	X		
Cooling medium	Leakage		X			
Temperature sensor (short-circuit, open circuit, supply failure)	Malfunction		X			
Bearings temperature	H	Thermometer	X			
Bearing lubrication oil pressure (for self-lubricated motor, when the speed is under the minimum RPM specified by the manufacturer, shutdown is to be activated)	L		X	X		
		Manometer				X
Rotation speed	H		X		X	
	Control					
	Indication		X			

(1) Alarm associated to slowdown or shutdown is to be arranged with pre-alarm
(2) For machine provided with forced ventilation
(3) For machine provided with a closed circuit cooling system and a heat exchanger
Note 1: Symbol convention: H = High, HH = High high, L = Low, LL = Low low, X = function is required

Table 8 : Additional control, monitoring, alarms and safeties for podded azimuth thruster

Monitored parameter	Monitoring			Automatic control		
	Limiting value	Local measuring instrument	Alarm, display at main control station **(1)**	Slow-down	Shut-down	Auxiliary stand-by start
ELECTRIC PROPULSION MOTOR						
Bearings temperature (including thrust bearings)	HH		X		X	
Bearings oil level	L+H		X			
	Leakage		X			
Bearing oil flow (for circulated lubrication systems)	L		X			
Humidity level (for motors with closed loop cooling system)	H		X			
Vibrations	H		X			
SHAFT LINES						
Brake or locking device	Engaged		X			
Shaft turning gear (if fitted)	Engaged		X			
Shaft seal oil gravity tank level	L		X			
POD HOUSING						
Bilge level	H		X			
	HH		X		X	
Humidity	H		X			
Fire sensor(s)	Detection		X			

(1) Alarm associated to slowdown or shutdown is to be arranged with pre-alarm
Note 1: Symbol convention: H = High, HH = High high, L = Low, LL = Low low, X = function is required

5.2 Indicating instrument

5.2.1 The following indications are to be provided at the main control station:

a) Temperature of the stator windings, for motors rated above 500 kW

b) Rotation speed

5.3 Alarm system

5.3.1 An alarm system is to be provided, in accordance with the requirements of NR467, Part C, Chapter 3. The system is to give an indication at the control positions when the parameters specified in [5.5] assume abnormal values or any event occurs which can affect the electric propulsion.

5.3.2 The alarm and monitoring system is to be independent of control and safety systems. Partial integration of the alarm and monitoring system with control system may be allowed for integrated systems provided that a single failure does not affect more than one of these functions.

5.3.3 The following requirements are applicable to the alarm system of electrical propulsion:

a) The alarms can be arranged in groups, and shown in the control station. This is acceptable when a discrimination is possible locally

b) Individual alarms are considered as critical and are to be individually activated at the control stations, and acknowledged individually

c) Shutdown activation is to be considered as an individual alarm.

5.4 Safety system

5.4.1 A safety system is to be provided, in accordance with the requirements of NR467, Part C, Chapter 3. System is to be activated automatically in the event of one of the conditions identified in [5.5] occurs.

5.4.2 The safety systems is to be independent of control and alarm systems including sensors so that the faults and failures of those systems including their supply systems would not influence the safety systems.

5.5 Controlled parameters

5.5.1 For electric propulsion motors, parameters according to Tab 7 are to be controlled or monitored, where applicable:

5.5.2 For podded azimuth thrusters, additional parameters according to Tab 8 are to be controlled or monitored, where applicable:

5.5.3 For thruster auxiliaries, additional parameters according to Tab 9 are to be controlled or monitored, where applicable:

Table 9 : Additional control, monitoring, alarms and safeties for thruster auxiliaries

Monitored parameter	Monitoring			Automatic control		
	Limiting value	Local measuring instrument	Alarm, display at main control station **(1)**	Slow-down	Shut-down	Auxiliary stand-by start
HYDRAULIC OIL SYSTEM						
Pump pressure	L	Manometer	X			
Oil temperature	H		X			
Tank level **(2)**	L	Level gauge	X		X	
CONTROLLABLE PITCH PROPELLER						
Hydraulic oil pressure	L	Manometer	X			
Control oil temperature before cooler	H		X			
Tank level	L	Level gauge	X			
Loss of auxiliary power supply (power supply to controls)			X			

(1) Alarm associated to slowdown or shutdown is to be arranged with pre-alarm

(2) The automatic stop of the hydraulic pumps is to be operated in the same circumstances, except where this stop can lead to propulsion stop

Note 1: Symbol convention: H = High, HH = High high, L = Low, LL = Low low, X = function is required

Section 5 Steering Units

1 General

1.1 Application

1.1.1 Types of units

This Section applies to steering-propulsion units intended for main propulsion and steering such as Z/L-geared azimuth thrusters and rotating podded electrical thrusters.

Steering-propulsion units used for dynamic positioning are not covered by this Rule Note.

1.2 Definitions

1.2.1 Steering gear definitions

The definitions relative to the steering gear are provided in Sec 1, [1.4].

1.2.2 Illustration

The steering gear definitions are illustrated in Fig 1.

1.3 Documents available on board

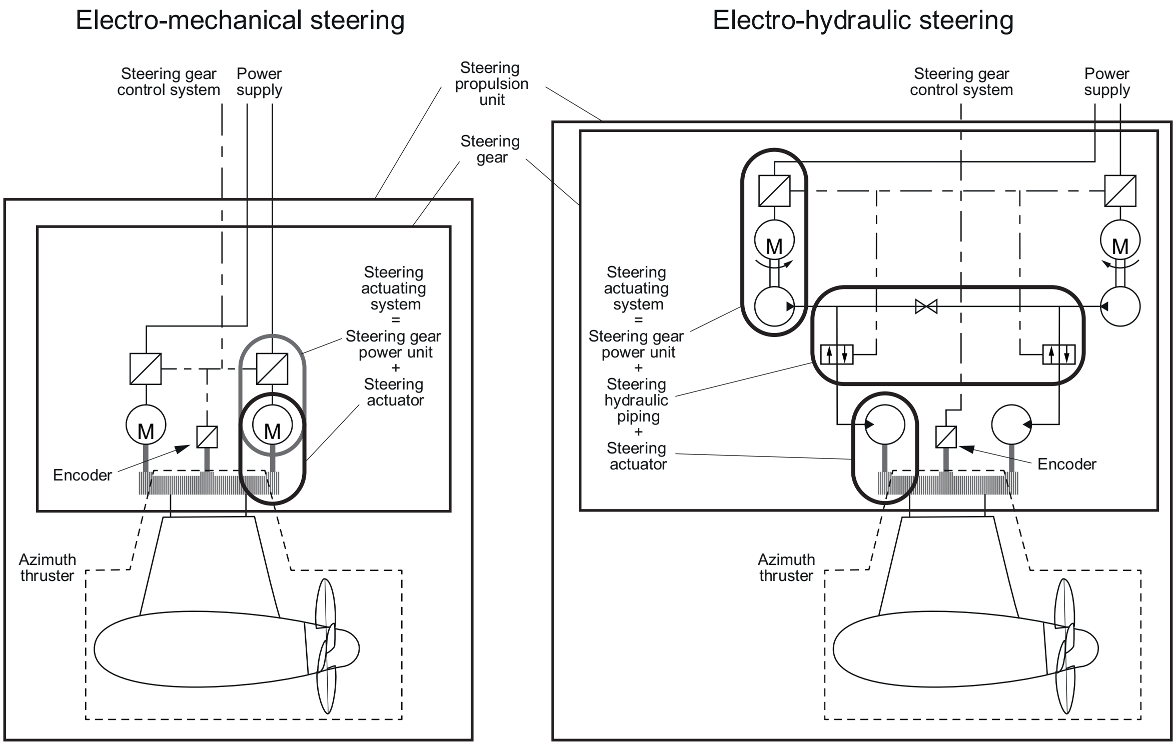
1.3.1 Operating manual and instructions

The operating manual and instructions are to be provided for being available on board of the ship, in a language understandable to the crew.

1.3.2 Performance for going astern

The stopping times, ship headings and distances recorded on trials, together with the results of trials to determine the ability of ships having multiple propulsion/steering arrangements to navigate and manoeuvre with one or more of these devices inoperative, are to be available on board for the use of the master or designated personnel.

Figure 1 : Steering illustrations



2 Steering unit design

2.1 General

2.1.1 Steering gears

Unless expressly provided otherwise, every ship shall be provided with a main steering gear and an auxiliary steering gear to the satisfaction of the Society.

2.1.2 Multiple steering-propulsion units

The steering gear of each steering-propulsion unit is to be power operated and fitted with two or more steering actuating systems capable of satisfying the requirements of [2.4.1], arranged so that after a single failure in its piping system or in one of the steering actuating systems, steering gear capability is maintained.

The above capacity requirements apply regardless whether the steering systems are arranged with common or dedicated power units.

This requirement is to be demonstrated by the submission of a detailed risk analysis in a form of FMEA (Failure Modes and Effects Analysis) or other methodology agreed by the Society.

2.1.3 Single steering-propulsion units

For a ship fitted with a single steering-propulsion unit, a steering gear provided with two or more steering actuating systems and in compliance with [2.4.1] is acceptable. A detailed risk assessment is to be carried out in order to demonstrate that in the case of any single failure in the steering gear, control system and power supply the ship steering is maintained.

2.2 Steering gear compartment

2.2.1 Arrangement

The steering gear compartment shall be:

- *readily accessible and, as far as practicable, separated from machinery spaces; and*
- *provided with suitable arrangements to ensure working access to steering gear machinery and controls. These arrangements shall include handrails and gratings or other non-slip surfaces to ensure suitable working conditions in the event of hydraulic fluid leakage.*

2.3 Design principle

2.3.1 General

The steering-propulsion unit is to be constructed with sufficient strength, capacity and the necessary supporting systems to provide reliable propulsion and steering to the vessel in all operating conditions including the maximum ahead service speed.

2.3.2 Operational limitations

It is to be possible to override operational limitations (e.g. steering angle limits) dedicated to equipment protection by remotely controlling the propulsion machinery from the navigating bridge, when required to initiate an emergency manoeuvre. The override procedure is to be documented in the operating manual of the unit and displayed at its control position(s). (See [2.11.7])

2.3.3 Ships fitted with multiple steering-propulsion units

For a ship fitted with multiple steering-propulsion units, each of them is to be equipped with its own dedicated steering gear or other device to change the steering angle.

2.3.4 Sealing

Sealing arrangement to prevent seawater from gaining access to internal parts of the steering-propulsion units is to contain at least two separate, closely effective sealing elements.

2.4 Strength, performance and power operation

2.4.1 Steering gear capability

The steering gear is to be capable:

a) When operated with all steering actuating systems of changing direction of the steering-propulsion unit from one side to the other at declared steering angle limits at an average turning speed of not less than 2,3°/s with the ship running ahead at maximum ahead service speed; and

b) When supplied by the alternative power supply required in [2.9.2], with one steering actuating system not in operation, of changing direction of the steering-propulsion unit from one side to the other at declared steering angle limits at an average turning speed of not less than 0,5°/s with the ship running ahead at one half of the maximum ahead service speed or 7 knots, whichever is the greater; and

c) To be operated at maximum astern speed and declared steering angle limits without damage.

2.4.2 Prevention of sudden turning

Steering-propulsion units are to be prevented from sudden turning in the case of single failure either in the steering gear or in control system power supply. Where brakes are provided they are to be of the fail-to-close type and their aggregate capacity is not to be less than peak steering torque.

2.4.3 Gears of the steering units

Gears of the steering gear are to be calculated in accordance with the provisions of NR467, Pt C, Ch 1, Sec 6 and are to have minimum safety factors corresponding to auxiliaries. The number of cycles to be considered is to be submitted to the Society for approval.

2.4.4 Sealing arrangements

For water/oil seals, considerations must be conducted to avoid flooding in case of single failure affecting the sealing. In this regard, water/oil seals are to be approved by the Society for the purpose of achieving the sealing against the possible sea water ingress and the leakage of the lubricant media.

2.5 Design torque for thruster in ice

2.5.1 Steering gear design torque in ice

The steering gear design torque in ice T_{steer}, in Nm, is to be obtained from the following formula:

$T_{steer} = 0,6\ (Q_{max} / 0,8\ R)\ L$

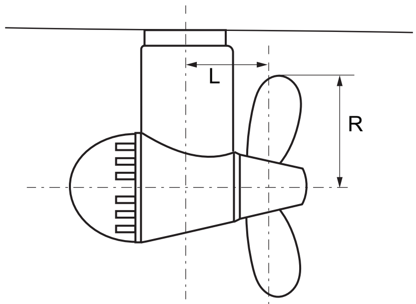
where:

Q_{max}: Maximum torque on the propeller due to ice propeller interaction, in Nm (see Sec 3, [2.6.1])

R : Propeller radius, in m

L : Distance from the propeller plane to steering axis, in m (see Fig 2).

Figure 2 : Distance from the propeller plane to steering axis



2.5.2 Protection of the steering gear against excessive torque

The steering gear is to be protected by torque relief arrangements, assuming the turning speed given in Tab 1, without an undue pressure rise or excessive current, as applicable.

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

Table 1 : Steering gear turning speeds

	POLAR CLASS 1 or **POLAR CLASS 2** **icebreaker 1** or **icebreaker 2**	**POLAR CLASS 3** to **POLAR CLASS 5** **icebreaker 3** to **icebreaker 5**	**POLAR CLASS 6** and **POLAR CLASS 7** **icebreaker 6** and **icebreaker 7**
Turning speed, in deg/s	10	7,5	6

2.6 Availability

2.6.1 Hydraulic power supply

The hydraulic system intended for main and auxiliary steering gear is to be independent of all other hydraulic systems of the ship.

2.6.2 Non-duplicated components

Special consideration is to be given to the suitability of any essential component which is not duplicated.

2.6.3 Hydraulic locking

Where the steering gear is so arranged that more than one system (either power or control) can be simultaneously operated, the risk of hydraulic locking caused by single failure is to be considered.

2.6.4 Emergency turning mechanism

For the case of single steering-propulsion unit application, an emergency turning mechanism may be required as a consequence of the risk assessment referred to in [2.1.3].

2.7 Hydraulic systems

2.7.1 General

The design pressure for calculations to determine the scantlings of piping and other steering gear components subjected to internal hydraulic pressure shall be at least 1,25 times the maximum working pressure to be expected under the operational conditions specified in [2.4.1], taking into account any pressure which may exist in the low pressure side of the system. At the discretion of the Society, fatigue criteria shall be applied for the design of piping and components, taking into account pulsating pressures due to dynamic loads.

2.7.2 Relief valves

Relief valves shall be fitted to any part of the hydraulic system which can be isolated and in which pressure can be generated from the power source or from external forces. The setting of the relief valves shall not exceed the design pressure. The valves shall be of adequate size and so arranged as to avoid an undue rise in pressure above the design pressure.

The setting pressure of the relief valves is not to be less than 1,25 times the maximum working pressure.

The minimum discharge capacity of the relief valve(s) is not to be less than the total capacity of the pumps which can deliver through it (them), increased by 10%. Under such conditions, the rise in pressure is not to exceed 10% of the setting pressure. In this respect, due consideration is to be given to the foreseen extreme ambient conditions in relation to oil viscosity.

2.7.3 Hydraulic oil reservoirs

Hydraulic power-operated steering gear shall be provided with the following:

- *a low level alarm for each hydraulic fluid reservoir to give the earliest practicable indication of hydraulic fluid leakage. Audible and visual alarms shall be given on the navigating bridge and in the machinery space where they can be readily observed; and*
- *a fixed storage tank having sufficient capacity to recharge at least one power actuating system including the reservoir, where the main steering gear is required to be power operated. The storage tank shall be permanently connected by piping in such a manner that the hydraulic systems can be readily recharged from a position within the steering gear compartment and shall be provided with a contents gauge.*

2.7.4 Hydraulic pumps

Provision is to be made to prevent the pumped fluid from penetration to the bearings. However, this does not apply to the pumps where the pumped fluid is employed for lubrication of bearings.

a) Safety devices

In pumps intended for transferring flammable liquids, the by-pass from safety valves is to be effected into the suction side of the pump or to the suction portion of the pipe.

Provision is to be made to prevent hydraulic impacts; use of the by-pass valves for this purpose is not acceptable.

b) Self-priming pumps

The pumps provided with self-priming devices are to ensure operation under “dry suction” conditions. The self-priming pumps are to have the place for connecting a vacuum pressure gauge.

2.7.5 Filters

a) Hydraulic power-operated steering gear shall be provided with arrangements to maintain the cleanliness of the hydraulic fluid taking into consideration the type and design of the hydraulic system.

b) Filters of appropriate mesh fineness are to be provided in the piping system, in particular to ensure the protection of the pumps.

2.8 Cooling systems

2.8.1 Design of ventilation and cooling systems

The ventilation and cooling systems are to maintain the machinery and equipment of steering-propulsion units within the temperatures for which they were designed to operate.

2.8.2 Water cooling

Where water cooling is used the cooler is to be arranged to avoid water leakage inside the steering-propulsion units.

2.9 Power supply

2.9.1 Main power supply

Each steering gear is to be served by at least two exclusive circuits fed directly from the main switchboard; however, one of the circuits may be supplied through the emergency switchboard.

2.9.2 Alternative power supply

This requirement applies to the steering-propulsion units having a certain proven steering capability due to ship speed also in case propulsion power has failed.

Where the propulsion power exceeds 2 500 kW per steering-propulsion unit, an alternative power supply, either from the emergency source of electrical power or from an independent source of power located in the steering gear compartment is to be provided, sufficient at least to supply the steering arrangement such that the latter is able to perform the duties defined in [2.4.1], item b).

This power source is to be activated automatically within 45 s in the event of failure of the main source(s) of electrical power. This independent source of power is to be used only for this purpose.

This alternative source is also to supply the steering gear control system and the steering gear response indicator.

In every ship of 10 000 gross tonnage and upwards, the alternative power supply is to have a capacity for at least 30 min of continuous operation and in any other ship for at least 10 min.

2.10 Electrical system

2.10.1 Availability

The electrical system of the steering gear is to be so arranged that any single failure will not render more than one steering actuating system inoperative.

2.10.2 Power circuit supply

The circuits supplying electro-mechanical steering gear or electro-hydraulic steering gear are to have adequate rating for supplying all motors which can be simultaneously connected to them and may be required to operate simultaneously.

2.10.3 Supply of motor and associated control circuits

Each electric motor of the steering gear power units is to be provided with its own separate motor starter gear, located as far as practicable within the POD room. Controls for starting and stopping of each motor are to be served by its own control circuits supplied from its own power circuit.

2.10.4 Electrical motors

The rating of electric motors driving the steering gear is to be determined based on the breakaway torque and maximum working torque of the steering gear under all operating conditions, but the motors have to satisfy at least the short-term operating conditions during not less than 30 min.

2.10.5 Starting and stopping of motors for steering gear power units

Electrical motors of steering gear power units are to be:

a) Arranged to restart automatically when power is restored after a power failure; and

b) Capable of being brought into operation from a position on the navigation bridge.

2.10.6 Electrical protection of electro-mechanical steering gears

Electric drives of electro-mechanical steering gears are to be protected against overcurrent (for example by converter, if applicable) and short circuit. They are to be able to supply 160% of the torque necessary for the rated speed of movement in accordance with [2.4.1] for a minimum of 60 seconds.

2.10.7 Electrical protection of electro-hydraulic steering gears

Electro-hydraulic steering gears are to be provided with short-circuit protection and an overload alarm. Protection against excess current, including starting current, if provided, is not to be less than twice the full load current of the motor or circuit so protected, and is to be arranged to permit the passage of the appropriate starting currents.

Where a three-phase supply is used an alarm is to be provided that will indicate failure of any one of the supply phases.

Steering gear motor circuits obtaining their power supply via an electronic converter, e.g. for speed control, and which are limited to full load current are exempt from the requirement to provide protection against excess current, including starting current, of not less than twice the full load current of the motor. The required overload alarm is to be set to a value not greater than the normal load of the electronic converter.

Note 1: Normal load is the load in normal mode of operation that approximates as close as possible to the most severe conditions of normal use in accordance with the manufacturer's operating instructions.

2.11 Control and monitoring

2.11.1 Steering gear control

Steering-propulsion units are to be provided with remote steering control from the navigation bridge and local steering control from the steering-propulsion unit room. These systems may have a common steering wheel or level.

The local control system is to be independent of the remote control system and is to include all commands and monitoring necessary for a safe operation of the steering system.

The remote steering control system from the navigating bridge is to include two independent means of steering:

- so arranged that a mechanical or electrical failure in one of them will not render the other one inoperative, and
- in accordance with [2.11.4]; and
- supplied also by the alternative power source when required by [2.9.2].

2.11.2 Multiple steering-propulsion units

Where the steering means of the ship consists of two or more steering-propulsion units:

- independent control systems are to be provided for the steering of each steering-propulsion unit; and
- their control system is to include a device ensuring an automatic synchronisation of the steering-propulsion unit rotation, unless each steering-propulsion unit is so designed as to withstand any additional forces resulting from the thrust exerted by the other steering-propulsion units

2.11.3 Control systems

Each steering gear control system is to comply with the following:

- it is to be served by its own separate circuit supplied from a steering gear power circuit from a point within the pod room, or directly from switchboard busbars supplying that steering gear power circuit at a point on the switchboard adjacent to the supply to the steering gear power circuit
- means are to be provided in the POD room for disconnecting any control system operable from the navigation bridge from the steering gear it serves
- the system is to be capable of being brought into operation from a position on the navigating bridge
- in the event of failure of electrical power supply to the control system, an audible and visual alarm is to be given on the navigation bridge
- short-circuit protection only is to be provided for steering gear control supply circuits.

2.11.4 Separation of control systems and components

a) All electrical components of the steering gear control systems are to be duplicated. This does not require duplication of the steering wheel or steering lever.

b) Duplicated steering gear control systems with their associated components are to be separated as far as practicable.

c) Wires, terminals and the components for duplicated steering gear control systems installed in units, control boxes, switchboards or bridge consoles are to be separated as far as practicable. Where physical separation is not practicable, separation may be achieved by means of a fire-retardant plate.

d) If a joint steering mode selector switch (uniaxial switch) is employed for both steering gear control systems, the connections for the control systems are to be divided accordingly and separated from each other by an isolating plate or air gap.

e) In the case of double follow-up control, the amplifier is to be designed and fed so as to be electrically and mechanically separated. In the case of non-follow-up control and follow-up control, it is to be ensured that the follow-up amplifier is protected selectively.

f) Where additional control systems are provided, e.g. steering lever or autopilot, their control circuits is to be designed for all-pole disconnection.

g) The feedback units and limit switches, if any, for the steering gear control systems are to be separated electrically and mechanically connected to the slewing gear separately.

2.11.5 Displays and alarms

The following displays and alarms are to be provided in the locations indicated in Tab 2, as far as applicable:

2.11.6 Angular position indication

The angular position of each steering-propulsion units is to be:

a) Indicated on the navigating bridge. The thruster angle indication is to be independent of the steering gear control system and be supplied through the emergency switchboard, or by an alternative and independent source of electrical power such as that referred to in [2.9.2];

b) Recognisable in POD room.

2.11.7 Angle limitation

The thrust azimuth angle is to be limited related to the set ship's speed so that the safety of the ship is not endangered (due to excessive thrust while turning). Similarly the propulsion power is to be limited, related to the actual azimuth angle so that the safety of the ship is not endangered. The limitation (interlock) is to be provided redundantly and independently of the control of the azimuth angle (pod turning).

Reaching or exceeding the permissible limitations of the azimuth angle is to be alarmed.

After triggering the limitation, it is to be possible to move the azimuth drive back to the permitted angles of the drive turn without manual reset.

Table 2 : Displays and alarms

<table>
<tr><th rowspan="2" colspan="2">Item</th><th rowspan="2">Display</th><th rowspan="2">Alarms (audible and visible)</th><th colspan="3">Location</th></tr>
<tr><th>Navigation Bridge</th><th>Engine Control Room (1)</th><th>POD Room</th></tr>
<tr><td colspan="2">Indication that electric motor of each power unit is running</td><td>X</td><td></td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Angular position of steering-propulsion units</td><td>X</td><td></td><td>X</td><td></td><td>X</td></tr>
<tr><td colspan="2">Power supply failure of each steering gear power unit</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Power supply failure of each control system</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Overload of electric motor of each steering gear power unit</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Phase failure of electric motor of each steering power unit</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Earth fault on AC and DC circuits</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td rowspan="4">Control system failures</td><td>Loop failures in closed loop systems, both command and feed back loops (2)</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td>Data communication errors</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td>Programmable system failures (Hardware and software failures)</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td>Deviation between thruster steering order and feedback (3)</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Electrical supply to the steering gear systems available</td><td>X</td><td></td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Low oil level in any tank of the hydraulic system</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Hydraulic locking</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Hydraulic oil leakage detection from actuating system</td><td></td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Hydraulic oil pump discharge pressure indication and low pressure alarm</td><td>X</td><td>X</td><td>X</td><td>X</td><td></td></tr>
<tr><td colspan="2">Steering brake engaged (if fitted)</td><td>X</td><td></td><td>X</td><td>X</td><td>X</td></tr>
<tr><td colspan="7">(1) Common alarm may be accepted if individual alarms are available locally.
(2) Normally short circuit, broken connections and earth faults
(3) Deviation alarm is to be initiated if the thruster actual position does not reach the set point within acceptable time limits for the closed loop control systems (e.g. follow-up control and autopilot). Deviation alarm may be caused by mechanical, hydraulic or electrical failures.</td></tr>
</table>

Section 6 Structural Assessment of the Thruster Body in Ice

1 General

1.1 Application

1.1.1 Types of units

Requirements of this Section apply, in addition to the general requirements provided in Sec 2, to the following types of propulsion thrusters intended for navigation in ice-infested waters:

- fixed and azimuth thrusters
- podded and geared thrusters
- thrusters with and without nozzle.

2 Design of main propulsion thrusters

2.1 Design principle

2.1.1 Load scenario types

Thrusters have to be designed for thruster body/ice interaction loads. Load formulae are given for estimating once in a lifetime extreme loads on the thruster body, based on the estimated ice condition and ship operational parameters. Two main ice load scenarios have been selected for defining the extreme ice loads. Examples of loads are illustrated in Fig 1. In addition, blade order thruster body vibration responses may be estimated for propeller excitation. The following load scenario types are considered:

a) Ice block impact to the thruster body or propeller hub

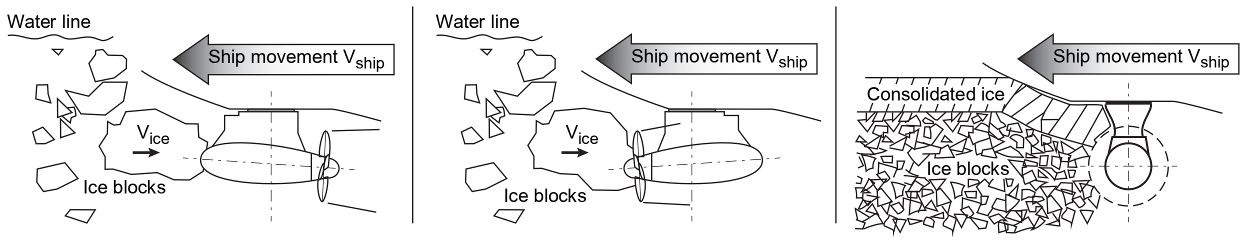
b) Thruster penetration into an ice ridge that has a thick consolidated layer

c) Vibratory response of the thruster at blade order frequency.

2.1.2 Blade damage

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

Figure 1 : Examples of load scenario types



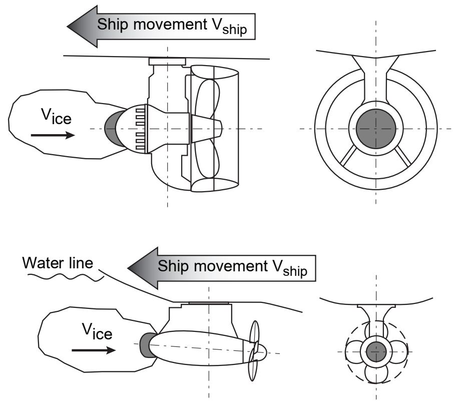
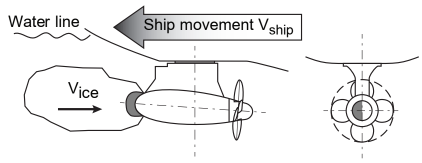
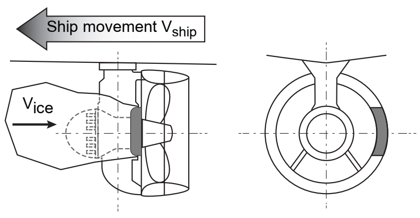
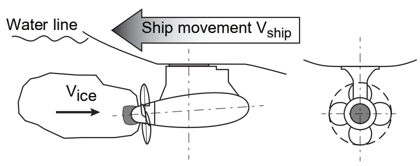
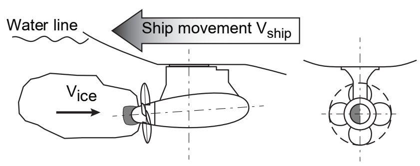
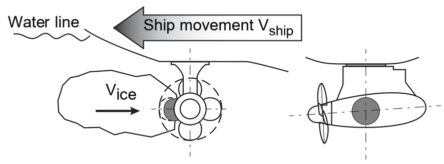
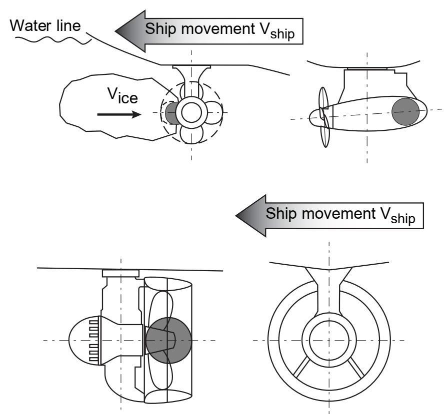
2.2 Extreme ice impact loads

2.2.1 Load cases

The thruster has to withstand the loads occurring when the design ice block having a thickness H_{ice} defined in Sec 3, Tab 2 impacts on the thruster body.

Load cases for impact loads are given in Tab 1. The impact speed values that correspond to the relative movement between the ship and the ice block are given in Tab 3 and Tab 4. The contact geometry is estimated to be hemispherical in shape. If the actual contact geometry differs from the shape of the hemisphere, a sphere radius has to be estimated so that the growth of the contact area as a function of penetration of ice corresponds as closely as possible to the actual geometrical shape penetration.

Table 1 : Load cases for azimuthing thruster impact loads

Load Case	Force	Loaded area	
Load case T1a Symmetric longitudinal ice impact on thruster	F_{ti}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T1b Non-symmetric longitudinal ice impact on thruster	50% of F_{ti}	Uniform distributed load or uniform pressure, which are applied on the other half of the impact area.	
Load case T1c Non-symmetric longitudinal ice impact on nozzle	F_{ti}	Uniform distributed load or uniform pressure, which are applied on the impact area. Contact area is equal to the nozzle thickness (H_{nz})* the contact height (Hi_{ce}).	
Load case T2a Symmetric longitudinal ice impact on propeller hub	F_{ti}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T2b Non-symmetric longitudinal ice impact on propeller hub	50% of F_{ti}	Uniform distributed load or uniform pressure, which are applied on the other half of the impact area.	
Load case T3a Symmetric lateral ice impact on thruster body	F_{ti}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T3b Non-symmetric lateral ice impact on thruster body or nozzle	F_{ti}	Uniform distributed load or uniform pressure, which are applied on the impact area. Nozzle contact radius R to be taken from the nozzle length.	

Note 1: For fixed thrusters, only the relevant load cases are to be considered.

2.2.2 Ice impact load

The ice impact contact load F_{ti}, in kN, is to be calculated as follows:

$$F_{ti} = 34{,}5 \cdot C_{DMI} \cdot R_C^{0,5} \cdot (m_{ice} v_S^2)^{0,333}$$

where:

R_C : Impacting part sphere radius, in m, as shown in Fig 2

m_{ice} : Ice block mass, in kg, as given in Tab 2

v_s : Impact speed, in m/s, as given in Tab 3 and Tab 4. On a case by case basis, v_s can also be taken as the ship in question's actual design operation speed in ice, subject to the Society agreement.

C_{DMI} : Dynamic magnification factor for impact loads. If unknown, C_{DMI} may to be taken from Tab 2

For impacts on non-hemispherical areas, such as the impact on the nozzle, R_c is to be replaced by the equivalent impact sphere radius R_{ceq}, in m, to be estimated using the equation below:

$$R_{ceq} = \sqrt{\frac{A}{\pi}}$$

where:

A : Contact area, in m², as shown in Tab 1

Alternatively, where the end of the thruster body opposite to the propeller is not hemispherical, the value to be considered for R_c may be taken as half of the largest cross-sectional area.

If the 2*R_{ceq} is greater than the ice block thickness given in Tab 2, the radius is set to half of the ice block thickness.

For impact on the thruster side, the pod body diameter can be used as basis for determining the radius. For impact on the propeller hub, the hub diameter can be used as basis for the radius.

Note 1: The longitudinal impact speed in Tab 3 and Tab 4 refers to the impact in the thruster's main operational direction. For the pulling propeller configuration, the longitudinal impact speed is used for load case T2, impact on hub; and for the pushing propeller unit, the longitudinal impact speed is used for load case T1, impact on thruster end cap. For the opposite direction, the impact speed for transversal impact is applied.

Figure 2 : Dimensions used for R_c

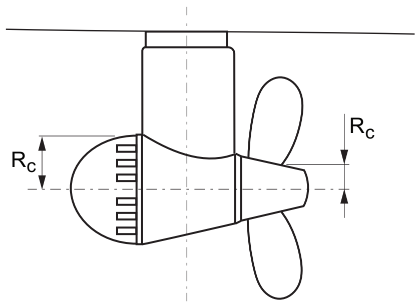


Table 2 : Parameter values for ice dimensions and dynamic magnification

Notations **POLAR CLASS** or **icebreaker**	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Thickness of the design ice block impacting thruster (2/3 of H_{ice}), in m	2,67	2,33	2	1,67	1,33	1,17	1,0
Extreme ice block mass (m_{ice}), in kg	103 530	69 360	43 680	25 280	12 940	8 760	5 460
C_{DMI} (if not known)	1,3	1,3	1,3	1,3	1,3	1,3	1,2

Table 3 : Impact speeds for aft centerline thruster, in m/s

Notations **POLAR CLASS** or **icebreaker**	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Longitudinal impact in main operational direction	11,5	10,75	9,6	8,4	7,2	6	5
Longitudinal impact in reversing direction (pulling unit propeller hub or pushing unit cover end cap impact)	7,5	7	6,25	5,5	4,7	4	3
Transversal impact in bow first operation	3,5	3,25	3,25	3	3	3	2
Transversal impact in stern first operation (double acting ship)	4,5	4,25	4	4	4	4	3

Table 4 : Impact speeds for aft, wing, bow centerline and bow wing thrusters, in m/s

Notations **POLAR CLASS** or **icebreaker**	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Longitudinal impact in main operational direction	11,5	10,75	9,6	8,4	7,2	6	5
Longitudinal impact in reversing direction (pulling unit propeller hub or pushing unit cover end cap impact)	7,5	7	6,25	5,5	4,7	4	3
Transversal impact	4,5	4,25	4	4	4	4	3

2.3 Extreme ice loads on thruster body when penetrating an ice ridge

2.3.1 Description of the scenario

In this load scenario, the ship is penetrating a ridge in thruster first mode with a ridge penetration speed. This situation occurs when a ship with a thruster at the bow moves forward, or a ship with a thruster astern moves in backing mode. The maximum load during such an event is considered the extreme load. An event of this kind typically lasts several seconds, due to which the dynamic magnification is considered negligible and is not taken into account.

2.3.2 Ridge penetration load

The ridge penetration load F_{tr}, in kN, is to be calculated for the load cases shown in Tab 7, using the formula below:

$$F_{tr} = 32 \cdot v_s^{0,66} \cdot H_r^{0,9} \cdot A_t^{0,74}$$

where:

v_s : Ridge penetration speed, in m/s, as given in Tab 5 and Tab 6. On a case by case basis, v_s can also be derived from the ship actual design operation speed in ice, subject to the Society agreement, in m/s

H_r : Total thickness of the design ridge, in m, as given in Tab 5 and Tab 6

A_t : Projected area of the thruster, in m^2, as shown in Tab 7

When calculating the contact area for thruster-ridge interaction, the loaded area in vertical direction is limited to the ice ridge thickness as shown in Fig 3.

The loads must be applied as uniform distributed load or uniform pressure over the thruster surface.

Figure 3 : Schematic figure showing the reduction of the contact area by the maximum ridge thickness

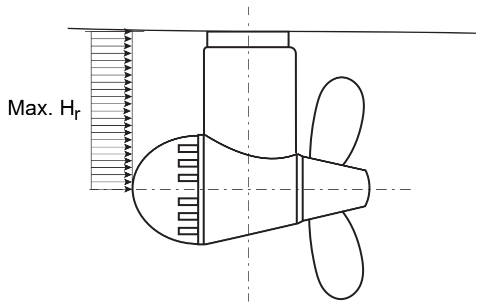


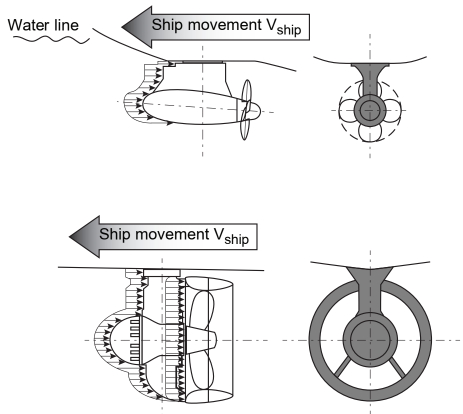
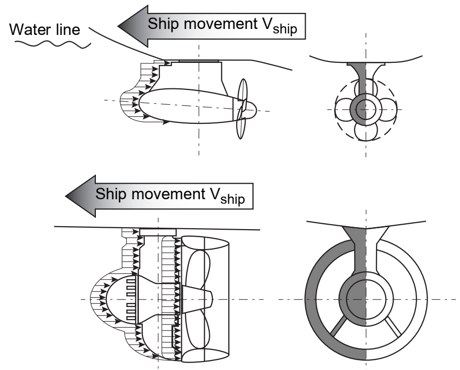
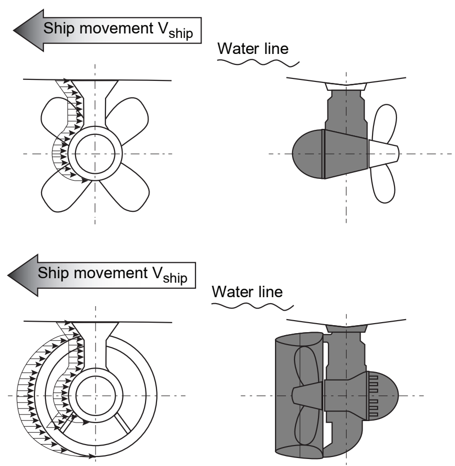
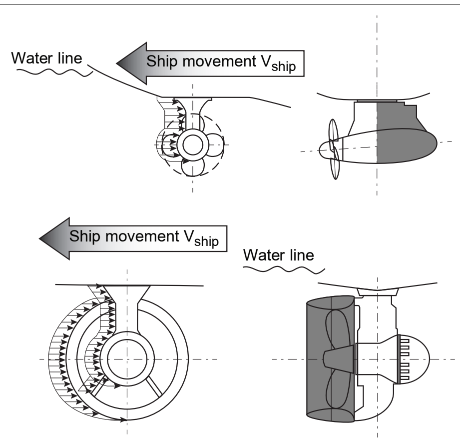
Table 5 : Parameters for calculating maximum loads when thruster penetrates an ice ridge
Aft thrusters, bow first operation

Notations **POLAR CLASS** or **icebreaker**		**1**	**2**	**3**	**4**	**5**	**6**	**7**
Total thickness of the design ridge (H_r), in m		21	17	14	12	10	8	8
Ridge penetration speed (v_s), in m/s	Longitudinal loads	7	6	5,5	5	4,5	4	2
	Transversal loads	3,5	3	2,75	2,5	2,25	2	1

Table 6 : Parameters for calculating maximum loads when thruster penetrates an ice ridge
Thruster first mode such as double acting ships

Notations **POLAR CLASS or icebreaker**		**1**	**2**	**3**	**4**	**5**	**6**	**7**
Total thickness of the design ridge (H_r), in m		21	17	14	12	10	8	8
Ridge penetration speed (v_s), in m/s	Longitudinal loads	8	7	6,75	6,5	6,25	6	4
	Transversal loads	4,5	3,5	3,4	3,25	3	3	2

Table 7 : Load cases for ridge ice loads

Load Case	Force	Loaded area	
Load case T4a Symmetric longitudinal ridge penetration loads	F_{tr}	Uniform distributed load or uniform pressure, which are applied symmetrically on the impact area.	
Load case T4b Non-symmetric longitudinal ridge penetration loads	50% of F_{tr}	Uniform distributed load or uniform pressure, which are applied on the other half of the contact area.	
Load case T5a Symmetric lateral ridge penetration loads for ducted azimuthing unit and pushing open propeller unit	F_{tr}	Uniform distributed load or uniform pressure, which are applied symmetrically on the contact area.	
Load case T5b Non-symmetric lateral ridge penetration loads for all azimuthing units	50% of F_{tr}	Uniform pressure applied on the other half of the contact area	
Note 1: For fixed thrusters, only the relevant load cases are to be considered.			

2.4 Acceptability criterion for static loads

2.4.1 Stresses in the thruster body

The stresses in the thruster have to be calculated for the extreme once in a lifetime loads described in [2]. The nominal von Mises stresses on the thruster body must have a safety margin of 1,3 against yielding strength of the material. At areas of local stress concentrations, stresses must have a safety margin of 1,0 against yielding.

2.4.2 Stresses in connection components

The slewing bearing, bolt connections and other components must be able to maintain operability without incurring damage that requires repair when subject to loads given in [2.2] and [2.3] multiplied by a safety factor of 1,3.

2.5 Thruster body global vibration

2.5.1 General

It is to be shown that either:

- the first blade order excitations frequencies (above 50% of maximum power) are outside a range of 20% above and 20% below the thruster natural frequencies corresponding to global modes of vibration
- the structure is designed to comply with the criterion defined in [2.4] when subject to vibratory loads during resonance above 50% of maximum power.

When estimating thruster global natural frequencies in the longitudinal and transverse direction, the damping and added mass due to water are to be taken into account. In addition to this, the effect of ship attachment stiffness is to be modeled.

Vibratory loads may be estimated using a recognized simplified method (i.e. Guidelines for the application of the Finnish-Swedish Ice Class Rules).

3 Design of the thruster components

3.1 Hull supports of the azimuth propulsion system

3.1.1 General

The scantlings of the hull supports of the thruster are to be verified through direct calculations, to be carried out in accordance with Sec 2, [1.7] with due consideration of the condition stated in Sec 2, [1.7.1].

3.1.2 FEM model

The FEM model is defined in Sec 2, [1.7.2].

3.1.3 Loads

The loads to be consider for the hull supports verification are defined in [2.2] and [2.3].

3.1.4 Stress Criteria

The von Mises stresses resulting from the FEM calculations performed in agreement with [3.1.1] to [3.1.3] are to comply with the following criteria:

$\sigma_{VM} \leq 1,25\sigma_{ALL}$

where:

σ_{ALL} : Allowable stress, in MPa, equal to:

$\sigma_{ALL} = 65 / k$

k : Material factor defined in Sec 2

In addition, the FEM calculated stresses are to comply with the buckling assessment criteria as given in NR467, Pt B, Ch 9, Sec 1.

Section 7 Tests

1 General

1.1 Scope

1.1.1 Application

This Section specifies the basic provisions on surveying and testing at the firm (manufacturer) of product prototypes, products at steady production and on-board tests.

For the items not indicated in this Section the provisions of NR266, Requirements for Survey of Materials and Equipment for the Classification of Ships and Offshore Units, are to be applied.

1.2 Inspection and testing procedures

1.2.1 Mechanical (Tensile, Impact, Hardness) tests

Mechanical tests are to be carried out in accordance with the relevant requirements of NR216, Rules on Materials and Welding for the Classification of Marine Units.

1.2.2 Magnetic Particle and liquid penetrant examination

Surface crack detection of steel forging is to be carried out in compliance with the relevant requirements of NR216.

1.2.3 Ultrasonic examination

Ultrasonic examination shall be carried out in accordance with the relevant requirements of NR216.

1.2.4 Chemical composition

The determination of the chemical composition analysis of the materials is to be carried out in accordance with the relevant requirements of NR216.

1.2.5 Visual inspection and dimensional check

Visual inspection and dimensional check are to be carried out with a view to establish that the product assembly:

- complies with approved technical documentation
- is constructed in accordance with the relevant constructional requirements.

1.2.6 Hydrostatic tests

The hydrostatic tests are to be carried out in accordance with the requirements of NR467, Pt C, Ch 1, Sec 10, [20.5], as applicable, after completion of manufacture and before installation on board.

The test pressure is to be equal to 1,5 times the design pressure of the concerned system.

1.3 Type test and routine test

1.3.1 Type test

Type test is a test performed on the first or one of the first new specimen with regard to verification of the expected equipment running behaviour.

Type tests are to be performed on an equipment or system strictly identical to the one(s) intended to be fitted onboard the ship.

1.3.2 Routine test

Routine test is a test performed on each individual product or a batch of products in order to verify conformity with the approved type and/or approved documentation.

Routine tests are part of product certification as required by Sec 8.

2 Factory inspection and testing

2.1 Steering propulsion units

2.1.1 General

An inspection and checks are to be carried out with a view to establish:

- compliance of product with approved technical documentation
- compliance of product with the applicable test requirements
- satisfactory operation of the product.

The following is to be verified during the inspection (including openings-up and single disassembly if needed):

a) Technical documentation for materials the product is made of

b) Accessories being part of the equipment inspected

c) Mounting of the electrical circuit of the product

d) Structural design of the product

e) Strength of connecting and fastening units, current-carrying parts, welded, screwed and other structural and contact joints

f) Provision of anti-corrosion coatings

g) Provision of necessary markings and inscriptions

h) Contact and protective terminations of cables and wires

i) Arrangements ensuring electrical safety (protective earthing, interlocks, etc.).

2.1.2 Scope of tests

The tests required after assembly of the main steering propulsion units are listed in Tab 1.

Table 1 : Tests to be carried out after assembly of the steering propulsion units

No.	Test	Type test	Routine Test
1	Measurements of insulation resistance	X	X
2	Running test	X	X
3	Tests of electrical insulating strength	X	X
4	Tests for compliance with operational conditions (mechanical and environmental)	X	
5	Tests of protective enclosures	X	X
6	Heat tests	X	
7	Overcurrent tests	X	X
8	Test of radio interference level	X	
9	Torque overload tests	X	
10	Stalling tests	X	
11	Overspeed tests	X	
12	Test of protection and alarm systems	X	X

2.2 Forging for propeller shaft, propeller blade bolts and propeller's blade

2.2.1 General

Before acceptance, all forgings are to be presented to the Surveyor for visual examination. Where applicable, this is to include the examination of internal surfaces and bores.

2.2.2 Scope of tests

The tests required for forgings, propeller shaft, propeller blade bolts and propeller's blade are listed in Tab 2.

Table 2 : Tests to be carried out for forgings, propeller shaft, propeller blade bolts and propeller's blade

No.	Test	Type test **(1)**	Routine Test **(2)**
1	Magnetic Particle and liquid penetrant examination		X
2	Ultrasonic examination		X
3	Chemical composition		X
4	Tensile test		X
5	Impact test		X
6	Hardness test **(3)**		X
7	Visual inspection and dimensional check		X
8	Balancing test **(4)**		X

(1) Type test on prototype of material or test on at least the first batch of materials
(2) The reports of material routine tested are to contain the manufacturer's forging number which has been tested and the test result
(3) Hardness test is not mandatory for forgings
(4) Balancing test is required for propellers

2.3 Shafts of azimuth thruster (finished and machined propeller shaft, intermediate shafts, and thrust shaft), thrust bearing unit, journal bearing, propeller hub (bosses), hull sealing of propulsion unit, propeller shaft seals

2.3.1 General

During external examination of the finished shafts, it is necessary to make certain that:

- treated surfaces for mounting the working elements of the pumps, half-couplings and linings, interference fits and working journals comply with the technical documentation;
- concentricity of the surfaces, run-out of the half-coupling end face have been checked by approved method.

2.3.2 Scope of tests

The tests to be carried out for shafts of azimuth thruster (finished and machined propeller shaft and thrust shaft), thrust bearing unit, journal bearing, propeller hub (bosses), hull sealing of propulsion unit, propeller shaft seals are listed in Tab 3.

Table 3 : Tests to be carried out for shafts of azimuth thruster (finished and machined propeller shaft, intermediate shafts, and thrust shaft), thrust bearing unit, journal bearing, propeller hub (bosses), hull sealing of propulsion unit, propeller shaft seals

No.	Test	Type test	Routine Test
1	Visual examination and dimensional check	X	X
2	Chemical composition (except for propeller shaft seals)	X	X
3	Mechanical tests	X	X
4	Hydraulic tests (for propeller shaft seals)	X	X

2.4 Castings (thruster body, upper part and propeller end)

2.4.1 General

All castings are to be free from surface or internal defects, which would be prejudicial to their proper application in service. The surface finish is to be in accordance with good practice and any specific requirements of the approved plan.

All castings are to be cleaned and adequately prepared for examination; suitable methods include pickling, caustic cleaning, wire brushing, local grinding, shot or sand blasting. The surfaces are not to be hammered, peened or treated in any way, which may obscure defects.

2.4.2 Scope of tests

Tests required for castings (thruster body, upper part and propeller end) are listed in Tab 4.

Table 4 : Tests required for castings (thruster body, upper part and propeller end)

No.	Test	Type test	Routine Test
1	Visual examination and dimensional check	X	X
2	Chemical composition (except for propeller shaft seals)	X	X
3	Mechanical tests	X	X
4	Hydraulic tests for the hull sealing of propulsion unit	X	X

2.5 Heat exchangers, cooling drive for azimuth thruster, cooling air unit, swivel, oil treatment units, hydraulic power units

2.5.1 Scope of tests

The tests required for heat exchangers, cooling drive, cooling air unit, swivel, oil treatment units, and hydraulic power units are listed in Tab 5.

Table 5 : Tests for heat exchangers, cooling drive, cooling air unit, swivel, oil treatment units, and hydraulic power units

No.	Test	Type test	Routine Test
1	Hydraulic tests	X	X
2	Functional tests	X	X

2.6 Mounting block, covers, and upper thruster body ring

2.6.1 General

The results of component measurement and fixing measurements submitted in the process of manufacturing the components and during installation thereof, are to encompass all measuring points specified by the working documentation and instructions on installation and operation of the machinery.

The control of the measurement results is to be exercised at random with the aim to determine the compliance of the design of the supervised item, its dimensions and inspection methods with the requirements of the working drawings

2.6.2 Scope of tests

The tests required for mounting block, covers and upper thruster body ring are listed in Tab 6.

Table 6 : Tests for mounting block, covers and upper thruster body ring

No.	Test	Type test	Routine Test
1	Visual check	X	X
2	Dimensional check	X	X

2.7 Pumps except hydraulic gear pumps

2.7.1 General

Provision is to be made to prevent the pumped fluid from penetration to the bearings. However, this does not apply to the pumps where the pumped fluid is employed for lubrication of bearings. The pump glands arranged on the suction side are recommended to be fitted with hydraulic seals.

2.7.2 Scope of tests

The tests required for pumps (except hydraulic gear pumps) are listed in Tab 7.

Table 7 : Tests required for pumps (except hydraulic gear pumps)

No.	Test	Type test	Routine Test
1	Hydrostatic test	X	X
2	Functional test	X	X
3	Safety devices		X

2.8 Hydraulic gear pump and pinions

2.8.1 Shafts and screws

During external examination of the finished shafts and screws, it is necessary to make sure that:

a) Treated surfaces for mounting, heat treatment comply with the technical documentation;

b) Concentricity of the surfaces, screw surface and teeth profiles, heat treatment of the working surfaces have been checked by approved methods.

2.8.2 Screw housings

During external examination of the finished screw housings, it is necessary to make sure that:

a) Treated surfaces for mountings, centre-to-centre distances of the bores for the screws comply with the technical documentation;

b) Concentricity of the bores, perpendicular position of their generatrices to the end faces, parallelism of the bore axes one to another and to the common axis and the centre-to-centre distances of the bores have been checked by approved methods.

2.8.3 Pinions

During external examination of the finished pinions, it is necessary to make sure that:

a) Treated surfaces for mounting and heat treatment comply with the technical documentation;

b) Tooth shape, toothing contact and heat treatment have been checked by approved methods.

2.8.4 Tests to be carried out

The tests required for hydraulic gear pump and pinions are listed in Tab 8.

Table 8 : Tests required for hydraulic gear pump and pinion

No.	Test	Type test	Routine Test
1	Functional test	X	X
2	Safety devices	X	X
3	Hydrostatic test	X	X
4	Running test (100 hours, see IACS UR M42.14.2)	X	

2.9 Gear rim and pinion

2.9.1 General

During external examination of the finished pinions, it is necessary to make sure that:

a) Treated surfaces for mounting and heat treatment comply with the technical documentation;

b) Tooth shape, toothing contact and heat treatment have been checked by approved methods.

2.9.2 Scope of tests

The tests required for gear rim and pinions are given in Tab 9.

2.9.3 Ship series

In case of ship series, the gear rim and pinion shaft intended for the first ship can be considered as a prototype.

Table 9 : Tests required for gear rim and pinion

No.	Test	Type test	Routine Test
1	Visual check	X	
2	Dimensional check	X	
3	Material check (except for propeller shaft seals)	X	
4	Static balancing test		X
5	Tooth meshing test (See Note)		X
6	Contact pattern check after the assembly		X

Note 1: The tooth meshing test is to be performed in the presence of the Surveyor. This test is to be carried out at a load sufficient to ensure tooth contact, with the journals located in the bearings according to the normal running conditions.
Before the test, the tooth surface is to be coated with a thin layer of suitable coloured compound. The results of such test are to demonstrate that the tooth contact is adequately distributed on the length of the teeth. Strong contact marks at the end of the teeth are not acceptable.

2.10 Steering gear bearings: roller bearing slewing ring, hull sealing of steering propulsion unit, connecting bolts of hull, shafts and steering gear rings

2.10.1 Scope of tests

The tests required for the steering gear bearings, roller bearing slewing ring, hull sealing of propulsion unit, connecting bolts of hull, shafts and steering gear rings are listed in Tab 10.

Table 10 : Tests for the steering gear bearings, roller bearing slewing ring, hull sealing of steering propulsion unit, connecting bolts of hull, shafts and steering gear rings

No.	Test	Type test	Routine Test
1	Visual check	X	X
2	Dimensional check	X	X
3	Material check (except for propeller shaft seals)	X	
4	Chemical composition	X	
5	Mechanical test	X	

2.11 Testing of electrical propulsion motors

2.11.1 General

All electric propulsion motors are to be tested by the manufacturer. All tests are to be carried out according to IEC 60092- 301. All propulsion motors are to be type approved or case-by-case approved and surveyed by the Society during testing and, if appropriate, during manufacturing. Tested motors are to be individually certified by the Society.

Where the test procedure is not specified, the requirements of IEC 60034-1 apply.

This section involves the basic provisions on surveying and testing at the firm (manufacturer) of product prototypes and products at steady production.

Type tests are to be carried out on a prototype propulsion motor or on the first batch of motors, and routine tests carried out on subsequent motors in accordance with Tab 11.

Table 11 : Tests to be carried out on electric propulsion motors

No.	Test	Type test **(1)**	Routine Test **(2)**
1	Assembly tests	X	X
2	Examination of the technical documentation visual inspection in compliance with design drawings	X	X
3	Insulation resistance measurement (stator and rotor windings)	X	X
4	Winding resistance measurement (stator and rotor)	X	X
5	Rated load test and temperature rise measurement	X	
6	Overtorque test	X	
7	Overspeed test	X	
8	Dielectric strength test (stator and rotor windings)	X	X
9	No load test	X	X
10	Verification of degree of protection	X	
11	Verification of bearings	X	X

(1) Type test on prototype machine or test on at least the first batch of machines.
(2) The reports of machines routine tested are to contain the manufacturer's serial number of the machine which has been type tested and the test result.

2.11.2 Assembly tests

Electrical propulsion motors developing a power of more than 1MW are to be subject to the following tests during their assembly:

a) Individual coils dielectric strength test

A high frequency high voltage test in accordance with IEC 60034-15 is to be carried out on the individual coils of high voltage motors in order to demonstrate a satisfactory withstand level of the inter-turn insulation to steep fronted switching surges.

b) Rotor winding assembly

Wound rotors are to be subjected to an individual dynamic balancing test or equivalent

c) Stator winding assembly

A dielectric test and an insulation resistance measurement is to be carried out after impregnation

d) Frame

Liquid penetrant test of 10% of the structure welds and 100% of the handling points is to be performed

2.11.3 Visual inspection

Technical documentation of electric propulsion motors are to be available for examination by the surveyor. A visual inspection and checks are carried out to ensure, as far as practicable, that the electrical propulsion motors assembly complies with the approved design drawings and technical documentation.

2.11.4 Insulation resistance measurement

Immediately after the high voltage tests the insulation resistances are to be measured using a direct current insulation tester between:

- all current carrying parts connected together and earth
- all current carrying parts of different polarity or phase, where both ends of each polarity or phase are easily accessible.

The minimum values of test voltages and corresponding insulation resistances are given in Tab 12. The insulation resistance is to be measured close to the operating temperature, or an appropriate method of calculation is to be used.

2.11.5 Winding resistance measurement

The resistances of the propulsion motors' windings are to be measured and recorded using an appropriate bridge method or voltage and current method.

Table 12 : Minimum insulation resistance

Rated voltage Un (V)	Minimum test voltage (V)	Minimum insulation resistance (MΩ)
Un ≤ 250	2 x Un	1
250 < Un ≤ 1 000	500	1
1 000 < Un ≤ 7 200	1 000	Un/1 000 + 1
500 < Un ≤ 15 000	5 000	Un/1 000 + 1

2.11.6 Rated load test and temperature rise measurements

The temperature rises are to be measured at the rated output, voltage and frequency and the duty for which the machine is rated and marked, in accordance with the testing methods specified in IEC 60034-1, or by means of the indirect test methods specified in IEC 60034-29.

The test of electrical machines for heating is to be carried out under the normal environmental conditions at an air temperature of 25 +10 °C up to a steady-state temperature.

The limits of temperature rise above ambient air temperature of 45°C for air-cooled machines are those given in Tab 13. For water-cooled machines the temperature rises are not to exceed the limits specified in IEC 60034-1 adjusted according to the temperature of the cooling water available on board the ship.

In relation to the evaluation of the temperature rise, it is necessary to consider the supplementary thermal losses induced by harmonic currents in the stator winding. Where the indirect test method is used, this estimation is to be documented. A justification based on a computer program calculation may be taken into consideration, provided that validation of such program is demonstrated by previous experience.

Table 13 : Temperature rise limits for air-cooled motors based on an ambient temperature of 45°C

No.	Part of machines	Method of measurement of temperature **(1)**	Temperature rise, in °C, by class of insulation				
			A	E	B	F	H
1	A.C. windings of motors having outputs of 5 000 kW (or kVA) or more	R ETD	55 60	— —	75 80	75 80	120 125
	A.C. windings of motors having outputs of less than 5 000 kW (or kVA)	R ETD	55 60	70 —	75 85	100 105	120 125
2	Field windings of A.C. motors having D.C. excitation other than those mentioned in item 3	T R	45 55	60 70	65 75	80 100	100 120
3	Field windings of synchronous motors with cylindrical rotors having D.C. excitation	R	—	—	85	105	130
	Low resistance field windings of more than one layer, and compensating windings	T, R **(2)**	55	70	75	95	120
	Single-layer windings with exposed bare surfaces	T, R **(2)**	60	75	85	105	130
4	Permanently short-circuited, insulated windings	T	55	70	75	95	120
5	Permanently short-circuited, uninsulated windings	The temperature rise of these parts is in no case to reach such a value that there is a risk of damage to any insulating or other material on adjacent parts					
6	Magnetic core and other parts not in contact with windings						
7	Magnetic core and other parts in contact with windings	T	55	70	75	95	120

(1) T: Measurement by the thermometer method
R: Measurement by the resistance method
ETD: Measurement by embedded temperature detectors.
(2) Temperature rise measurement is to use the resistance method R whenever practicable.

2.11.7 Overtorque test

Electric propulsion motors are to withstand the following torque overloads:

- AC induction motors: 60% in excess of the torque that corresponds to the rating, for 15 s, without stalling or abrupt change in speed (under gradual increase of torque), the voltage and frequency being maintained at their rated value.
- AC synchronous motors with salient poles: 50% in excess of the torque that corresponds to the rating, for 15 s, without falling out of synchronism, the voltage, frequency and excitation current being maintained at their rated values.

Note 1: The overtorque test can be replaced at routine test by an overcurrent test.

Note 2: For electrical propulsion motors supplied by converters, the torque overload value may be lower based on setting of the overload protection of converters.

2.11.8 Overspeed test

Electric propulsion motors, except the squirrel cage motors are to withstand an overspeed test at 1,2 times their maximum rated speed for a duration of 2 minutes. The test duration is counted off since the moment when the machine has reached its test speed. Following the test, the machine is to be thoroughly examined for any damages and deformations.

2.11.9 Dielectric strength test

New and completed propulsion motors are to withstand a dielectric test as specified in IEC 60034-1.

When it is necessary to perform a high voltage test in addition to the withstand test at full voltage performed on the windings, this is to be carried out after further drying, with a test voltage of 80% of that specified in IEC 60034-1.

2.11.10 No load test

Propulsion motors are to be operated at no load and rated speed whilst being supplied at rated voltage and frequency. During the running test, the vibration level of the motor, the operation of the auxiliaries such as cooling system or bearing lubrication system, if appropriate, as well as repeated starts, stops and reverse operation are to be checked.

2.11.11 Verification of degree of protection

As specified in IEC 60034-5.

2.11.12 Verification of bearings

Upon completion of the above tests, propulsion motors which have sleeve bearings are to be opened upon request for examination by the Surveyor, to establish that the shaft is correctly seated in the bearing shells.

2.12 Testing of electrical slip rings

2.12.1 General

All slip rings are to be type approved or case-by-case approved and surveyed by the Society during testing and, if appropriate, during manufacturing. Tested slip rings are to be individually certified by the Society.

Electric slip ring assemblies are to be subjected to the tests stated in Tab 14. Where the test procedure is not specified, the requirements of IEC 61439-1 and IEC 61439-2 or IEC 62271-200 apply.

Tests procedure is to be submitted to the Society for approval.

Type tests are to be carried out, unless the manufacturer can produce evidence based on previous experience indicating the satisfactory performance of such equipment.

Table 14 : Tests to be carried out on electrical slip rings

No.	Tests	Type test **(1)**	Routine test **(2)**
1	Examination of the technical documentation, as appropriate, and visual inspection including check of protection index, clearances and creepage distances **(3)**	X	X
2	Mechanical tests	X	X
3	Electric contact resistance test	X	X
4	Insulation resistance measurement	X	X
5	Dielectric strength test	X	X
6	Signal attenuation (for fibre optic swivel)	X	X
7	Functional test of auxiliaries (e.g. sensors, data transfer)	X	X
8	Endurance test	X	–
9	Temperature rise test	X	–
10	Impulse voltage withstand test	X	–
11	Short-circuit withstand test	X	–
12	Verification of degree of protection	X	–
13	Environmental tests	X	–

(1) Type test on prototype slip ring or test on at least the first batch of slip rings.
(2) The certificates of slip rings routine tested are to contain the manufacturer's serial number of the slip ring which has been type tested and the test result.
(3) A visual examination is to be made of the slip ring, as far as practicable, that it complies with technical documentation.

2.12.2 Visual inspection

An inspection and checks are carried out with a view to establish that the electrical slip ring assembly:

a) Complies with approved technical documentation

b) Maintains the prescribed degree of protection

c) Is constructed in accordance with the relevant constructional requirements, in particular as regards creepage and clearance distances

2.12.3 Mechanical tests

The electrical slip ring is to be subjected to the following mechanical tests:

a) Free rotation test on the full 360° in both clockwise and counter-clockwise. No hard point is to be encountered

b) Free rotation test on the full 360° in clockwise and counter-clockwise to measure and record the breakout torque and the steady running torque

c) Free rotation test on the full 360° in clockwise and counter-clockwise when the slip ring is inclined at an angle of 22,5° (roll) to measure and record the breakout torque and the steady running torque.

Tests are to be carried out at the specified rotation speed and the measured values are not to exceed data given by Manufacturer.

2.12.4 Electric contact resistance test

The electrical slip ring is to be subjected to a resistance test to measure and record the contact resistance across all paths on the full 360° in clockwise and counter-clockwise. Test is to be carried out at the specified rotation speed. The measured values are not to exceed the values given by the Manufacturer.

2.12.5 Insulation resistance measurement

Immediately after the high voltage test, the insulation resistance between all current carrying parts and earth (and between each polarity and the other polarities) is to be measured using a direct current insulation tester.

The minimum values of test voltages and corresponding insulation resistances are given in Tab 15.

Table 15 : Minimum insulation resistance

Rated voltage Un (V)	Minimum test voltage (V)	Minimum insulation resistance (MΩ)
Un ≤ 250	2 x Un	1
250 < Un ≤ 1 000	500	1
1 000 < Un ≤ 7 200	1 000	Un/1 000 + 1
500 < Un ≤ 15 000	5 000	Un/1 000 + 1

2.12.6 Dielectric strength test

Slip rings are to be subjected to a high voltage test:

a) Between all live parts connected together and exposed conductive parts

b) Between each polarity and, all other polarities and exposed conductive parts connected together

The test voltage is to be as given in Tab 16 to Tab 18. The test voltage is to be applied for at least 60 seconds and is to have a substantially sinusoidal waveform and a frequency equal to the rated frequency of the assembly with a tolerance of ±25%.

Table 16 : Test voltages for main circuits of low voltage slip rings

Rated insulation voltage Ui (V)	Test voltage a.c. (r.m.s) (V)
Ui ≤ 60	1 000
60 < Ui ≤ 300	1 500
300 < Ui ≤ 690	1 890
690 < Ui ≤ 800	2 000
800 < Ui ≤ 1 000	2 200

Table 17 : Test voltages for auxiliary and control circuits of low voltage slip rings

Rated insulation voltage Ui (V)	Test voltage a.c. (r.m.s) (V)
Ui ≤ 12	250
12 < Ui ≤ 60	500
Ui < 60	See Tab 16

Table 18 : Test voltages for high voltage slip rings

Nominal voltage (kV)	Highest voltage for equipment (kV)	Test voltage a.c. (r.m.s) (kV)
3 - 3,3	3,6	10
6 - 6,6	7,2	20
10 - 11	12	28
15	17,5	38

2.12.7 Signal attenuation test (for fibre optic swivel)

The fibre optic swivel is to be subjected to signal attenuation test to measure and record the signal attenuation across all paths on the full 360° in clockwise and counter-clock-wise. Test is to be carried out at the specified rotation speed.

The measured values are not to exceed the values given by the Manufacturer.

2.12.8 Endurance test

An endurance test at a rotation speed of 1 rpm is to be carried out as follows:

a) 100 rotations with 10% rated current (In);

b) 100 rotations with 90% rated current (In);

c) 1 rotation with 150% rated current (In);

d) 100 rotation without current;

After the test the slip rings electric contact resistance measurement is to be repeated.

2.12.9 Temperature rise test

A temperature rise test is to be carried out to measure the temperature rise of the active parts and the slip ring enclosure when all paths are loaded at their rated current and with normal cooling capability. Test may be performed in static condition. The measured values are not to exceed the values given by the Manufacturer.

2.12.10 Short-circuit withstand test

A short-circuit withstand test is to be performed to demonstrate that active parts with their supports are mechanically dimensioned and fixed to withstand the stresses caused by sudden short-circuit current without damage. The test value is not to be lower than the maximum prospective value of the short-circuit current on the slip ring. A calculation note is admitted as alternative to the test.

2.12.11 Impulse voltage withstand test

A impulse voltage withstand test in accordance with requirements of IEC 61439-1 or IEC 62271-200 is to be carried out in case of the air clearance does not satisfy the requirements of Sec 4, [3.2.6] and Sec 4, [3.2.7].

2.12.12 Verification of the degree of protection As specified in IEC 60529.

2.12.13 Environmental tests

To validate that the slip rings assemblies are designed and constructed to operate satisfactorily under the environmental conditions expected on board they are to be subjected to the following environmental tests. During testing the slip rings are to be in normal operating conditions. For large slip rings, a case by case evaluation based on calculations and/or documented measures is admitted as alternative to the vibration test and shock test.

a) Ambient temperature test in compliance with IEC 60068-2-2, at temperatures specified in Sec 1, [1.5.1]

b) Humidity resistance test in compliance with IEC 60068-2-30 Test Db, at a relative humidity specified in Sec 1, [1.5.2]

c) Inclination test at inclination angles specified in Sec 1, [1.5.4]

d) Vibration test in compliance with IEC 60068-2-6 Test Fc, at vibration levels specified in Sec 1, [1.5.5] for podded thrusters.

e) Shocks test in compliance with IEC 60068-2-27, with the following test parameters: acceleration 5,0 g, duration 6-30 ms, number of shocks in each position 20, 40-80 shocks per min.

2.13 Monitoring, alarm and safety systems

2.13.1 General

Acceptance tests are to be carried out at the manufacturer's facilities to demonstrate that the monitoring, alarm and safety systems have been correctly installed and are in good working order before the shipment of the equipment. This applies to the thruster and to its auxiliary systems, in particular to the hydraulic, cooling, lubricating, draining and ventilation systems.

Acceptance tests refer to hardware and software tests as applicable. Tests are to be carried out in accordance with approved test programs. The test program is to specify in detail how the various functions are to be tested and what are the expected results. Test results are to be recorded.

2.13.2 Hardware acceptance tests

Hardware acceptance tests include, where applicable:

a) Visual examinations

b) Operational tests and, in particular:

   1) tests of all alarm, control and safety functions

   2) verification of the required performance (range, calibration, repeatability, etc.) for analogue sensors

   3) verification of the required performance (range, set points, etc.) for on/off sensors

   4) verification of the required performance (range, response time, etc.) for actuators

   5) verification of the required performance (full scale, etc.) for indicating instruments

c) Failure tests (as realistically as possible)

Additional tests may be required by the Society.

2.13.3 Software acceptance tests

Software acceptance tests of computer-based systems are to be carried out according to the relevant requirements of NR467, Pt C, Ch 3, Sec 3.

3 On-board tests

3.1 General

3.1.1 Application

After installation on-board, the azimuth propulsion systems are to be tested and inspected according to the requirements given in [3.2] and [3.3].

3.2 Shipboard and mooring trials

3.2.1 Scope of tests

The mooring trials of the azimuth propulsion systems are to include the following tests:

a) Smooth operation of the manually controlled turning mechanism

b) Operation of the control, alarm, protection and blocking systems

c) Operation of the steering-propulsion unit. During this test, the following is to be checked:
   - operation of the reduction gear and seals
   - turning time
   - pressure and temperature of the systems.

3.3 Sea trials

3.3.1 Ship manoeuvrability tests

Ship manoeuvrability tests, such as according to IMO Resolution MSC.137(76) on Standards for ship manoeuvrability, are to be carried out with steering angles not exceeding the declared steering angle limits (See Sec 1, [1.5.3]).

3.3.2 Azimuth propulsion system tests

The sea trials of the azimuth propulsion systems are to include the following tests:

a) Verification of the capacity and safety functions for the azimuth propulsion system,

b) Function testing of the control system, including shifting between different operation modes,

c) Testing of the alarms and safety functions.

3.3.3 Leak inspection

Upon completion of the sea trials, watertightness of the sealing arrangements are to be visually inspected to confirm the absence of leak.

Section 8 Certification

1 General

1.1 Minimum requirements

1.1.1 Certification scheme

The certification of products used in the azimuth propulsion system is to comply with the relevant requirements of NR216 and NR320, taking into account the provisions of Tab 1 where the following definitions apply:

TA : Type Approval Certificate: Document certifying compliance of products with the relevant Society's Rules based on the design assessment, and type testing results, as relevant.

DA : Design Approval: The design approval, DA, corresponds to a review of technical documentation for a specific unit in compliance with the relevant requirements of the Society's Rules and/or the agreed requirements. DA is a step within the Classification process which is followed by inspection at works, as relevant, and by construction survey activities, as applicable to the concerned ship's part or product.

C : Product Certificate: Document certifying compliance of materials and/or products with the relevant Society's Rules.

W : Work's certificate issued by the manufacturer: Certificate of Conformity stating the results of the tests performed.

h : Hydraulic pressure test (or equivalent)

This Section provides minimum requirements for the certification of products.

In case of products are operated in unusual conditions not covered by this Rule Note, the Society may specify additional requirements. Such requirements could cover chemical composition, mechanical properties, testing, etc.

Table 1 : Elements certification

No.	Item	Product certification			
		Design assessment / Approval	Raw material certificate	Examination and testing	Product certificate
GENERAL					
1	Azimuth thruster body	DA	C	X	C
2	Thruster body casting (upper part)	DA	C	X	C
3	Thruster body casting (Thruster body propeller end)	DA	C	X	C
4	Mounting block with cover of azimuth thruster	DA	C	X	C
5	Steering control unit of the azimuth thrusters	DA	W	X	C
MACHINERY EQUIPMENT					
6	Main steering propulsion units without remote and local back-up control systems	DA	W	X	C
7	Shafts of azimuth thruster: finished and machined (propeller shaft, intermediate shafts and thrust shaft)	DA	C	X	C
8	Forging for propeller shaft	DA	C	X	C
9	Propeller shaft seals	DA or TA	C / W **(1)**	X	C / W **(1)**
10	Thrust bearing unit	TA	W	X	W
11	Journal bearing	TA	W	X	W
12	Propeller Hub (bosses)	DA	W	X	C
13	Propeller blade bolts	DA	W	X	C
14	Propeller's blade	DA	C	X	C
15	Cooling air units	DA	W	X	C
16	Heat Exchanger	DA	C / W **(2)**	Xh	C
17	Swivel (machineries of hydraulic steering system)	DA	C	X	C
18	Oil treatment units	DA	W	X	C
19	Steering gear unit	DA	C	X	C

(1) As per conditions set in TA

(2) For class 1 'mass produced" heat exchangers: materials certificate C may be waived, and materials certificate W accepted at the Society's discretion.

No.	Item	Product certification			
		Design assessment / Approval	Raw material certificate	Examination and testing	Product certificate
20	Upper thruster body ring	DA	W	X	C
21	Hydraulic power units	DA	W	X	C
22	Pump except hydraulic gear pumps				
	1- When belonging to a class I piping system	DA	C	Xh	C
	2- When belonging to a class II piping system		W	Xh	C
	3- When belonging to a class IIII piping system if design pressure exceeds 0,35 MPa		W	Xh	C
	4- When belonging to other class III piping system			Xh	W
23	Flexible hoses for azimuth thruster's HPU hydraulic, cooling and lubricating systems	TA	W	Xh	C
24	Hydraulic gear pump	TA or DA	C	Xh	C
25	Gear rim	DA	W	X	C
26	Pinion shafts	DA	C	X	C
27	Steering gear bearing: roller bearing slewing ring	DA	W	X	C
28	Hull sealing of propulsion unit	TA	W	Xh	W
29	Connecting bolts of hull, shafts and steering gear rings	DA	W	X	C
ELECTRICAL EQUIPMENT					
30	Azimuth thruster electrical propulsion motor				
	1- For P≥100kW	DA	C	X	C
	2- For P<100kW	DA	W	X	C
31	Housings for electrical items (enclosure)	Not Applicable	W	X	W
32	Electric cable	TA		X	C / W **(1)**
33	Slip rings devices for podded azimuth propulsion	DA	W	X	C
34	Auxiliary sensors (leak detection, temperature sensor, etc,...)	TA	W	X	W
35	Variable frequency drives (VFD):				
	1- with power equal or more than 50kVA	TA	W	X	C
	2- with power less than 50kVA	TA	W	X	W
36	Electric motors with power of 100 kW and over	DA	W	X	C
37	Electric motors with power less than 100 kW	TA			W
38	Low voltage (LV) circuit-breaker	TA			W
39	Low voltage (LV) contactor	TA			W
40	Low voltage (LV) thermal protection	TA			W
41	Low voltage (LV) motor starter	TA			W
42	Automation components • Programmable Logic Controller (I/O rack) • HMI (display/screen) • Network switch • Power supplies	TA			W
43	Switchboards	DA			C
44	High voltage circuit breakers	TA			C
45	High voltage contactors	TA			C
46	High voltage protective devices	TA			C
47	Batteries and charger	TA			C
48	Propulsion remote control system	TA		X	C / W **(1)**
49	Control and monitoring systems	TA		X	C / W **(1)**

(1) As per conditions set in TA
(2) For class 1 'mass produced" heat exchangers: materials certificate C may be waived, and materials certificate W accepted at the Society's discretion.

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