RULES
FOR THE CLASSIFICATION AND CONSTRUCTION OF SMALL SEA-GOING SHIPS

PART VI
MACHINERY AND PIPING SYSTEMS

July
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GDAŃSK

RULES FOR THE CLASSIFICATION AND CONSTRUCTION OF SMALL SEA-GOING SHIPS developed and edited by Polski Rejestr Statków S.A., hereinafter referred to as PRS, consist of the following Parts:

Part I – Classification Regulations
Part II – Hull
Part III – Hull Equipment
Part IV – Stability and Freeboard
Part V – Fire Protection
Part VI – Machinery and Piping Systems
Part VII – Electrical Installations and Control Systems,

whereas the materials and welding shall fulfil the requirements specified in *Part IX – Materials and Welding*, of the *Rules for Classification and Construction of Sea-going Ships*.

Part VI – Machinery and Piping Systems – July 2023 was approved by PRS Executive Board on 29 June 2023 and enters into force on 1 July 2023.

The requirements of this *Part VI* apply to the existing ships to the full extent to new ships.

As for the existing ships, the requirements of this *Part IV* apply to an extent resulting from the provisions of *Part I – Classification Regulations*.

The requirements of *Part VI – Machinery and Piping Systems* are extended by the following Rules and Publications:

Rules for Statutory Survey of Sea-going Ships, Part IX – Environmental Protection.
Publication 7/P – Repair of Cast Copper Alloy Propellers
Publication 8/P – Calculation of Crankshafts for Diesel Engines
Publication 23/P – Pipelines Prefabrication
Publication 33/P – Air Pipe Closing Devices
Publication 53/P – Plastic Pipelines on Ships
Publication 57/P – Type Approval of Mechanical Joints
Publication 69/P – Marine Diesel Engines. Control of Nitrogen Oxides Emission
Publication 100/P – Safety requirements for sea-going passenger ships and high-speed passenger craft engaged on domestic voyages.

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1 GENERAL PROVISIONS

1.1 Application

1.1.1 This *Part VI – Machinery and Piping Systems* applies to the machinery spaces and their equipment, shafting, propellers, machinery and ship piping systems as well as to special piping systems related to the ship function.

1.1.2 This *Part VI* is not applicable to steam boilers, steam generators, water heating system boilers, as well as thermal oil boilers and related installations.

Where boilers are fitted on board the small sea-going ships, construction, arrangement and such boilers and related installations are subject to PRS consideration in each particular case.

1.1.3 The requirements specified in chapters 1 to 24 are rudimentary for all types of ships assigned the main symbol of class of a ship built under PRS survey.

Chapters 25 to 29 contain additional requirements for ships assigned an additional mark in the symbol of class specified in sub-chapter 3.4 of *Part I – Classification Regulations*.

1.2 Definitions and Explanations

Definitions and explanations of the general terminology used in the *Rules for the Classification and Construction of Small Sea-going Ships* (hereinafter referred to as the *Rules*) are contained in *Part I – Classification Regulations*.

For the purposes of *Part VI*, the following additional definitions have been adopted:

Auxiliary machinery – machinery providing for the operation of main engines, supply of the ship with electric and other power, as well as for the operation of the shipboard systems and arrangements.

Design pressure – pressure not lower than the opening pressure of safety valves or other protecting devices.

Engine room – machinery space containing main engines.

Escape route – way from the lowermost level of the machinery space to the exit from that space.

Exit – opening in a bulkhead, deck or shell plating provided with means for closing and intended for the passage of persons.

Local control station – position fitted with the operating controls, instrumentation and – where necessary – means of communication, located in close vicinity to or directly on the machine.

Machinery spaces – spaces (including trunks to such spaces) containing main engines, generators and auxiliary machinery.

Main engines – machinery intended for the ship propulsion.

Rated power – the power, defined by the engine manufacturer, developed for unlimited time at the ambient conditions with mechanical and thermal load not exceeding the values defined by the manufacturer, taken for the calculations required by the *Rules*.

Rated speed – number of revolutions per minute corresponding to the rated power.

Remote control station – position from which remote adjustment of working parameters, as well as possible remote starting and stopping of the engines and machinery is possible.

Working pressure – the highest permissible pressure during normal course of long lasting operation.

1.3 Technical Documentation

1.3.1 General Requirements

Prior to commencement of the ship/equipment construction, the below listed technical documentation shall be submitted to PRS Head Office for approval. In the case of the ships undergoing modification or reconstruction, the below listed documentation shall be submitted for approval in the scope which covers the modification or reconstruction. The documentation shall be submitted in triplicate.

1.3.2 Documentation of Machinery Spaces

Documentation concerning machinery spaces shall include:

.1 Arrangement plan of machinery and plants in machinery spaces as well as in the spaces of emergency power sources, including the means of escape;
.2 Characteristics of machinery including the data necessary for the required calculations;
.3 Diagram and specification of remote control of main machinery, including the data of remote control stations' fitting with control devices, instrumentation, warning devices, means of communication and other equipment;
.4 Drawings of seating the main engines on the foundation;

1.3.3 Documentation of I.C. Engines

1.3.3.1 For engine type approval, the following documentation of I.C. engines shall be submitted to PRS:

.1	Data for crankshaft calculation in accordance with *Publication No. 8/P – Calculation of Crankshafts for Diesel Engines*	**W**
.2	Engine transverse sectional drawing	**W**
.3	Engine longitudinal section drawing	**W**
.4	Drawing of cylinder head assembly	**W**
.5	Drawing of engine block [1)]	**W**
.6	Drawing of cylinder jacket	**W**
.7	Drawing of crankshaft with details for each number of cylinders	**Z**
.8	Drawing of crankshaft assembly for each number of cylinders	**Z**
.9	Drawing of counterweights with connecting bolts (unless integral with the crankshaft)	**Z**
.10	Drawing of torsion damper	**Z**
.11	Drawing of connecting rod	**W**
.12	Drawing of connecting rod assembly [1)]	**W**
.13	Drawing of piston assembly	**W**
.14	Drawing of camshaft drive assembly	**W**
.15	Material specifications of essential parts with detailed information on non-destructive and pressure tests	**Z**
.16	Arrangement of foundation bolts (for main engines only)	**Z**
.17	Schematic layout or other equivalent documents of starting air system on the engine [2)]	**Z**
.18	Schematic layout or other equivalent documents of fuel oil system on the engine [2)]	**Z**
.19	Schematic layout or other equivalent documents of lubricating oil system on the engine [2)]	**Z**
.20	Schematic layout or other equivalent documents of cooling water system on the engine [2)]	**Z**
.21	Schematic layout of the engine control system and safety systems [2)]	**Z**
.22	Assembly drawing of shielding and insulation of exhaust pipes	**W**
.23	Shielding of high pressure fuel pipes, assembly	**Z**
.24	Operation and service manuals [3)], [4)]	**W**

.25 Type test programme **Z**
.26 Product test programme **Z**

References:

1) Required when the engine cross-sections do not contain all the details.
2) Documentation of the complete system, if its components are supplied by the engine manufacturer.
3) One copy only.
4) Operation and service manuals shall contain maintenance requirements (for servicing and repair), including details of special tools and gauges that shall be used with their fittings/settings together with any test requirements on completion of maintenance.

Notes:

1. The documentation marked with code **Z** shall be approved by PRS.
2. The documentation marked with code **W** shall be submitted for reference; it may, however, be the subject of certain requirements by PRS.

1.3.3.2 Updated documentation of the engine type is the basis for PRS survey of the engine manufacture.

1.3.3.3 If the engine is being built under licence and the engine manufacturer does not posses a *Type Approval Certificate* for the engine, then the manufacturer shall provide the documentation in the scope specified in paragraph 1.3.2.1 with a detailed listing of the introduced changes with the reference to the approved type and design. PRS may require confirmation of presented changes by the licence holder being in possession of *Type Approval Certificate*.

1.3.4 Documentation of Shafting

Documentation of shafting shall include:
.1 General arrangement plan;
.2 Drawings of stern tube and attached parts;
.3 Drawings of shafts (propeller, intermediate and thrust), including the connections and couplings;
.4 Drawing of seating the thrust bearing on the foundation unless it is built in the main engine or main gear;
.5 Calculation of torsional vibration of main engine – propeller set, for internal combustion engines in excess of 75 kW rated power, and auxiliary engine – power receiver set, for internal combustion engines of more than 110 kW rated power.

1.3.5 Documentation of Propellers

Documentation of a typical screw propeller shall include:
.1 General drawing;
.2 Drawings of blades, boss and fastening elements (for built-up propellers and c.p. propellers);
.3 Diagrams and specifications of control systems for c.p. propellers;
.4 Drawings of essential parts of pitch control gear in the boss of c.p. propeller.

The scope of documentation of other propellers than typical screw propellers shall be specified by PRS in each particular case.

1.3.6 Documentation of Thrusters

1.3.6.1 For thruster approval, the following documentation shall be submitted to PRS:
.1 Technical description and basic technical specification **Z**
.2 Assembly drawing in cross section with dimensions **Z**

.3	Drawings of casings, shafts and gears	**Z**
.4	Drawings of the nozzle and screw propeller or other propulsion device	**Z**
.5	Drawings of pitch control device or vanes of cycloidal type propellers	**Z**
.6	Drawings of bearings, dynamic seals of the propeller shaft and rotating column	**Z**
.7	Hydraulic, electrical, and pneumatic diagrams including specification of the components	**Z**
.8	Diagrams of lubricating and cooling system, if applicable	**Z**
.9	Diagram showing variation of the starting torque of the motor causing rotation of propeller column	**W**
.10	Material specification for all essential parts specified in .3, .4 and .5 including the particulars concerning non-destructive tests, pressure tests as well as special manufacturing procedures	**Z**
.11	Torsional vibrations calculations	**Z**
.12	Calculations of gears and roller bearings	**W**
.13	Operation and Service Manual	**W**
.14	Type Test Programme [1)]	**Z**
.15	Product Test Programme [1)]	**Z**

References:

1) Test programmes shall include acceptance criteria. In the case of production of a single unit, a separate test program for type and product is not required.

Notes:

1. Documentation marked with code **Z** is subject to PRS approval in each particular case.
2. Documentation marked with code **W** shall be submitted for reference; it may, however, be subject to special requirements by PRS.

1.3.6.2 Updated documentation of thruster type is the basis for PRS survey of the thruster manufacture.

1.3.6.3 If the thruster is being built under licence and the thruster manufacturer does not posses a *Type Approval Certificate* for the engine, then the manufacturer shall provide the documentation in the scope specified in paragraph 1.3.6.1 with a detailed listing of the introduced changes with the reference to the approved type and design. PRS may require confirmation of presented changes by the licence holder being in possession of *Type Approval Certificate*.

1.3.7 Documentation of Machinery

Documentation of machinery including gears, clutches and all auxiliary and deck machinery shall include:

.1	Technical description and basic technical specification	**W**
.2	General arrangement with cross section and dimensional data	**W**
.3	Drawings of foundations, crankcases, columns and casings showing all details and welding procedures	**W/Z**
.4	Drawings of cylinder heads and cylinder liners	**W**
.5	Drawings of connecting rods and pistons	**W**
.6	Drawings of rotors of blowers and compressors	**W**
.7	Drawings of piston rods, connecting rods assemblies and pistons	**Z**
.8	Drawings of pinions and toothed gear wheels (see also 6.2.1.2)	**Z**
.9	Drawings of disengaging and flexible couplings (see also 6.3.1.2).	**Z**
.10	Drawings of the main gear unit thrust bearing, unless built-in	**Z**
.11	Drawings of torsional vibrations dampers	**Z**
.12	Diagrams of control, alarm and safety systems within the machinery installation	**Z**

.13 Drawings of the following: fuel oil, lubricating oil, cooling water and hydraulic piping within the particular machinery component – together with information on flexible joints applied **Z**
.14 Thermal insulation drawings, including exhaust pipes **W**
.15 Drawings of foundations of the main machinery, gears, steering gears, windlasses, mooring and towing winches **Z**
.16 Material specification for essential parts with all details of non-destructive testing, pressure testing and special technologies used during manufacturing **Z**
.17 Test programme [1)] **Z**

References:

1) Type Test Programme and Product Test Programme shall be provided where applicable.

Notes:

1. Documentation marked with code **Z** is subject to PRS approval.
2. Documentation marked with code **W** shall be submitted for reference; it may, however, be subject of certain requirements by PRS.
3. In case of the documentation marked with code **W/Z**, the first letter applies to the cast structure while the second applies to the welded structure.

1.3.8 Documentation of Pressure Vessels and Heat Exchangers

The documentation for boilers, pressure vessels and heat exchangers shall include:

.1 Design drawings of the boiler drums, casings of heat exchangers and pressure vessels with all data needed for checking the dimensions defined in *Part VI* and arrangement of all welds with dimensions;
.2 Drawings of other parts of boilers, pressure vessels and heat exchangers which are subject to approval except supercharging air coolers whose dimensions are specified in this *Part VI*;
.3 Arrangement of valves and fittings including their specification;
.4 Safety valves, their characteristics and data for calculation of their size;
.5 Material specification with the particulars concerning welding consumables;
.6 Welding and heat treatment procedures;
.7 Test programme.

1.3.9 Documentation of Piping Systems

Documentation of piping systems shall include:

.1 Diagram of gravity overboard drain system;
.2 Diagram of bilge system [1)];
.3 Diagram of oil residues system [1)];
.4 Diagram of ballast system;
.5 Diagrams of air, overflow and sounding pipes;
.6 Diagram of exhaust gas system including drawings of silencers and spark arresters;
.7 Diagrams of ventilation and air conditioning systems;
.8 Diagrams of fuel oil systems;
.9 Diagrams of lubricating oil systems;
.10 Diagrams of cooling water systems;
.11 Diagram of compressed air system;
.12 Diagrams of diagram of pipelines for heating and blowing through bottom and side sea chests, heating side valves and fittings, heating liquids in tanks;
.13 Drawing of bottom and side sea chests fittings;
.14 Diagram of sanitary system [1)];
.15 Diagram of liquefied gas system;
.16 Diagrams of hydraulic systems driving machinery and equipment;

.17 Diagrams of special systems related to the ship function.

References:

1) Requirements concerning the system which result from the marine environment are specified in the *Rules for Statutory Survey of Sea-going Ships, Part IX – Environmental Protection*.

1.4 Scope of Survey

1.4.1 General provisions concerning the survey of production and building of ship machinery and systems covered by the *Part VI* requirements are specified in *Part I – Classification Regulations*.

1.4.2 Systems, machinery and equipment, whose documentation is subject to consideration and approval, are surveyed during the ship construction or modification.

1.4.3 Subject to the survey to be exercised by PRS in the process of manufacture are those products whose documentation is subject to approval, except for the fans which are not required to be explosion proof and for the hand-operated machinery.

Exempted from survey in the process of their manufacture are also compressed gas bottles produced in accordance with the relevant standards and under the survey of a competent technical inspection body recognised by PRS.

1.4.4 The following essential parts of the products are subject to survey in the process of manufacture for compliance with the approved documentation:

.1 Internal combustion engines:
- crankshafts [M)];
- pistons;
- connecting rods with bearing covers [M)];
- cylinder blocks and liners [M1)];
- cylinder covers [M1)];
- steel gear wheels for camshaft drive.

.2 Shafts and shafting components:
- thrust, intermediate and propeller shafts (steel forgings or rolled steel). Material for shaft of more than 300 kg in mass shall be subjected to ultrasonic testing;
- coupling flanges together with screws (steel forgings or rolled steel);
- tail-shaft liners (copper alloy);
- stern glands (cast steel, steel plates, nodular cast iron, grey cast iron);
- separate thrust bearing casings (cast steel, steel plates, nodular cast iron, grey cast iron).

The possibility for particular components' construction of other materials than those specified above is subject to PRS consideration in each particular case. The materials shall fulfil the requirements specified in the *Rules for the Construction and Classification of Sea-going Ships, Part IX – Materials and Welding*.

Spare parts are subject to PRS survey in the process of their manufacture in the same scope as the basic components.

.3 Propellers and their components
- Solid propellers, vanes and bosses of and built-up propellers (cast steel, copper alloy);
- Propeller blade fastening parts, shaft nuts (cast steel, stainless steel).
- The possibility for particular components' construction of other materials than those specified above is subject to PRS consideration in each particular case. The materials shall fulfil the requirements specified in the *Rules for the Construction and Classification of Sea-going Ships, Part IX – Materials and Welding*.

.4 Thrusters:
- movable and stationary casings [M2)];
- columns [M2)];
- propeller shaft and intermediate shafts [M2)];
- propellers [M2)];
- nozzles;
- fastening elements and keys;
- piping and fittings.

.5 Gears, disengaging and flexible couplings:
- casings;
- shafts [M)];
- pinions, gear wheels, toothed-wheel rims [M)];
- torque transmitting parts of couplings: rigid parts [M)], flexible parts;
- connecting bolts.

.6 Piston-type compressors and pumps:
- crankshafts [M)];
- connecting rods;
- pistons;
- cylinder blocks and cylinder liners;
- cylinder covers.

.7 Centrifugal pumps, fans, air blowers and turboblowers:
- shafts;
- rotors;
- casings.

.8 Steering gears:
- tillers of main and emergency gear [M)];
- rudder quadrant [M)];
- rudderstock yoke [M)];
- pistons with piston rods [M)];
- cylinders [M)];
- drive shafts [M)];
- gear wheels, toothed-wheel rims [M)].

.9 Windlasses, mooring and towing winches:
- drive, intermediate and output drive shafts [M)];
- gear wheels, toothed-wheel rims;
- sprockets;
- claw clutches;
- brake bands.

.10 Hydraulic drives, screw, gear and rotary pumps:
- shafts and screw rotors;
- rods;
- pistons;
- casings, cylinders, screw pump cases;
- gear wheels.

.11 Pressure vessels and heat exchangers:
- shells, distributors, end plates, headers and covers [M1)];
- tube plates [M1)];
- tubes [M1)];

– bodies of the valves for working pressure 0.7 MPa and more and of 50 mm and more in diameter [M1)].

Notes and index explanations:

M) – material shall be PRS approved.

M1) – material for parts of pressure vessels and heat exchangers of class I and II (see sub-chapter 14.1) shall be PRS approved.

M2) – material approved by PRS. Where the drive power of auxiliary thrusters is less than 200 kW, material manufacturer's certificate is acceptable. The material shall be examined by PRS surveyor and hardness test shall be performed in his presence.

1.4.5 Pipe tubes and fittings for piping of class I and II (see paragraph 15.1.2) as well as bottom and side valves and fittings intended to be installed on the collision bulkhead and remote-controlled fittings are subject to survey in the process of their manufacture.

1.4.6 Subject to PRS survey are fitting of mechanical equipment of the machinery spaces as well as assembly and operation tests of the machinery components listed below:

.1 main engines, their gears and couplings;
.2 shafting and propellers;
.3 main thrusters;
.4 auxiliary I.C. engines of rating power above 20 kW, their gears and couplings;
.5 auxiliary thrusters of rating power above 20 kW;
.6 pressure vessels and heat exchangers;
.7 auxiliary machinery;
.8 hydraulic drive systems;
.9 control and monitoring systems;
.10 piping specified in 1.3.9.

The above-mentioned products shall be furnished with certificates issued by PRS: *Test Certificates* or *Type Approval Certificates*.

1.5 Pressure Tests

1.5.1 Components of I.C. Engines

Components of I.C. engines shall be subjected to pressure tests in accordance with Table 1.5.1.

Table 1.5.1

Item	Part name		Test pressure [MPa]
1	Cooling space of the cylinder cover [1)]		0.7 MPa
2	Cylinder liner over the whole length of the cooled space		0.7 MPa
3	Cooling space of cylinder block		1.5 *p*, however not less than 0.4 MPa
4	Exhaust valve cooling space		1.5 *p* however not less than 0.4 MPa
5	High pressure fuel injection system	Fuel injection pump body, pressure side	1.5 *p* or *p*+30 whichever less
		Fuel injection valve	1.5 *p* or *p*+30 whichever less
		Fuel injection pipes	1.5 *p* or *p*+30 whichever less
6	Turbocharger, cooling space		1.5 *p* however not less than 0.4 MPa
7	Exhaust pipe, cooling space		1.5 *p* however not less than 0.4 MPa
8	Coolers, at both sides [2)]		1.5 *p* however not less than 0.4 MPa
9	Working spaces of engine driven pumps (lubricating oil, water, fuel and bilge pumps)		1.5 *p* however not less than 0.4 MPa

Notes to Table 1.5.1:

1) For forged steel cylinder covers and forged steel piston crown, test methods other than pressure testing may be accepted, e.g. appropriate non-destructive testing and dimensional control properly recorded.
2) Supercharging air coolers may be tested only at the water side subject to PRS consent.
p – maximum working pressure for the specific part.

1.5.2 Shafting and Propellers

1.5.2.1 The following components shall be subjected to pressure tests upon completion of machining:

.1 propeller shaft liners – with pressure equal to 0.2 MPa;
.2 stern tubes – with pressure equal to 0.2 MPa.

1.5.2.2 The boss of controllable pitch propeller, after assembly of the propeller, shall be tested for tightness to an internal pressure equal to the head of working level of lubricating oil in the gravity tank. It is recommended that the blades should be put several times from one extreme position to another during the tests.

1.5.2.3 The complete boss of controllable pitch propeller, after assembly of the propeller, shall be tested for tightness to an internal pressure equal to the head of working level of lubricating oil in the gravity tank or the corresponding pressure induced by a pump.

The blades shall be put several times from one extreme position to another during the tests.

1.5.3 Machinery Components and Fittings

1.5.3.1 Upon completion of machining, but before application of protective coatings, parts of machinery and fittings working under pressure shall be tested with the hydraulic pressure determined using the following formula:

$$p_{pr} = (1.5 + 0.1K)p \quad \text{[MPa]} \tag{1.5.3.1}$$

where:

p – working pressure, [MPa];
K – coefficient determined in accordance with Table 1.5.3.1.

The test pressure, however, shall never be less than:
– the pressure with the fully opened safety valve,
– 0.4 MPa for cooled spaces and their seals, and
– 0.2 MPa in other cases.

Where either working temperature or working pressure exceeds the values specified in Table 1.6.3.1, the test pressure shall be subject to PRS consent in each particular case.

Table 1.5.3.1

Material	Working temperature up to, [°C]	120	200	250	300	350	400	430	450	475	500
Carbon and carbon-manganese steel	p, [MPa], up to	no limit	20	20	20	20	10	10	–	–	–
	K:	0	0	1	3	5	8	11	–	–	–
Molybdenum and molybdenum-chromium steel with molybdenum content 0.4% and more	p, [MPa], up to	no limit				20	20	20	20	20	20
	K:	0	0	0	0	0	1	2	3.5	6	11

Material	Working temperature up to, [°C]	120	200	250	300	350	400	430	450	475	500
Cast iron	*p*, [MPa], up to	6	6	6	6	–	–	–	–	–	–
	K:	0	2	3	4	–	–	–	–	–	–
Bronze, brass and copper	*p*, [MPa], up to	20	3.1	3.1	–	–	–	–	–	–	–
	K:	0	3.5	7	–	–	–	–	–	–	–

1.5.3.2 Pressure tests of machinery parts can be performed separately for each space, applying the test pressure determined according to the working pressure and temperature in the specific space.

1.5.3.3 Parts or assemblies of engines and machinery containing petrol products or their vapours (reduction gear casings, drip trays, etc) under hydrostatic or atmospheric pressure shall be tested for tightness applying the procedure accepted by PRS. In welded structures, only welded joints shall be tested for tightness.

1.5.4 Pressure Vessels and Heat Exchangers

1.5.4.1 Upon completion of construction and assembly, all parts of pressure vessels and heat exchangers shall be pressure tested in accordance with Table 1.5.4.1.

Table 1.5.4.1

Item	Specification	Test pressure, [MPa]	
		upon completion of construction or assembly of the strength members of shell elements, less mountings and fittings	upon completion of construction or assembly including mountings and fittings
1	Pressure vessels and heat exchangers 1)	1.5 p_w , not less than p_w + 0.1 MPa	–
2	Mountings and fittings of pressure vessels and heat exchangers	in accordance with 1.5.3.1	closure tightness test for pressure equal to 1.25 p_w

Notes to Table 1.5.4.1:

1) Pressure tests shall be performed for each side of the heat exchanger. For tests of IC engine coolers, – see Table 1.5.1.

p_w – working pressure, [MPa].

1.5.4.2 Pressure tests shall be performed upon completion of all welding operations and prior to the application of insulation and protective coatings.

1.5.4.3 Where an all-round inspection of the surfaces to be tested is difficult or impossible to perform after assembling the individual components and units, the components and units in question shall be tested prior to the assembly.

1.5.4.4 Compressed air vessels, after being installed on board the ship (with fittings and mountings), shall be tested with compressed air under the working pressure.

1.5.5 Piping and Piping Fittings

1.5.5.1 Valves and fittings installed on the piping systems of class I and II (see paragraph 15.1.2) shall be tested by hydraulic pressure in accordance with paragraph 1.5.3.1.

1.5.5.2 Valves and fittings designed for working pressures 0.1 MPa or less, as well as for underpressure shall be tested by hydraulic pressure equal to at least 0.2 MPa.

1.5.5.3 Valves and fittings installed on bottom and side sea chests as well as on external shell plating, below the maximum load waterline, shall be tested by hydraulic pressure of not less than 0.2 MPa.

Where compressed air is used for blowing-through the bottom and side sea chests, then the test pressure shall be 1.5 the value of the clearing air pressure.

1.5.5.4 Completely assembled valves and fittings shall be tested for closing tightness by hydraulic pressure equal to the design pressure.

1.5.5.5 During the testing of fittings, the requirements specified in the following standards shall be taken into account**:**

– PN-W-74017 – Armatura okrętowa – Wymagania i badania.
– PN EN 12266-1 – Industrial valves – Testing of valves – Part 1: Pressure tests, test procedures and acceptance criteria – Mandatory requirements.

1.5.5.6 Piping systems of class I and II (see paragraph 15.1.2) as well as all feed water, compressed air, thermal oil and oil fuel piping of design pressure exceeding 0.35 MPa, irrespective of their class, shall be tested by hydraulic pressure, in the presence of PRS Surveyor upon completion of fabrication and final machining, but prior to their insulation. The test pressure p_{pr} shall be determined using the following formula:

$$p_{pr} = 1,5p \quad [\text{MPa}] \tag{1.5.5.5}$$

where:

p – design pressure, [MPa]

In no case the stresses occurring during the pressure tests shall exceed 0.9 of the material yield point at the test temperature.

The requirements concerning pressure tests of liquefied gas piping and fittings are specified in sub-chapter 12.3.10.

1.5.5.7 Where, for technical reasons, a complete pressure test of pipes cannot be performed prior to installing them on the ship, the test programme for particular sections of piping, especially for assembly connections, shall be subject to PRS acceptance.

1.5.5.8 Subject to PRS consent, the pressure test may be omitted for pipes of nominal diameter less than 15 mm.

1.5.5.9 Tightness of piping shall be checked, in the presence of PRS Surveyor, during an operation test upon assembly on shipboard. This does not apply to oil fuel piping which shall be tested, in the presence of PRS Surveyor, by hydraulic pressure not less than the value determined using formula 1.5.5.5 and not less than 0.4 MPa.

1.5.5.10 Where, for technological reasons, the pipes have not been pressure tested in the workshop, the tests shall be performed upon completion of assembly on shipboard.

1.6 Materials and Welding

1.6.1 Materials intended for construction of parts of internal combustion engines, pieces of machinery and equipment covered by the requirements specified in this *Part VI* shall fulfil the relevant requirements of the *Rules for the Classification and Construction of Sea-going Ships, Part IX – Materials and Welding.*

1.6.2 In general, butt joints shall be used. The structures with fillet joints or the joints affected by bending stress are subject to PRS consideration in each particular case.

The exemplary welded joints are presented in the Annex to this *Part VI*.

1.6.3 Arrangement of the longitudinal welds in single straight line in the structures composed of several sections is subject to PRS consideration in each particular case.

1.6.4 Where high strength alloy steels (including creep resisting and heat resisting steels), cast steel or alloy cast iron are intended to be used for construction of the machinery parts, it is necessary to submit to PRS the particulars concerning chemical composition, mechanical and other special properties of the material to confirm its suitability for the production of the specific part.

1.6.5 Carbon and carbon-manganese steels may be used for parts of pressure vessels and heat exchangers with design temperatures not exceeding 400°C.

Components operating at higher temperatures may be made of the above mentioned steels provided the values taken for strength calculation, creep strength $R_z/100\ 000$ inclusive, are guaranteed by the manufacturer and comply with the standards in force.

1.6.6 Upon PRS consent, hull steels complying with Chapter 3, *Part IX – Materials and Welding* of the *Rules for the Classification and Construction of Sea-going Ships,* may be used in the construction of pressure vessels and heat exchangers operating at design temperatures below 250°C.

1.6.7 The use of alloy steels for the construction of boilers, pressure vessels and heat exchangers is subject to PRS approval in each particular case. The particulars concerning mechanical properties and creep strength of the steel and welded joints at the design temperature, technological properties, welding procedure and heat treatment shall be submitted for acceptance.

1.6.8 Parts and fittings of pressure vessels and heat exchangers of the shell diameter up to 1000 mm for working pressures up to 1.6 MPa may be manufactured of ferritic nodular cast iron in accordance with the requirements specified in Chapter 15, *Part IX – Materials and Welding* of the *Rules for Classification and Construction of Sea-going Ships*.

Other applications of cast iron are subject to PRS consideration in each particular case.

1.6.9 Copper alloys may be used for parts and fittings of pressure vessels and heat exchangers operating at the working pressures up to 1.6 MPa and design temperatures up to 250°C.

Other applications of copper alloys are subject to PRS consideration in each particular case.

1.6.10 In general, seamless pipes shall be used for parts being the subject, of this *Part* of the *Rules*. Unless any special requirements have been provided, longitudinally or spiral welded pipes may be used upon PRS acceptance in each particular case, where their equivalence with seamless pipes has been demonstrated.

1.6.11 Materials containing asbestos must not be used. This requirement does not apply to:

.1 vanes of rotary compressors and vacuum pumps;

.2 watertight joints and linings used in installations where there is a risk of fire, corrosion or intoxication at high temperatures (over 350 ºC) or high pressures (over 7 MPa);

.3 flexible parts of thermal insulation for temperatures over 1000 °C.

Note: From 1 January 2011, new installation of materials which contain asbestos is prohibited. The prohibition applies to all ships.

1.7 Heat Treatment

1.7.1 Components whose material structure may change as a result of welding or plastic forming shall be subjected to an appropriate heat treatment.

The heat treatment procedure of a welded structure shall take into account the requirements specified in *Chapter 23, Part IX – Materials and Welding* of the *Rules for Classification and Construction of Sea-going Ships*.

1.7.2 The following parts shall be subjected to normalising:

.1 cold formed parts with inner bend radius less than 9.5 times their thickness;
.2 cold formed: bottom plates of thickness exceeding 8 mm and other parts previously welded;
.3 hot formed parts when this operation was completed at the temperature lower than that required by the appropriate standard for plastic forming.

1.7.3 The following equipment shall be subjected to stress relief annealing after welding:

.1 welded structures of carbon steel with carbon content exceeding 0.25%;
.2 heat exchangers and pressure vessels belonging to Class I (see Table 14.1) made of steel with wall thickness exceeding 20 mm;
.3 heat exchangers and pressure vessels belonging to Class II (see Table 14.1) made of carbon or carbon-manganese steel with tensile strength more than 400 MPa and wall thickness exceeding 25 mm;
.4 heat exchangers and pressure vessels made of alloy steel where heat treatment is required by relevant standards;
.5 welded tube plates, the annealing being recommended to be performed prior to drilling of the holes.

1.8 Non-destructive Testing

1.8.1 The following parts of engines and machinery shall be subjected to non-destructive tests in the process of their manufacture:

.1 crankshafts forged as a single piece;
.2 connecting rods;
.3 steel piston crowns;
.4 bolts subjected to direct variable loads (bolts of the main bearings, big end bearings, and cylinder covers);
.5 steel cylinder covers;
.6 steel gear wheels of the camshaft drive;
.7 shafts of main reduction gears and tillers of mass exceeding 100 kg;
.8 gear wheels and toothed rims of mass exceeding 250 kg.

1.8.2 Ultrasonic testing, with Maker's signed certificate, is required for the parts of internal combustion engines specified in .1, .3 and .5 under 1.8.1.

1.8.3 Surface defect detecting tests by magnetic-particle inspection or liquid-penetrant inspection shall be performed in the locations indicated by PRS surveyor for the internal combustion engines' parts specified in .1 and .2 under 1.8.1.

1.8.4 PRS may require the non-destructive tests to be performed also for the parts other than mentioned above as well as their welded joints if failures are suspected.

1.8.5 Non-destructive tests shall be performed in accordance with the requirements specified in *Part IX – Materials and Welding*.

1.9 Working in Heel and Trim Conditions

The main engines and auxiliary machinery as well as machinery installations required by the *Rules* to ensure the running and safety of the ship shall be capable of operating under the conditions of:
– prolonged list – do 22.5^{o},
– prolonged trim – do 10^{o}.

1.10 Main Engine Controls

1.10.1 Starting and reversing arrangements shall be so designed and situated that each engine can be started or reversed by one person.

1.10.2 The direction of control levers or hand-wheels movement shall be clearly indicated by an arrow and relevant inscription.

1.10.3 At the local and remote control stations, moving the control levers of main engines ahead or to the right, and in the case of control hand-wheels turning them clockwise, shall correspond with the ahead running of the ship.

1.10.4 The design of main engine controls shall preclude the possibility of self-change of pre-set position.

1.11 Control Stations

1.11.1 Local control stations of the main engines shall be provided with:
.1 controls;
.2 gauges indicating the pressure in the lubricating oil system, in the starting air system as well as the temperature of cooling water and charging of the starting batteries;
.3 tachometers of crankshafts where the engine rating power is 75 kW or more;
.4 indicators of the direction and speed of the propeller shaft rotation where the engine rating power is 75 kW or more;
.5 indicator of blade position of controllable pitch propeller where the engine rating power is 75 kW or more;
.6 other instrumentation not mentioned above, as determined by the manufacturer;
.7 means of communication.

Local control station of the main engine to which direct access is impossible (e.g. the engine is enclosed in a box) may be situated on the navigation bridge.

1.11.2 In ships equipped with several main engines, reversing gears or controllable pitch propellers, a combined control station shall be provided.

1.11.3 Where remote or remote-automatic control of the main propulsion machinery is provided, the relevant requirements specified in Chapters 14 and 15 of *Part VII – Electrical Installations and Control Systems*.

1.12 Means of Communication

1.12.1 Each control station of the main engines and propellers located outside the navigating bridge shall be provided with at least two independent means of two-way communication with the navigating bridge. One of these shall be an engine room telegraph and the other – a facility for conversation.

1.12.2 In the case of remote control of the main engine from the navigating bridge, only one means may be installed for two-way communication which provides for verification of engine orders and responses. Such two-way communication may be waived if the control station on the navigating bridge is situated just over the main control station in the engine room and direct visual communication is possible.

1.12.3 Engine room telegraphs shall be fitted with signal devices clearly audible throughout the room and well distinct in tone from any other signals which may resound in the room.

Engine room telegraphs shall also be fitted with optical warning device signal ensuring visual verification of orders and responses concerning the control of the engines.

1.12.4 Where means of oral communication is provided, measures shall be taken to ensure clear audibility when the machinery is running.

1.13 Instrumentation

1.13.1 Instruments, with the exception of liquid thermometers, shall be checked and accepted by a competent administration body in accordance with the state rules in force.

1.13.2 Each pressure vessel shall be fitted with at least one pressure gauge connected to the vessel by a three-way cock or valve to enable shutting the gauge off the pressure vessel, blowing-through the pipe and connection of the control pressure gauge.

1.13.3 Pressure gauge scale range shall cover pressures applied for hydraulic tests.

Pressure range corresponding to the working pressure shall be marked on the gauge scale in green. The pressure range corresponding to the relief valve opening pressure shall be marked on the gauge scale in red.

1.13.4 Heat exchangers shall be fitted with pressure gauges and thermometers to control the flowing media. If a heat exchanger cannot be shut off the installation during its operation, the pressure gauges and thermometers may be installed beyond such a heat exchanger.

1.13.5 Piping systems shall be fitted with instrumentation necessary for monitoring of their correct operation. While specifying the number of instruments, guidance provided by manufacturers of the machinery and equipment employed in particular installation shall be taken into consideration.

1.13.6 Accuracy of tachometer indication shall be within ±2.5% of the measuring range. Where barred speed ranges for main engines are specified (see sub-chapter 8.4), they shall be clearly and durably marked on the indicating dials of all tachometers.

1.14 Machinery Automatic Equipment

Where remote or automatic control systems and systems for monitoring the operating parameters are provided, such systems shall fulfil the requirements specified in chapters 14 and 15 of *Part VII – Electrical Installations and Control Systems*.

2 MACHINERY SPACES

2.1 General Requirements

2.1.1 The width of passages from the engines' and machinery control stations and attendance positions to the escape routes shall be at least 500 mm over the whole length whereas at the control station – not less than 700 mm.

The width of passages in way of electrical distribution boards shall not be less than 600 mm.

2.1.2 The width of gangways in escape routes and the width of doors in exits shall not be less than 500 mm.

2.1.3 Each machinery space shall have at least 2 exits providing access to lifeboats and life rafts. Routes of escape shall be arranged as far apart as possible and formed by steel stairways leading to the machinery space exit doors or by steel ladders leading to a manhole.

Where the size or configuration of the machinery space makes it impracticable (e.g. the maximum distance to the door in that space is 5 m or less), one of the means of escape may be omitted.

2.1.4 If two adjacent machinery spaces intercommunicate through a door and each of them has only one means of escape through a trunk, the trunks shall be located at the opposite sides of the ship.

2.1.5 All the doors, as well as the covers of companionways and skylights through which it is possible to leave the machinery spaces, shall be capable of being opened and closed both from the inside and outside. The covers of such companionways and skylights shall bear a clear inscription prohibiting the stowage of any objects on them. Covers of the skylights which do not serve as exits shall be fitted with closing devices arranged for locking them from the outside.

2.1.6 The requirements specified in paragraphs 2.1.1 to 2.1.5 are not applicable to ships with engines to which direct access is impossible (e.g. the engine is enclosed in the box). Boxes enclosing propelling engines on open decks shall be tight up to the height specified in *Part IV – Stability and Freeboard*, their covers shall be readily and safely opened and their insulation shall be both non-combustible and non-absorbable.

2.1.7 The surfaces of machinery, equipment and pipelines, which can heat up to temperatures exceeding 220 °C shall be provided with thermal insulation. The insulation shall be made of non-combustible materials not containing asbestos (see also paragraph 1.6.11). The insulation shall be of a type and so supported that it will not crack or deteriorate when subjected to vibrations.

If exhaust gas lines or other components exposed to heating penetrate structures which are not equivalent to steel in respect of fire integrity shall be so arranged that the temperature of the insulation outer surface and the temperature of parts situated in the immediate vicinity of the insulation outer surface does not exceed 60^{o}C.

If the insulation absorbs oil, it shall be covered with a metal shield or other oil resistant material. The insulation shall also be protected against mechanical damage.

2.1.8 Each machinery space shall be fitted with mechanical ventilation in accordance with the requirements specified in Chapter 19.

2.1.9 Machinery spaces shall be fitted with proper lugs or rails to enable machinery equipment repair and maintenance.

2.2 Arrangement of Engines, Machinery and Equipment Components

2.2.1 Engines, machinery, equipment, pipes, valves and fittings shall be so arranged as to provide free access to them for attendance, repairs in case of failure, as well as dismounting and removal from the ship. The requirements specified in paragraph 2.1.1 shall also be fulfilled.

2.2.2 Oil fuel, lubricating oil and other flammable oil tanks shall not be located directly above hot surfaces such as boilers, steam pipes, exhaust manifolds, silencers and other equipment requiring thermal insulation. Means shall be provided (e.g. shields), to prevent contact with sources of ignition of any possible leakage of oil fuel, lubricating oil or other flammable oil under pressure from each pump, filter, heater or pipeline.

2.2.3 Air compressors shall be installed in such places where the contamination by flammable liquid vapours of air drawn by the compressor is as low as possible.

2.3 Foundation of Engines, Machinery and Equipment Components

2.3.1 Engines, machinery and equipment constituting the machinery installations shall be installed on strong and rigid foundations. The foundation design shall fulfil the relevant requirements specified in sub-chapter 2.17, 3.8, or 4.13 of *Part II – Hull*.

Small-size machinery and equipment may be installed on pads welded directly to the inner bottom plating or to the platform.

2.3.2 Where it is necessary to install engines or machinery on elastic pads, the pads shall be of a design approved by PRS.

2.3.3 The installation of engines on composite material pads is subject to PRS consideration in each particular case. The composite material shall be approved by PRS.

2.3.4 Main engines and their gears as well as thrust bearings shall be fixed to the foundations, entirely or in part, with fitted bolts or special stops.

Where necessary, fitted bolts may also be applied to fix auxiliary machinery to its foundations.

2.3.5 The bolts fixing main engines, auxiliary engines and machinery, and shaft bearings to their foundations as well as the bolts connecting particular segments of the shafting shall be secured against loosening.

2.3.6 Engines and machinery with horizontally arranged shafts shall be installed parallel to the ship centre line. Other orientation may be accepted, provided that the engine or machinery construction permits its operation in the conditions specified in sub-chapter 1.9, being so installed.

2.3.7 The generators' prime movers shall be installed on a common frame with the generators.

2.4 Limitation on Oil Fuel Application

Unless provided otherwise in the *Rules*, the following provisions apply to the use of oil fuel in ships:

.1 except the below listed cases, no oil fuel with a flashpoint of less than 55 ºC shall be used;

.2 for the drive of the machinery located outside the machinery spaces, oil fuel with a flashpoint of less than 43°C may be used provided that:
- oil fuel tanks (except for the double-bottom tanks) are situated outside the machinery spaces,
- on the suction pipe of the oil fuel pump a device for monitoring the oil fuel temperature is fitted,
- shut-off valves or cocks are fitted on the inlet and outlet pipes of filters are fitted,
- pipes are connected, as far as practicable, by butt welds or pipe unions.

3 INTERNAL COMBUSTION ENGINES

3.1 General Requirements

3.1.1 The requirements specified in this Chapter apply to all internal combustion engines of 55 kW rated power and more.

Application of these requirements to diesel engines below 55 kW is subject to special consideration by PRS in each particular case.

3.1.2 Rated power of main engines shall be sufficient to ensure good manoeuvrability of the ship in any conditions. The speed achieved by the ship shall not be less than 6 knots during free navigation in still water.

3.1.3 Main propulsion power shall ensure 70% of the ahead rated rpm while the ship is running astern for at least 30 min.

3.1.4 In the case of main propulsion with reverse gear or controlled pitch propeller, the running astern shall not cause overloading of the propulsion machinery.

3.1.5 Main engines and engines driving electric power generators shall be adjusted for operation in the heel and trim conditions specified in sub-chapter 1.9.

3.1.6 A type of engine shall be defined by:

.1 cylinder bore;
.2 piston stroke;
.3 fuel injection method (direct or indirect);
.4 type of fuel;
.5 working cycle (four-stroke or two-stroke);
.6 gas exchange (naturally aspirated or supercharged);
.7 maximum rated power output per cylinder, rated rotational speed and maximum effective pressure;
.8 method of pressure charging (pulse system or constant pressure system);
.9 charging air cooling system (with or without intercoolers, number of intercooling stages);
.10 cylinder arrangement (in-line or V-type).

Engines are considered to be of the same type when all the parameters and data specified above are the same and when there are no essential differences in design, components and materials.

3.1.7 Rated power[*)] shall be ensured at the ambient conditions specified in Table 3.1.7.

Table 3.1.7

Ambient conditions	For restricted service ships – marks: **I**, **II** and **III**
Atmospheric pressure	100 kPa (750 mm Hg)
Air temperature	+40 °C
Relative air humidity	50%
Sea water temperature	+25 °C

[*)] As the rated power is assumed the power, defined by the manufacturer, developed for unlimited time at the ambient conditions specified in Table 3.1.7 with mechanical and thermal load not exceeding the values defined by the manufacturer and confirmed by engine operational test.

3.1.8 In the event of turboblower failure, the main engine of a single-engine arrangement shall develop a power not less than 20 % of the rated power.

3.1.9 The requirements concerning NO_X emissions by the marine diesel engines resulting from Annex VI to *MARPOL 73/78* are contained in *Publication No. 69/P – Marine Diesel Engines. Control of Nitrogen Oxides Emissions*.

3.2 Engine Construction and Equipment

3.2.1 In general, crankcases shall not be provided with ventilation, nor any arrangements shall be fitted which could cause the inrush of outside air into the crankcase. Where forced gas exhaust from the crankcase is applied (e.g. to detect smoke inside crankcase), the vacuum shall not exceed 0.25 kPa.

Turbo-blowers may be used for crankcase ventilation provided reliable oil separators are fitted.

The diameter of crankcase venting pipes shall be as small as practicable. The ends of venting pipes shall be provided with flame-arresting fittings and arranged in the way preventing water from getting into the engine. The vent pipes shall be led to the weather deck to the places excluding the suction of vapours into accommodations and service spaces.

3.2.2 The crankshaft shall be designed for loads resulting from the engine rated power. The dimensions of the parts of monoblock or semi-built shafts shall fulfil the requirements of *PRS Publication No. 8/P – Calculation of Crankshafts for Diesel Engines*.

3.2.3 Construction of crankshafts not covered by *PRS Publication No. 8/P* or crankshafts made of nodular cast iron with $500 \leq R_m \leq 700$ MPa is subject to PRS consideration in each particular case, provided that complete strength calculations or experimental data are submitted.

3.2.4 The fillet radius at the shaft junction into flange shall not be less than 0.08 of the shaft diameter.

3.2.5 Surface hardening of the crank pins and journals shall not be applied to the fillets except that the whole shaft has been subjected to surface hardening.

3.2.6 High pressure fuel pipelines shall be made of thick-wall seamless steel pipes without welded or soldered intermediate joints.

3.2.7 All external high pressure fuel pipelines led between high pressure fuel pumps and injectors shall be protected by a shielding system which is capable of retaining fuel in case of damage to high pressure pipeline. The shielding and the high pressure fuel pipeline shall form inseparable integrity. The shielding system shall be provided with leak collecting devices and a fuel pipeline damage alarm.

3.2.8 Fuel system of engines with rated power less than 55 kW and, in special cases, with greater rated power, may fulfil the requirements specified in paragraph 3.2.9 instead of those specified in paragraph 3.2.7 subject to PRS consent in each particular case.

3.2.9 The system shielding high pressure fuel pipes on the engine may be replaced by smoke detectors properly positioned over the engine and connected to fire alarm system located on the navigating bridge. For engines without turbocharging whose exhaust manifolds are fully liquid-cooled and their location is readily visible from the navigating bridge, installation of the smoke detectors and fire alarm system may be waived.

3.2.10 For single engine propulsion systems, the arrangement of the fuel system or construction of fuel filters shall be such as to ensure uninterrupted supply of filtered fuel during cleaning of the filtering equipment. This requirement also applies to single electric generating sets whose operation is necessary for the ship's safe service (see also paragraph 20.7.1).

3.2.11 Main engines and auxiliary engines of power output more than 37 kW equipped with alarm devices giving audible and luminous alarms in the case of the lubricating system failure.

3.2.12 For single engine propulsion systems, the arrangement of the lubricating oil system or construction of lubricating oil filters shall be such as to ensure uninterrupted supply of filtered lubricating oil during cleaning of the filtering equipment (see paragraph 21.2.3.3). This requirement also applies to single electric generating sets whose operation is necessary for the ship's safe service (see also paragraph 20.7.1).

In restricted service ships assigned with mark **III** in the symbol of class fitted with the main engine of a rated power not exceeding 150 kW and the oil pump situated in the lubricating oil sump, a simplex oil filter is sufficient, provided that an alarm device is provided to give alarm of excessive pressure drop across lubricating oil filter, and in readily visible plate is fitted place on the engine to indicate the number of working hours determining the filter cartridge lifetime.

3.2.13 It is recommended that electrically started engines be equipped with engine driven generators for automatic charging the starting batteries.

3.2.14 Where the engine is started manually by crank, this shall be self-disengaging or so arranged as to ensure its safe operation. This shall be achieved without exceeding a manual force of 160 N applied to the crank turned by one man.

3.2.15 All surfaces whose temperature exceeds 220 ºC where there is a risk of fuel stream blow-out from the damaged fuel pipeline shall be adequately insulated in accordance with the requirements specified in paragraph 2.1.7.

3.2.16 Notwithstanding the provisions of paragraph 15.1.9, in the system of internal circulation of fresh water cooling the engine short segments of hose pipe connected with a pipeline by a hose clip is permitted. Pipes connected to the hose pipes shall be safely fixed to the engine, and the hoses so shaped and fixed with band pipe hangers as to preclude their disconnection due to the engine vibration.

3.2.17 Main engines shall be fitted with limiters of torque (fuel dose) preventing the engine load exceeding the rated torque.

If, according to owner's demand, it should be possible to overload the engine in operation, the maximum overload torque shall not exceed 1.1 of the rated torque. In that case the engine shall be fitted with torque limiter meeting one of the following requirements:

.1 the torque limiter shall be of two-stage type to be changed-over by the crew into the rated torque and maximum overload torque, the change-over into the overload torque being indicated on the engine control stand;

.2 the torque limiter shall be set to the maximum overload torque and a visual or audible signalling device shall be provided to give a continuous signal when the rated torque is exceeded.

3.2.18 Engines of power generating sets shall be capable of withstanding a short duration overload with torque equal to 1.1 of the rated torque, at the rated engine speed.

The engines of power generating sets shall be fitted with limiters of torque (fuel dose) preventing the engine against load exceeding 1.1 of the rated torque.

3.2.19 Each main engine shall be provided with speed governor preventing the rated speed from being exceeded by more than 15%.

Apart from the speed governor, each main engine of an output of 220 kW and more which may have a disengaged clutch or which drives a controllable pitch propeller, shall be provided with a separate overspeed governor to prevent the rated speed from being exceeded by more than 20%.

An alternative solution is subject to PRS consent in each particular case.

The device protecting against overspeed, inclusive of the dedicated driving system, shall be independent of the required rotation speed controller – governor.

3.2.20 Each engine intended to drive the main or emergency power generator shall be provided with a governor ensuring fulfilment of the following requirements:

.1 Prime movers for driving generators of the main and emergency sources of electrical power shall be fitted with a speed governor which will prevent transient frequency variations in the electrical network in excess of ± 10 % of the rated frequency with a recovery time to steady state conditions not exceeding 5 seconds, when the maximum electrical step load is switched on or off.

.2 Within the range of loads 0 – 100% of the rated load, the permanent speed after a change of load shall not be more than ±5% from the rated speed.

.3 Application of electrical load shall be possible with two load steps (see also .4 below) – so that the generator running at no load can be loaded to 50% of the rated output of the generator, followed by the remaining 50% after restoring the steady state speed. The steady state condition shall be achieved in no more than 5 seconds. The steady state conditions are those at which the fluctuation of speed variation do not exceed +1% of the declared speed at the new load.

.4 In special cases, PRS may permit the application of electrical load in more than two load steps in accordance with Fig. 3.2.20.4, provided that this has been already allowed for at the design stage and confirmed by the tests of the ship electric power plant. In that case, the power of electrical equipment switched on automatically and sequentially after the voltage recovery in bus-bars shall be taken into account.

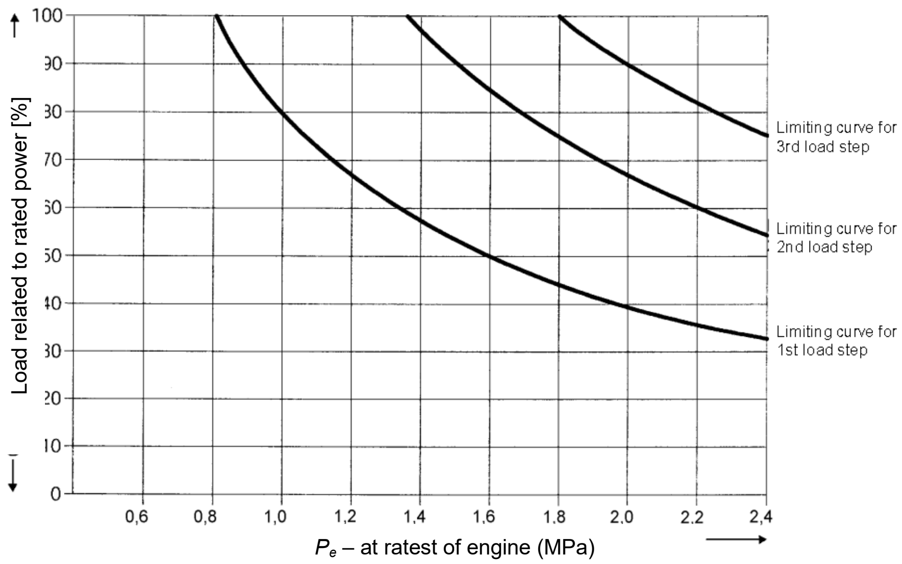


Fig. 3.2.20.4
Limiting curves for loading 4-stroke engines step-by-step from "no load" to rated power as the function of brake mean effective pressure P_e, [MPa]

Each engine driving a generator of rated power 220 kW and more shall be fitted with a separate overspeed protective device so adjusted that the speed cannot exceed the rated value by more than 15%.

3.2.21 Construction of a viscous torsional vibration damper shall enable oil sampling.

3.2.22 Engines shall be fitted with instrumentation in accordance with the requirements specified in paragraphs 1.11.1 and 1.13.6.

4 SHAFTING

4.1 General Requirements

4.1.1 The formulae given in this Chapter determine the minimum shaft diameters without allowance for subsequent machining of journals in the process of repairs.

Diameters calculated in accordance with the formulae given in sub-chapters 4.2, 4.4 and 4.5 are sufficient if additional stresses caused by torsional vibrations do not exceed the permissible values determined in Chapter 8.

4.1.2 Intermediate, thrust, and propeller shafts shall be made of steel with tensile strength R_m ranging from 400 to 800 MPa.

4.2 Intermediate Shaft

Design diameter, d_p, of intermediate shaft shall not be less than that determined in accordance with the following formula:

$$d_p = 0.95Fk\sqrt[3]{\frac{PB}{nA}} \quad [\mathrm{mm}] \tag{4.2-1}$$

where:

F – main propulsion type coefficient:

  F = 95 –for diesel engine drive where a slip clutch (hydrodynamic or electromagnetic coupling) is fitted and for electric motor drive;

  F = 100 – for other types of drive;

k – shaft design factor:

  k = 1.0 – for shafts forged together with couplings (see also 4.6.4) and for shafts with shrink-fitted couplings; for the values of k for shafts with key-fitted couplings and for shafts with keyways, holes and cuts – see sub-chapter 4.3;

P – rated power on intermediate shaft, [kW];

B – material factor determined using the formula below:

$$B = \frac{560}{R_m+160} \tag{4.2-2}$$

for intermediate and thrust shafts $1 \geq B \geq 0.5833$, and for propeller shafts $1 \geq B \geq 0.7368$;

R_m – shaft material tensile strength, [MPa];

n – rated number of intermediate shaft revolutions, [r.p.m.];

A – correction coefficient of the coaxial hole in hollow shafts determined in accordance with the formula below:

$$A = 1 - \left(\frac{d_0}{d_a}\right)^4 \tag{4.2-3}$$

where:

d_o – coaxial hole diameter, [mm];

d_a – actual outside diameter of shaft, [mm];
where $d_o \leq 0.4d_a$, $A = 1$ may be taken.

4.3 Holes and Cuts in Shafts

4.3.1 Where shafts are provided with keyways, radial holes or longitudinal cuts, the following values of coefficient k shall be taken in formula 4.2-1:

.1 $k = 1.10$ for the shaft portion with a keyway:
- after a length of not less than $0.2d_p$ from the end of the keyway where the fillet radii in the transverse section of the bottom of the keyway shall not be less than $0.0125d_p$ or 1 mm, whichever is greater, and
- over the length of 0.2dp from the cone base where the coupling flange or disk is fitted on the key; this requirement does not apply to the propeller shaft cone where the propeller is fitted;

The dimensions of the key and keyways in both shaft and coupling shall fulfil the requirements specified in paragraph 4.6.6.

.2 $k = 1.10$ for the shaft portion with a radial hole or a hollow in the middle of this portion, over the length of not less than 7 diameters of the hole, while the hole diameter shall not exceed $0.3d_p$ and its edges shall be rounded off to a radius of not less than 0.35 of the hole diameter and the inner surface shall be thoroughly ground;

.3 $k = 1.20$ for the portion of shaft with longitudinal slots over the length exceeding at least $0.25d_p$ at each side of the slot length; while the slot length shall not exceed $1.4d_p$ and the breadth – $0.2d_p$ (calculated for $k = 1$); the ends of the slots shall be rounded off to a radius equal to 0.5 of the slot breadth, the edges shall be rounded off to a radius not less than 0.35 of the same breadth, the surface of the slot shall be thoroughly ground.

4.3.2 For holes and cuts other than those specified in 4.3.1, the value of coefficient k is subject to PRS consideration in each particular case.

4.3.3 Beyond the portions specified in 3.3.1, the shaft diameter may be smoothly reduced to diameter d_p calculated for $k = 1$.

4.4 Thrust Shaft

Diameter d_{op} of the thrust shaft shall not be less than determined using formula 3.2.1-1 for k = 1.10. This applies to the shaft portion of the length not less than diameter d_{op} on both sides of the thrust collar where slide bearings are used and to shaft portion in way of the axial bearing where a roller bearing is used as a thrust bearing.

The shaft diameter outside the above-determined lengths may be smoothly reduced to the diameter of the intermediate shaft.

4.5 Propeller Shaft

4.5.1 Diameter $d_{śr}$ of the propeller shaft shall not be less than the value determined using formula 4.2-1, where:

F = 100 for all types of drive;
A = 1(i.e. $d_o \leq 0.4\ d_a$);

The value of coefficient k for the propeller shaft is:
k = 1.22 – where the propeller is fitted on the propeller shaft cone using an approved keyless shrink method or fixed to a flange integrally forged with the propeller shaft and the

propeller shaft is provided with a continuous liner or is oil lubricated and provided with an oil sealing gland of the approved type;

k = 1.26 – where the propeller is key-fitted on the propeller shaft and the propeller shaft is provided with a continuous liner or is oil lubricated and provided with an oil sealing gland of the approved type.

The above values of coefficient k apply to the portion of propeller shaft between the forward edge of the after shaft bearing and the forward face of the propeller boss or, if applicable, the forward face of the propeller shaft flange, but over a length of not less than $2.5d_{śr}$;

k = 1.15 – for the forward portion of the propeller shaft between the forward edge of the after shaft bearing and the forward edge of the forward sealing gland.

For water lubricated propeller shafts without continuous bosses, the values of k determined above shall be increased by 2%.

The diameter of propeller shaft may be smoothly reduced to the actual diameter of the intermediate shaft over the distance from the forward edge of the forward seal.

For other designs of the propeller shaft, the value of coefficient k is subject to PRS consideration in each particular case.

4.5.2 Where the propeller is joined with the propeller shaft with a key, the taper of the propeller shaft cone shall not exceed 1:12. Any exemption from the requirement is subject to PRS consent in each particular case.

If the propeller shaft or screw propeller is not made of a material resistant to the corrosive effect of sea-water, effective sealing glands shall be used to prevent penetration of sea water onto the propeller shaft cone.

Means shall be provided to secure the propeller nut against unscrewing by structural fixing it to the shaft. For shafts with a dimension not exceeding 100 mm, the nut may be fixed to the propeller boss.

The outside thread diameter of nut securing the propeller on the cone shall not be less than 0.6 of that cone base diameter.

4.5.3 The end of the keyway in the propeller shaft cone intended for the propeller shall be at a distance, from the cone base, not less than 0.2 of the propeller shaft diameter $d_{śr}$. For shafts of 100 mm in diameter and over, the end of the keyway shall be so designed that the forward end of the groove makes a gradual transition to the full shaft section. The edges of the keyway at the surface of the shaft taper shall not be sharp. The lower keyway corners shall be rounded to a radius of about 0.0125 of the propeller shaft diameter $d_{śr}$, however not less than 1.0 mm.

The dimensions of a key and keyways in both the shaft and the propeller boss shall be such as to ensure that the unit interface pressure induced by the average torque at the rated number of revolutions and rated output of the engine on the side surface of the keyway does not exceed 0.5 of the yield point of the material of the shaft or flange respectively.

4.5.4 It is recommended that propeller shafts be effectively protected by shaft liners or glands against contact with sea water. The protection method is closely related to the minimum diameter of the propeller shaft (see 4.5.1) and the frequency of shaft surveys.

4.5.5 Propeller shaft liners shall be made of high-quality copper alloy resistant to the corrosive effect of sea-water.

The thickness of liner, s, shall not be less than that determined using the following formula:

$$s \geq 0.03\, d_{śr} + 7.5 \quad [\text{mm}] \tag{4.5.5}$$

where:

$d_{śr}$ – see paragraph 4.5.1.

The thickness of the shaft liner between the bearings may be reduced to 0.75s.

4.5.6 In general, continuous, i.e. solid, liners shall be used. Liners consisting of lengths may be recognised as the continuous ones, provided the joining methods are accepted by PRS and the joints are not in way of bearings.

Non-continuous liners, where parts between them are coated with the materials approved by PRS and also using a method accepted by PRS, may be recognised as a means of effective protection of the propeller shaft.

4.5.7 Only propeller shaft glands type-approved by PRS may be used as glands effectively protecting against the contact with sea water.

4.5.8 For ships with ice class, the requirements specified in sub-chapter 25.2 shall, additionally, be fulfilled.

4.6 Shaft Couplings

4.6.1 In general, all coupling bolts at the flanges of shafts shall be fitted. The number of fitted bolts may be reduced to 50% of the total number; however, the number shall not be smaller than three.

Flange joints transmitting the torque by friction only (without fitted bolts) may also be used but their use is subject to special consideration by PRS.

Coupling bolts nuts shall be protected against loosening.

4.6.2 The diameter, d_s, of fitted coupling bolts shall not be less than that determined in accordance with the formula below:

$$d_s = 0.65\sqrt{\frac{{d_p}^3(R_{mp}+160)}{iDR_{ms}}} \quad [\text{mm}] \tag{4.6.2}$$

where:

d_p – design diameter of intermediate shaft, taking into account the ice strengthening, if required, [mm]; when the diameter is increased due to torsional vibrations, d_p shall be taken equal to the actual diameter of the intermediate shaft;
i – number of fitted bolts in the coupling;
D – diameter of the pitch circle of the coupling bolts, [mm];
R_{mp} – tensile strength of shaft material, [MPa];
R_{ms} – tensile strength of bolt material, [MPa]; where $R_{mp} \leq R_{ms} \leq 1.7R_{mp}$, however not exceeding 1000 MPa.

4.6.3 The thickness of coupling flanges (under the bolt heads) of the intermediate shafts and thrust shafts and of the forward coupling flange of the propeller shaft shall not be less than 0.2d_p or d_s, determined in accordance with formula 4.6.2 for the shaft material, whichever is greater.

The thickness of the coupling flange of the propeller shaft, by means of which the propeller shaft is connected with the propeller, shall not be less than 0.25 of the actual shaft diameter $d_{śr}$ in way of the flange.

The use of flanges having non-parallel external surfaces is subject to PRS consideration in each particular case, however their thickness shall not be less than d_s.

4.6.4 The fillet radius at the base of coupling shall not be less than 0.08 of the actual shaft diameter.

The fillet may be performed by the variable radii, provided however, that the coefficient of the stress concentration is not greater than that obtained by one radius used to carry out the fillet. The fillet surface shall be smooth and not affected by the recesses for heads and nuts of coupling bolts.

4.6.5 The cone taper shall not exceed 1:12 where a coupling is key-fitted.

4.6.6 Dimensions of both the keyway and key shall be such as to ensure that the unit interface pressure induced by the average torque at the rated number of revolutions and rated output of the engine on the side surface of the keyway does not exceed 0.5 of the yield point of the material of the shaft or flange respectively. The edges of the keyway at the surface of the shaft taper shall not be sharp. The lower keyway corners shall be rounded to a radius of about 0.0125 of the propeller shaft diameter $d_{śr}$, however not less than 1.0 mm.

4.6.7 Where threaded holes are provided to accommodate the securing screws for propeller keys, such holes shall be located either in the middle length of the keyway or in the after half of the keyway.

4.7 Propeller Shaft Bearings

4.7.1 The length of the shaft bearing next to the propeller shall be determined as follows:

.1 for water lubricated bearings lined with lignum vitae or rubber-based materials – not less than $4d_{sr}$ (d_{sr} – see paragraph 4.5.1);

.2 for bearings lined with white metal – not less than $2d_{sr}$, however, if the nominal bearing pressure due to the mass of propeller and propeller shaft does not exceed 0.65 MPa;

.3 for bearings of synthetic material – subject to PRS agreement in each particular case.

4.7.2 If water is used for lubrication of the stern tube bearings, then a non-return valve shall be fitted on the stern tube or afterpeak bulkhead.

The line supplying water for the stern tube bearing lubrication shall be fitted with a flow indicator.

4.7.3 Oil lubricated stern tube bearings shall be provided with means of forced cooling of the oil, unless the afterpeak tank is always filled with water.

Means shall be provided for continuous monitoring of the oil temperature in oil lubricated bearings.

4.7.4 For oil lubricated bearings the gravity tanks shall be located above the waterline and provided with level indicators and low oil level alarm.

4.8 Braking Devices

Shafting shall be provided with a braking device; the following devices may be used: brake, turning gear or other locking equipment precluding the shafting rotation in case of failure of the main propulsion machinery.

5 PROPELLERS

5.1 General Requirements

5.1.1 The design of propellers other than classical screw propellers is subject to PRS consideration in each particular case.

5.1.2 Guidelines for the repair of propellers are specified in *Publication No. 7/P – Repair of Cast Copper Alloy Propellers*.

5.2 Blade Thickness

5.2.1 Blade thickness, s, shall not be less than that determined in accordance with the formula below:

$$s = 0.95 \frac{3.65A}{\sqrt[3]{(0.312+\frac{H}{D})^2}} \sqrt{\frac{P}{nbZM}} \tag{5.2.1}$$

where:

s – maximum thickness of expanded cylindrical section of blade, measured perpendicularly to the blade pressure side or geometrical chord of the section at the radius 0.2R for solid propellers, 0.25R or 0.3R for built-up propellers, 0.35R for CP propellers and 0.6R for all propellers, irrespective of their design, [mm];

A – coefficient determined in accordance with Table 5.2.1, for the radius 0.2R, 0.25R, 0.3R, 0.35R or 0.6R, respectively, and also for the required rake at blade tip; if the rake differs from the values given in the Table, coefficient A shall be assumed as for the nearest maximum value of that rake;

Table 5.2.1
Values of coefficient *A*

Blade radius, [m]	Rake at blade tip, as measured along the blade pressure side, [deg]								
	0	2	4	6	8	10	12	14	16
0.20R	390	391	393	395	397	400	403	407	411
0.25R	378	379	381	383	385	388	391	394	398
0.30R	367	368	369	371	373	376	379	383	387
0.35R	355	356	357	359	361	364	367	370	374
0.60R	236	237	238	240	241	243	245	247	249

P – rated power of main engine, [kW];
n – rated number of propeller shaft revolutions, [rpm];
Z – number of blades;
b – developed blade width of cylindrical sections at the radius of 0.2R, 0.25R, 0.3R, 0.35R or 0.6R, respectively, [m];
D – propeller diameter, [m];
R – propeller radius, [m];
$\frac{H}{D}$ – pitch ratio at the radius of 0.7R;
M = $0.6R_{m(s)} + 180$, however not more than 570 MPa for steel and not more than 610 MPa for non-ferrous alloys;
$R_{m(s)}$ – ultimate tensile strength of the blade material, [MPa].

5.2.2 The thickness at the blade tip shall not be less than 0.0035D.

5.2.3 Intermediate thicknesses of blade shall be so chosen that the contour lines of the maximum blade thickness sections run smoothly from the root, through intermediate profiles to the tip.

5.2.4 In justified cases PRS may consider proposals different from the requirements specified in paragraphs 5.2.1 and 5.2.2, provided that detailed strength calculations are submitted.

5.2.5 For ships with ice class, the requirements specified in sub-chapter 25.3 shall also be fulfilled.

5.3 Bosses and Blade Fastening Parts

5.3.1 Fillet radii of the transition from blade to boss at the location of maximum blade thickness shall be at least 0.04D at the blade suction side and at least 0.03D at the blade pressure side (D – propeller diameter).

If the blade is not raked, the fillet radius at both sides shall be at least 0.03D.

5.3.2 In propellers with glands in accordance with the requirements specified in paragraph 4.5.2, the propeller boss shall be provided with holes to fill the void spaces between the boss and the shaft cone with grease. The grease shall also fill the void space inside the propeller cap.

The grease used for filling the above-mentioned spaces shall not cause corrosion.

5.3.3 Where propeller blades are bolted to the hub, the thread root diameter of these bolts shall not be less than d_s determined using the following formula:

$$d_s = ks\sqrt{\frac{bR_{m(s)}}{d_1 R_{m(sm)}}} \quad \text{[mm]} \tag{5.3.3}$$

where:

k = 0.33 for 3 bolts used on blade pressure side;
k = 0.30 for 4 bolts used on blade pressure side;
k = 0.28 for 5 bolts used on blade pressure side;
s – maximum thickness of blade, measured at the boss, in the section calculated in accordance with 5.2.1, [mm];
b – developed blade width (calculated section) measured at the boss, [m];
$R_{m(s)}$ – tensile strength of the blade material, [MPa];
$R_{m(sm)}$ – tensile strength of the bolt material, [MPa];
d_1 – diameter of the fixing bolts' circle; for different arrangement of bolts, i.e. outside the circle, d_1 = 0.85l.
l – distance between the remotest bolts), [m].

5.4 Material for Propellers

Solid propellers, blades, as well as bosses of built-up propellers and controllable pitch propellers shall be made of copper alloys or cast alloy steel in accordance with the requirements specified in the *Rules for the Classification and Construction of Sea-going Ships*, *Part IX – Materials and Welding*, sub-chapter 17.3 or 13.2, respectively.

For ships without ice strengthening, such a material as MM 55 in accordance with standard PN-H-87026:1991, having the following properties: $R_m \geq 450$ MPa, $R_e \geq 180$ MPa, $A_5 \geq 15$ %, may also be accepted.

Application of other materials is subject to PRS consideration in each particular case.

5.5 Balancing Screw Propellers, Thruster Propellers and Active Rudders

5.5.1 After final machining, screw propellers shall be balanced in accordance with the requirements of the relevant standards.

5.5.2 The difference in mass between basic and spare blades of built-up propellers and controllable pitch propellers shall not exceed 1.5 %.

5.6 Controllable Pitch Propellers

5.6.1 Hydraulic power operating system of the propeller blades pitch setting device of restricted service ships – mark I shall be served by two independent pumps of equal capacity – one service and one standby pump. One of the pumps may be driven from the main engine; in that case this pump shall be capable of operating the propeller blades under all operating conditions of the engine. For ships fitted with two independent propulsion systems (engine with a controllable pitch propeller) standby pumps are not required. Restricted service ships – marks: II and III, irrespective of the number of propulsion systems, need not be fitted with standby pumps.

5.6.2 The propeller blades pitch setting device shall be so designed as to allow the positioning of blades for running ahead in case of failure of the hydraulic power operating system.

5.6.3 The time of reversing the propeller blades from ”full ahead” to ”full astern” position, with the main engine not running, shall not exceed 20s.

6 GEARING, DISENGAGING AND FLEXIBLE COUPLINGS

6.1 General Requirements

6.1.1 The construction of a gear shall ensure normal operation in the heel and trim conditions defined in sub-chapter 1.9.

6.1.2 Rotating parts of gears and couplings shall be balanced by the manufacturer with the accuracy defined by general and manufacturer's standards. The balancing shall be documented by a report.

.1 Static balancing shall be applied to parts rotating with the following tangential velocity:
$v \geq 40$ m/s, if subjected to complete machining securing their alignment;
$v \geq 25$ m/s, if not subjected to such machining.

.2 Dynamic balancing shall be applied to parts rotating with a tangential velocity:
$v \geq 50$ m/s.

6.2 Gearing

6.2.1 General Provisions

6.2.1.1 The requirements specified in this sub-chapter apply to the propulsion gears and auxiliary gears with cylindrical wheels of external and internal mesh having spur or helical teeth of involute profile.

Other types of transmission gear are subject to PRS consideration in each particular case.

6.2.1.2 The technical documentation of gears (see paragraph 1.3.3.2) shall contain all the data necessary for calculation carried out following the procedure specified in sub-chapter 6.2.3.

The calculation applies to gear wheels and shafts transmitting the power from the engine output to gear output.

6.2.2 Input Data for Stress Calculation in Gear Wheel Teeth

6.2.2.1 The symbols and definitions used in this sub-section are based mainly on standards ISO 6336, PN-92/M-88509/00 and PN-93/14-88509/01 concerning the calculation of gear transmission capacity taking into account the contact stress (following the procedure specified in sub-chapter 6.2.4) and bending stress in the tooth root (following the procedure specified in sub-chapter 6.2.5).

6.2.2.2 In order to simplify the requirements' provisions, the following definitions have been adopted:

pinion – the gear wheel of the pair with the smaller number of teeth (all symbols concerning this wheel are marked with subscript character 1),

wheel – the gear wheel of the pair with the greater number of teeth (all symbols concerning this wheel are marked with subscript character 2).

For the purposes of ship gearings' (gear wheels) calculation the following symbols apply:

a – centre distance, [mm];
b – face width, [mm];
b_1 – toothed rim width – pinion, [mm];
b_2 – toothed rim width – wheel, [mm];
d – pitch cylinder diameter (reference diameter), [mm];
d_1 – pitch cylinder diameter – pinion, [mm];
d_2 – pitch cylinder diameter – wheel, [mm];
d_{a1} – tip circle diameter – pinion [mm];
d_{a2} – tip circle diameter – wheel, [mm];
d_{b1} – base circle diameter – pinion, [mm];
d_{b2} – base circle diameter – wheel, [mm];
d_{f1} – root circle diameter – pinion, [mm];
d_{f2} – root circle diameter – wheel, [mm];
d_{w1} – working circle diameter – pinion, [mm];
d_{w2} – working circle diameter – wheel, [mm];
F_t – rated tangential force at working cylinder, [N];
F_b – rated tangential force at transverse section of base cylinder, [N];
h – tooth depth, [mm];
m_n – normal module, [mm];
m_t – transverse module, [mm];
n_1 – rotational speed – pinion, [rpm];
n_2 – rotational speed – wheel, [rpm];
P – maximum power transmitted by gearing (for main gears for ships with ice class, the requirements specified in paragraph 25.1 shall be taken into account), [kW];
T_1 – torque transmitted by pinion, [Nm];
T_2 – torque transmitted by wheel, [Nm];
u – gear ratio;
v – tangential velocity at generating cylinder, [m/s];
x_1 – correction coefficient of basic rack tooth profile – pinion
x_2 – correction coefficient of basic rack tooth profile – wheel;
z_1 – number of teeth – pinion;
z_2 – number of teeth – wheel;

z_n – virtual number of teeth;
α_n – profile angle at normal section of pitch cylinder, [°];
α_t – profile angle at transverse section of pitch cylinder, [°];
α_{tw} – profile angle at transverse section of working cylinder, [°];
β – base helix angle at pitch cylinder, [°];
β_b – base helix angle at base cylinder, [°];
ε_α – transverse contact ratio, [–];
ε_β – pitch contact ratio, [–];
ε_γ – total contact ratio, [–];
$inv\ \alpha$ – tooth profile involute angle associated with considered profile angle α, [rad];
α – profile angle (for definition of involute angle), [°].

Notes:

1. z_2, α, d_2, d_{a2}, d_{b2} and d_{w2} are negative for internal mesh.
2. In the formula defining the teeth contact stress, b is the mesh width at the working cylinder.
3. In the formula defining the bending stress in teeth roots, b_1 and b_2 are the widths at respective teeth roots. In no case b_1 and b_2 shall be greater than b by more than one module (m_n) at each side.
4. Gearing width b may be used in the formula defining the bending stress in teeth roots if barrel shape or relieve of teeth tips has been applied.

6.2.2.3 Selected Formulae for Gearing:

Gearing ratio is defined as follows:

$$u = \frac{z_2}{z_1} = \frac{d_{w2}}{d_{w1}} = \frac{d_2}{d_1} \qquad (6.2.2.3)$$

where u takes the following signs:

– plus – for external mesh,
– minus – for internal mesh.

$$\mathrm{tg}\alpha_t = \frac{\mathrm{tg}\alpha_n}{\cos\beta}$$

$$\mathrm{tg}\beta_b = \mathrm{tg}\beta \cdot \cos\alpha_t$$

$$d = \frac{z \cdot m_n}{\cos\beta}$$

$$d_b = d \cdot \cos\alpha_t = d_w \cdot \cos\alpha_{tw}$$

$$a = \frac{d_{w1} + d_{w2}}{2}$$

$$z_n = \frac{z}{\cos^2\beta_b \cdot \cos\beta}$$

$$m_t = \frac{m_n}{\cos\beta}$$

$$inv\alpha = \mathrm{tg}\alpha - \frac{\pi \cdot \alpha}{180}$$

$$inv\alpha_{tw} = inv\alpha_t + 2\mathrm{tg}\alpha_n \cdot \frac{x_1 + x_2}{z_1 + z_2}$$

$$\varepsilon_a = \frac{0{,}5\sqrt{d_{a1}^2 - d_{b1}^2} \pm 0{,}5 \cdot \sqrt{d_{\alpha2}^2 - d_{b2}^2} - \alpha \cdot \sin\alpha_{tw}}{\pi \cdot m_n \cdot \frac{\cos\alpha_t}{\cos\beta}}$$

Note:
In the above formula (±) symbol shall be interpreted as follows:
"+" for external mesh,
"–" for internal mesh.

$$\varepsilon_\beta = \frac{b \cdot \sin\beta}{\pi \cdot m_n}$$

Note:
For double helical gear, *b* shall be taken as the single helical width.

$$\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$$

$$v = \frac{\pi \cdot d_1 \cdot n_1}{60\,000} = \frac{\pi \cdot d_2 \cdot n_2}{60\,000}$$

$$d_{w1} = 2\alpha \cdot \frac{z_1}{z_1 + z_2};\ \ d_{w2} = 2\alpha \cdot \frac{z_2}{z_1 + z_2}\ \ [\mathrm{mm}]$$

6.2.2.4 Rated Tangential Force F_t

Rated tangential force F_t, tangent to working cylinder and positioned in the plane perpendicular to the rotation axis is calculated from the maximum continuous power transmitted by the gear taking account of the requirement specified in paragraph 25.1, using the following formulae:

$$T_1 = 9549\frac{P}{n_1};\ T_2 = 9549\frac{P}{n_2} \tag{6.2.2.4-1}$$

$$F_t = 2000\frac{T_1}{d_1} = 2000\frac{T_2}{d_2}\ \ [\mathrm{N}] \tag{6.2.2.4-2}$$

6.2.3 Coefficients Common for Checked Strength Conditions (Contact and Bending Stresses)

This section defines the coefficients applied in the formulae checking gear wheel teeth strength for the contact stress (in accordance with 6.2.4) and for the bending stress (in accordance with sub-chapter 6.2.5). Other coefficients specific for the strength formulae are included in sub-chapters 6.2.4 and 6.2.5.

All the coefficients shall be calculated from the respective formulae or in accordance with the particular instructions.

6.2.3.1 Application Factor K_A

The application factor takes into account the dynamic overloads generated in the gear by the external forces.

For gears designed for unlimited life-span, the K_A shall be defined as the ratio of maximum torque occurring in the gear (assuming periodically variable load) to the rated torque.

The rated torque used in further calculations shall be taken as the ratio of rated power to the rated rotational speed.

K_A factor depends mainly on:
– driving and driven equipment characteristics,
– mass ratio,
– type of couplings,
– operating conditions (overspeed, variation of propeller load, etc.).

Operating conditions shall be carefully analysed in the rotational speed range near the critical speed.

K_A factor shall be determined by measurements or using an analytical method approved by PRS. Where the factor is impossible to be determined that way, its value may be taken in accordance with Table 6.2.3.1.

Table 6.2.3.1
Values of K_A for different applications

Gear driving machine	K_A	
	Main propulsion gears	Auxiliary gears
Diesel engine with hydraulic or electromagnetic slip clutch	1	1
Diesel engine with highly elastic coupling	1.3	1.2
Diesel engine with other couplings	1.5	1.4
Electric motor	–	1

6.2.3.2 Load Sharing Factor K_γ

The load-sharing factor takes into account uneven distribution of load in multi-stage or multi-way gears (double tandem, planetary, double helical, etc. gears).

K_γ is defined as the ratio of the maximum load in true mesh to the evenly distributed load. This factor depends mainly on accuracy and flexibility of gear stages and the ways of load distribution.

K_γ shall be determined by measurements or using an analytical method. Where such methods are unavailable, K_γ shall be calculated as follows:

– for planetary gears:

$$K_\gamma = 1 + 0.25\sqrt{n_{pl} - 3} \qquad (6.2.3.2\text{-}1)$$

where:

$n_{pl} \geq 3$ – number of planet wheels

– for double tandem gears:

$$K_\gamma = 1 + \frac{0.2}{\varphi} \qquad (6.2.3.2\text{-}2)$$

where:

ϕ – twist of shaft relieving liner at full load, [°]

– for double-helical gears:

$$K_\gamma = 1 + \frac{F_{ext}}{F_t \cdot \mathrm{tg}\beta} \qquad (6.2.3.2\text{-}3)$$

where:

F_{ext} – external axial force (generated outside the gear), [N].

6.2.3.3 Dynamic Factor K_v

Dynamic factor K_v takes into account the dynamic load arising inside the gear as a result of vibrations of pinion and wheel with respect to each other.

K_v is defined as the ratio of the maximum load acting on the tooth side surface to the maximum external load defined as ($F_t \cdot K_A \cdot K_\gamma$).

This factor depends mainly on:

– mesh errors (depending on pitch and profile errors),
– pinion's and wheel's weights,
– changes in mesh rigidity during the wheel load cycle,
– tangential velocity at working cylinder,

– dynamical unbalance of wheels and shaft,
– rigidity of shaft and bearings,
– gear damping characteristics.

Where all the following conditions are fulfilled:
a) steel gear wheels or wheels with heavy rims,
b) $\frac{F_t}{b} > 150$ [N/mm],
c) $z_1 < 50$,
d) parameter $\frac{v \cdot z_1}{100}$ is within the sub-critical range:
   – for helical gears $\frac{v \cdot z_1}{100} < 14$;
   – for spur gears $\frac{v \cdot z_1}{100} < 10$;
   – for other types of gears $\frac{v \cdot z_1}{100} < 3$.

dynamic factor K_v may be calculated as follows:

.1 for spur gears:
K_v – in accordance with Fig. 6.2.3.3-2,

.2 for helical gears:
– if $\varepsilon_\beta > 1$
K_v – in accordance with Fig.6.2.3.3-1,
– if $\varepsilon_\beta < 1$
K_v – is obtained by linear interpolation using the following formula
$K_v = K_{v2} - \varepsilon_\beta \cdot (K_{v2} - K_{v1})$,

where:
K_{v1} – value of K_v for helical gears, see Fig.. 6.2.3.3-1,
K_{v2} – value of K_v for spur gears, see Fig. 6.2.3.3-2.

.3 For all gear types, factor K_v may also be calculated using the following formula:

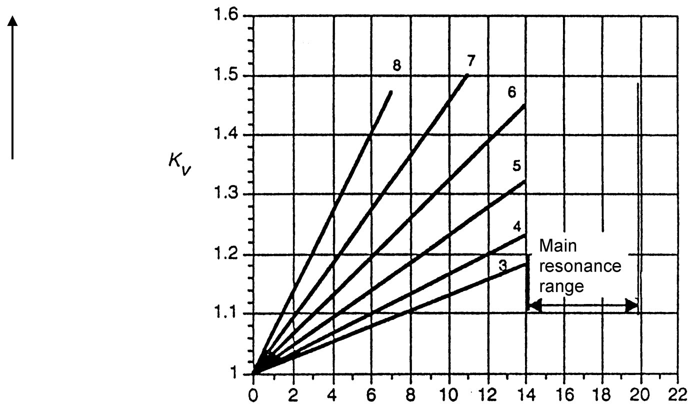
$$K_v = 1 + K_1 \cdot \frac{v \cdot z_1}{100} \qquad (6.2.3.3.3)$$

where:
K_1 – in accordance with Table 6.2.3.3.

Table 6.2.3.3
Values of K_1 for calculation of K_v

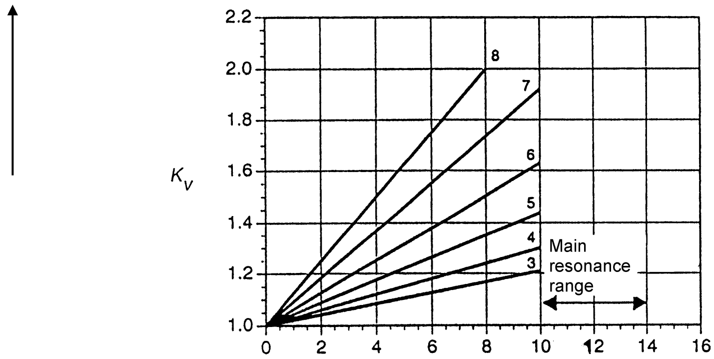
	K_1					
	Accuracy class acc. to ISO 1328					
	3	4	5	6	7	8
Spur gear	0.022	0.030	0.043	0.062	0.092	0.125
Helical gear	0.0125	0.0165	0.0230	0.0330	0.0480	0.0700

Note:
If gear wheels have been made with different accuracy classes, then the lowest class shall be taken for calculation.



$\frac{vz_1}{100}$ [m/s]

Fig. 6.2.3.3-1
Dynamic factor for helical gears. Accuracy classes 3 ÷ 8 acc. to ISO 1328



$\frac{vz_1}{100}$ [m/s]

Fig. 6.2.3.3-2
Dynamic factor for spur gear. Accuracy classes 3 ÷ 8 acc. to ISO 1328

For other gears than those specified above, factor K_v shall be calculated in accordance with the requirements of standard ISO 6336 – method B.

6.2.3.4 Longitudinal Load Distribution Factors $K_{H\beta}$ and $K_{F\beta}$

Longitudinal load distribution factors: $K_{H\beta}$ – for contact stress and $K_{F\beta}$ – for tooth root bending stress, take into account the effects of uneven load distribution throughout the tooth face width.

$K_{H\beta}$ is defined as:

$$K_{H\beta} = \frac{\max contact\ stress}{mean\ contact\ stress}$$

$K_{F\beta}$ is defined as:

$$K_{F\beta} = \frac{tooth\ foot \max bending\ stress}{tooth\ foot\ mean\ bending\ stress}$$

The tooth foot mean bending stress is referred to the face width b_1 or b_2 under consideration.

Factors $K_{H\beta}$ and $K_{F\beta}$ depend mainly on:
– teeth machining accuracy;
– assembly errors due to hole boring errors;

– bearings' clearances;
– misalignment of pinion and wheel axes;
– deformations due to insufficient rigidity of gear parts, shafts, bearings, casing and foundation;
– thermal elongations and other deformations at working temperature;
– compensating construction of parts (barrel shape, tooth tips' relief etc.).

The relationship between factors $K_{F\beta}$ and $K_{H\beta}$ is as follows:

.1 For greater interface pressure at tooth tips, $K_{F\beta}$ shall be determined in accordance with the following equation:

$$K_{F\beta} = \left(K_{H\beta}\right)^{N} \qquad (6.2.3.4.1)$$

where: $N = \frac{(\frac{b}{h})^2}{1+\frac{b}{h}+(\frac{b}{h})^2}$ $\quad \frac{b}{h} = \min(\frac{b_1}{h_1}; \frac{b_2}{h_2})$

Note:

For double helical gear, b shall be taken as a half of the wheel width.

.2 Where the teeth tips are subjected to low interface pressure or are relieved (barrel shape, tips' relief):

$$K_{F\beta} = K_{H\beta}$$

Contact load distribution factor $K_{H\beta}$ and tooth root bending load distribution $K_{F\beta}$ may be determined in accordance with the requirements specified in standard ISO 6336/1 – method C2.

6.2.3.5 Transverse Load Distribution Factors $K_{H\alpha}$ and $K_{F\alpha}$

Transverse load distribution factors such as:

$K_{H\alpha}$ – for contact stress,
$K_{F\alpha}$ – for tooth root bending stress,

involve the effects of pitch and profile errors on the transverse distribution of the load between two or more pairs in mesh.

Factors $K_{H\alpha}$ and $K_{F\alpha}$ depend mainly on:
– general rigidity of mesh;
– total tangential force ($F_t \cdot K_A \cdot K_\gamma \cdot K_v \cdot K_{H\beta}$);
– pitch error on pitch cylinder;
– tooth tip blunting;
– permissible variability of tangential velocity.

Transverse load distribution factors $K_{H\alpha}$ – for contact stress and $K_{F\alpha}$ – for tooth root bending stress shall be determined in accordance with the requirements specified in standard ISO 6336 – method B.

6.2.3.6 Factor selection methods other than those specified in sub-chapter 6.2.3 may be used on condition that they have been approved by PRS.

6.2.4 Contact Stress in Gear Wheel Teeth

6.2.4.1 The criterion of strength for the contact stress is specified using Hertzian formulae for calculation of interface pressure at the active mesh point (or at the internal mesh point) of a single pair of teeth. Contact stress σH shall not exceed the value of permissible contact stress σHP.

6.2.4.2 The basic formula for calculation of contact stress σ_H is as follows:

$$\sigma_H = \sigma_{H0}\sqrt{K_A \cdot K_\gamma \cdot K_v \cdot K_{H\alpha} \cdot K_{H\beta}} \le \sigma_{HP} \quad [\mathrm{N/mm^2}] \qquad (6.2.4.2)$$

where:

σ_{H0} – basic value of contact stress in pinion and wheel determined using the following formulae:

$$\sigma_{H0} = Z_B \cdot Z_H \cdot Z_\varepsilon \cdot Z_\beta \cdot Z_E \cdot \sqrt{\frac{F_t}{d_{w1} \cdot b} \cdot \frac{u+1}{u}} \quad [\mathrm{N/mm^2}] \text{ for pinion,}$$

$$\sigma_{H0} = Z_D \cdot Z_H \cdot Z_\varepsilon \cdot Z_\beta \cdot Z_E \cdot \sqrt{\frac{F_t}{d_{w2} \cdot b} \cdot \frac{u+1}{u}} \quad [\mathrm{N/mm^2}] \text{ for wheel,}$$

where:

F_t, b, d, u (see sub-chapter 6.2.2);
Z_B – single tooth pair contact factor – for pinion (see paragraph 6.2.4.4);
Z_D – single tooth pair contact factor – for wheel (see paragraph 6.2.4.4);
Z_H – zone factor (see paragraph 6.2.4.5);
Z_E – flexibility factor (see paragraph 6.2.4.6);
Z_ε – contact ratio factor (see paragraph 6.2.4.7);
Z_β – helix angle factor (see paragraph 6.2.4.8);
K_A – application factor (see paragraph 6.2.3.1);
K_γ – load sharing factor (see paragraph 6.2.3.2);
K_v – longitudinal load distribution factor (see paragraph 6.2.3.3);
$K_{H\alpha}$ – transverse load distribution factor (see paragraph 6.2.3.5);
$K_{H\beta}$ – longitudinal load distribution factor (see paragraph 6.2.3.4).

6.2.4.3 Calculation of Allowable Contact Stress σ_{HP}

Allowable load stresses σ_{HP} shall be calculated separately for each gear pair (pinion and wheel) using the following formula:

$$\sigma_{HP} = \frac{\sigma_{Hlim}}{S_H} \cdot Z_N \cdot Z_L \cdot Z_v \cdot Z_R \cdot Z_W \cdot Z_X \quad [\mathrm{N/mm^2}] \qquad (6.2.4.3)$$

where:

σ_{Hlim} – tooth material fatigue strength for contact stress, [N/mm²] (see paragraph 6.2.4.9);
S_H – safety factor for contact stress (see paragraph 6.2.4.14);
Z_N – life factor for contact stress (see paragraph 6.2.4.10);
Z_L – lubrication factor (see paragraph 6.2.4.11);
Z_v – velocity factor (see paragraph 6.2.4.11);
Z_R – roughness factor (see paragraph 6.2.4.11);
Z_W – hardness ratio factor (see paragraph 6.2.4.12);
Z_X – size factor (see paragraph 6.2.4.13).

6.2.4.4 Single Tooth Pair Contact Factors Z_B and Z_D

Single tooth pair contact factors, Z_B – for pinion and Z_D – for wheel, take account of the tooth side curvature effect on the contact stress at the pitch point (line) of single pair of teeth with respect to Z_H.

These factors enable conversion of the contact stress determined at the pitch point into the contact stress taking account of the tooth side surface curvatures at the central point of a single pair contact.

Factors: Z_B – for pinion and Z_D – for wheel shall be determined as follows:

– for spur gearing ($\varepsilon_\beta = 0$):

$$Z_B = \max(M_1;1) \quad (6.2.4.4\text{-}1)$$

$$Z_D = \max(M_2;1) \quad (6.2.4.4\text{-}2)$$

where:

$$M_1 = \frac{\mathrm{tg}\alpha_{tw}}{\sqrt{\left[\sqrt{\left(\frac{d_{a1}}{d_{b1}}\right)^2 - 1} - \frac{2\pi}{z_1}\right] \cdot \left[\sqrt{\left(\frac{d_{a2}}{d_{b2}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\frac{2\pi}{z_2}\right]}}$$

$$M_2 = \frac{\mathrm{tg}\alpha_{tw}}{\sqrt{\left[\sqrt{\left(\frac{d_{a2}}{d_{b2}}\right)^2 - 1} - \frac{2\pi}{z_2}\right] \cdot \left[\sqrt{\left(\frac{d_{a1}}{d_{b1}}\right)^2 - 1} - (\varepsilon_\alpha - 1)\frac{2\pi}{z_1}\right]}}$$

– for helical gearing where,

if $\varepsilon_\beta \geq 1$

$$Z_B = Z_D = 1$$

if $\varepsilon_\beta < 1$, the values of Z_B and Z_D shall be determined by linear interpolation from the corresponding values of Z_B and Z_D for spur gears and for helical gears, for which $\varepsilon_\beta \geq 1$.

Therefore:

$$Z_B = \max\{[M_1 - \varepsilon_\beta \cdot (M_1 - 1)];\ 1\} \quad (6.2.4.4\text{-}3)$$

$$Z_D = \max\{[M_2 - \varepsilon_\beta \cdot (M_2 - 1)];\ 1\} \quad (6.2.4.4\text{-}4)$$

6.2.4.5 Zone Factor Z_H

Zone factor Z_H takes account of the effect of tooth side curvature at the pitch point on the interface pressure defined by Hertzian formulae and of the ratio of the tangent forces at pitch cylinder to the normal forces at working cylinder.

Zone factor Z_H shall be calculated using the following formula:

$$Z_H = \sqrt{\frac{2\cos\beta_b \cdot \cos\alpha_{tw}}{\cos^2\alpha_t \cdot \sin\alpha_{tw}}} \quad (6.2.4.5)$$

6.2.4.6 Material Elasticity Factor Z_E

Material elasticity factor Z_E takes account of the effect of the material elasticity determined by Young's modulus and Poisson ratio on the contact stress calculated using Hertzian formulae.

Factor Z_E shall be calculated using the following formula:

$$Z_E = \sqrt{\frac{E_1 \cdot E_2}{\pi[(1-\nu_1^2) \cdot E_1 + (1-\nu_2^2) \cdot E_2]}} \quad [\mathrm{N}^{1/2}/\mathrm{mm}] \quad (6.2.4.6)$$

where:

E_1, E_2 – Young's modulus for tooth material, [N/mm^2];
ν_1, ν_2 – Poisson ratio for tooth material, [–].

For steel gear wheels where $E_1 = E_2 = 206\,000$ N/mm^2 and $\nu_1 = \nu_2 = 0.3$, the elasticity factor is:

$$Z_E = 189.8\ ;\quad [\mathrm{N}^{1/2} / \mathrm{mm}].$$

Standard ISO 6336 may be used to determine the value of Z_E.

6.2.4.7 Contact Ratio Factor Z_ε

Contact ratio factor Z_ε takes account of transverse contact ratio ε_α and overlap ratio ε_β on the specific teeth contact load.

Contact ratio factor Z_ε shall be calculated as follows:
– for spur gears, using the following formula:

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}} \qquad (6.2.4.7\text{-}1)$$

– for helical gears, using an appropriate alternative formula:

if $\varepsilon_\beta < 1$

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3} \cdot \left(1-\varepsilon_\beta\right) + \frac{\varepsilon_\beta}{\varepsilon_\alpha}} \qquad (6.2.4.7\text{-}2)$$

if $\varepsilon_\beta \geq 1$

$$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}} \qquad (6.2.4.7\text{-}3)$$

6.2.4.8 Helix Angle Factor Z_β

Helix angle factor Z_β takes account of the effect of helix angle on the surface durability, considering such variables as load distribution along the contact line. Z_β depends on the helix angle only.

Helix angle factor Z_β shall be calculated using the following formula:

$$Z_\beta = \sqrt{\cos\beta} \qquad (6.2.4.8)$$

6.2.4.9 Endurance Limit for Hertzian Contact Stress $\sigma_{H\lim}$

The value of $\sigma_{H\lim}$ represents the permissible continuously repeated contact stress for a certain material. This value may be considered a level of contact stress which the material can endure throughout at least 5×10^7 stress cycles with no pitting effect.

For this purpose pitting may be determined:
– for not hardened surfaces of teeth, if the pitting area exceeds 2% of the total working surface,
– for hardened surfaces of teeth, if pitting area is greater than 0.5% of the total working surface or exceeds 4% of a single tooth total surface.

The value of $\sigma_{H\lim}$ corresponds to 1% (or lower) probability of damage.

The endurance limit for Hertzian contact stress depends mainly on:
– material composition, homogeneity and defects;
– mechanical properties;
– residual stress;
– hardening process, hardened layer depth, hardening gradient;
– material structure (forged, rolled, cast).

The allowable value of contact stress $\sigma_{H\lim}$ shall be determined in accordance with the test results of the material used for the construction. If such results are unavailable, the contact stress shall be determined in accordance with the requirements of standard ISO 6336/5 – Quality Class MQ.

6.2.4.10 Life Factor for Contact Stress Z_N

Life factor for contact stress Z_N takes account of a higher allowable contact stress where limited durability (i.e. lower number of load cycles) is required.

The factor depends mainly on:
– material and hardening method;
– number of load cycles;
– Z_R, Z_v, Z_L, Z_W, Z_X factors.

Life factor for contact stress Z_N shall be determined in accordance with the requirements specified in standard ISO 6336/2 – method B.

6.2.4.11 Lubrication, Velocity and Roughness Factors Z_L, Z_v and Z_R

Lubrication factor Z_L takes account of the lubricant type and viscosity, velocity factor Z_v , and also takes account of the effect of tangential velocity (v) at pitch diameter, while roughness factor Z_R takes account of the effect of surface roughness on its durability.

These factors shall be calculated for the softer material where the intermashing teeth have different hardness.

These factors depend mainly on:
– the lubricating oil viscosity in way of the teeth contact;
– the sum of momentary velocities on the teeth surfaces;
– the load;
– the relative radius of curvature at pitch point;
– roughness of tooth surface;
– hardness of pinion and wheel.

These factors shall be determined as follows:

.1 lubrication factor Z_L shall be calculated using the following formula:

$$Z_L = C_{ZL} + \frac{4(1-C_{ZL})}{(1.2+\frac{134}{v_{40}})^2} \tag{6.2.4.11.1}$$

where:

v_{40} – rated kinematic viscosity of the oil used in the gear at temperature of 40 °C.

$C_{ZL} = \left(\frac{\sigma_{Hlim}-850}{350}\right)0.08 + 0.83$ for $850 \le \sigma_{H\text{lim}} \le 1200$ [N/mm²]

Note:

If $\sigma_{H\text{lim}} < 850$ MPa, then $C_{ZL} = 0.83$.
If $\sigma_{H\text{lim}} > 1200$ MPa, then $C_{ZL} = 0.91$.

.2 Velocity factor Z_v shall be calculated using the following formula:

$$Z_v = C_{ZV} + \frac{2(1-C_{ZV})}{\sqrt{0.8+\frac{32}{v}}} \tag{6.2.4.11.2}$$

where:

$C_{ZV} = \left(\frac{\sigma_{H\text{lim}}-850}{350}0{,}08\right) + 0{,}85$ for $850 \le \sigma_{H\text{lim}} \le 1200$, [N/mm²]

Note:

If $\sigma_{H\text{lim}} < 850$ MPa, then $C_{ZV} = 0.85$.
If $\sigma_{H\text{lim}} > 1200$ MPa, then $C_{ZV} = 0{,}93$.

.3 Roughness factor Z_R shall be calculated using the following formula:

$$Z_R = \left(\frac{3}{R_{Z10}}\right)^{C_{ZR}} \tag{6.2.4.11.3}$$

where:

$C_{ZR} = 0.32 - 0.0002\sigma_{H\,\text{lim}}$ for $850 \le \sigma_{H\text{lim}} \le 1200$ [N/mm²]

Note:

If $\sigma_{H\lim} < 850$ N / mm², then $C_{ZR} = 0.150$.
If $\sigma_{H\lim} > 1200$ N / mm², then $C_{ZR} = 0.080$.

R_{Z10} – mean amplitude of roughness in intermashing wheels referred to the relative radius of teeth curvature, [µm]

$$R_{Z10} = R_{red} \sqrt[3]{\frac{10}{\rho_{red}}}$$

where:

R_{red} – mean amplitude of roughness height in intermashing wheels (to be calculated in accordance with standard ISO 6336), [µm]

$$R_{red} = \frac{R_{Z1} + R_{Z2}}{2} \quad \text{where}$$

if the roughness is given as mean value – R_a

$$R_{Z1} = 6R_{a1}$$

$$R_{Z2} = 6R_{a2}$$

where:

R_{Z1} – pinion roughness height, [µm];
R_{Z2} – wheel roughness height, [µm];
R_{a1} – arithmetic mean of profile deviation from mean pinion profile, [µm];
R_{a2} – arithmetic mean of profile deviation from mean wheel profile, [µm].

Note:

Roughness shall be measured on faces of several teeth.

ρ_{red} – relative radius of teeth curvature in intermashing gear wheels

$$\rho_{red} = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2}$$

where:

$$\rho_{1,2} = 0{,}5 d_{b1,2} tg \alpha_{tw}$$

Note:

d_{b2} is negative for inner gearing.

6.2.4.12 Hardness Ratio Factor Z_W

Hardness ratio factor Z_W takes account of the effect of durability of teeth made of soft steel, intermashing with much harder teeth with smooth surface.

Factor Z_W applies only to softer teeth and depends mainly on:
– softer teeth hardness;
– alloying components of softer teeth;
– roughness of harder teeth faces.

Factor Z_W shall be calculated using the following formula:

$$Z_W = 1.2 - \frac{HB - 130}{1700} \qquad (6.2.4.12)$$

where:

HB – softer material Brinell hardness (BHN),
– if $HB < 130$, then $Z_W = 1.2$;
– if $HB > 470$, then $Z_W = 1$.

6.2.4.13 Size Factor Z_X

Size factor Z_X takes account of the tooth size effect on the permissible contact stress as well as inhomogeneity of the materials' properties.

This factor depends mainly on:
– material and heat treatment;
– teeth and gear box sizes;
– hardening depth ratio to tooth dimensions;
– hardening depth ratio to virtual radius of curvature.

For through hardened teeth and surface hardened teeth with hardening depth appropriate to both the teeth size and relative radius of curvature $Z_X = 1$. If hardening depth is relatively low, then the lesser value of Z_X shall be taken.

6.2.4.14 Contact Stress Safety Factor S_H

The magnitude of safety factor for contact stress S_H depends on the intended use of a gear box, as well as whether it is intended to be used as a single unit or as an element of a set consisting of two or more gear boxes.

The safety factor shall be selected from Table 6.2.4.14.

Table 6.2.4.14

	S_H	
	Multiple set	Single set
Main propulsion gears	1.2	1.4
Auxiliary gears	1.15	1.2

For gearing of independent duplicated propulsion or auxiliary machinery installed onboard the ship in the number greater than required by the *Rules*, the reduced value of S_H may be taken subject to PRS consent in each particular case.

6.2.5 Bending Stress in Gear Wheel Tooth Root

6.2.5.1 A criterion for bending stress in tooth root determines the permissible level of local tensile stress in the tooth root. The root bending stress σ_F and the permissible root bending stress σ_{FP} shall be calculated separately for the pinion and wheel. The value of σ_F shall not exceed that of σ_{FP}. The following formulae apply to gears with toothed rim thickness greater than 3.5 m_n for $\alpha_n \leq 25°$ and $\beta \leq 30°$. For greater values of α_n and β the calculation results shall be confirmed experimentally or verified in accordance with the requirements specified in standard ISO6336 – Method A.

6.2.5.2 The basic formula for bending stress calculation is as follows:

$$\sigma_F = \frac{F_t}{b \cdot m_n} \cdot Y_F \cdot Y_S \cdot Y_\beta \cdot K_A \cdot K_\gamma \cdot K_v \cdot K_{F\alpha} \cdot K_{F\beta} \leq \sigma_{FP} \quad [\mathrm{N/mm^2}] \qquad (6.2.5.2)$$

where:

F_t, b, m_n (see paragraph 6.2.2.2);
Y_F – tooth-form factor (see paragraph 6.2.5.4);
Y_S – stress correction factor (see paragraph 6.2.5.5);
Y_β – helix angle factor (see paragraph 6.2.5.6);
K_A – application factor (see paragraph 6.2.3.1);
K_γ – load sharing factor (see paragraph 6.2.3.2);

K_v – dynamic factor (see paragraph 6.2.3.3);
$K_{F\alpha}$ – transverse load distribution factor (see paragraph 6.2.3.5);
$K_{F\beta}$ – longitudinal load distribution factor (see paragraph 6.2.3.4).

6.2.5.3 The basic formula for allowable bending stress calculation σ_{FP} is as follows:

$$\sigma_{FP} = \frac{\sigma_{FE}}{S_F} \cdot Y_d \cdot Y_N \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X \quad [\mathrm{N/mm^2}] \tag{6.2.5.3}$$

where:

σ_{FE} – endurance limit for bending stress, [N/mm²] (see paragraph 6.2.5.7);
S_F – safety factor for root bending stress (see paragraph 6.2.5.13);
Y_d – design factor (see paragraph 6.2.5.8);
Y_N – life factor for tooth root (see paragraph 6.2.5.9);
$Y_{\delta relT}$ – relative notch sensitivity factor (see paragraph 6.2.5.10);
Y_{RrelT} – relative surface finish factor (see paragraph 6.2.5.11);
Y_X – size factor (see paragraph 6.2.5.12).

6.2.5.4 Tooth Profile Factor Y_F

Tooth profile factor Y_F takes into account an effect of the tooth profile on the nominal bending stress caused by the force applied in the single tooth pair external contact. Factor Y_F shall be determined separately for the pinion and wheel. For helical gears, the tooth profile factor shall be determined for the normal section, i.e. for the virtual spur gear with virtual number of teeth z_n .

Tooth profile factor Y_F shall be determined in accordance with the formula below:

$$Y_F = \frac{6 \cdot \frac{h_F}{m_n} \cdot \cos\alpha_{Fen}}{(\frac{s_{Fn}}{m_n})^2 \cdot \cos\alpha_n} \quad \text{for } \alpha \leq 25° \text{ and } \beta \leq 30° \tag{6.2.5.4}$$

where:

h_F – bending moment arm for root stress caused by the force applied in the single tooth pair external contact, [mm];
s_{Fn} – tooth root chord in critical section, [mm];
α_{Fen} – pressure angle in the single tooth pair external contact at normal section, [°].

Note:

The quantities used to determine Y_F are shown in Fig. 6.2.5.5.

To determine h_F, s_{Fn} and α_{Fen}, the guidelines specified in standard ISO 6336 may be applied.

6.2.5.5 Stress Concentration Factor Y_S

Stress concentration factor Y_S is used for conversion of the nominal bending stress into local stress in the tooth root at the assumption that not only bending stress occurs in the tooth root.

Factor Y_S concerns the force applied in the single tooth pair external contact and shall be determined separately for the pinion and wheel.

Stress concentration factor Y_S shall be determined using the formula below:

$$Y_S = (1.2 + 0.13 \cdot L) \cdot q_S^{\left(\frac{1}{1.12 + \frac{2,3}{L}}\right)} \quad \text{for } 1 \leq q_S < 8 \tag{6.2.5.5}$$

where:

q_S – notch parameter determined using the formula below:

$$q_S = \frac{S_{Fn}}{2\rho_F}$$

where:

ρ_F – tooth root fillet radius, [mm];
L – tooth bending factor determined using the formula below:

$$L = \frac{S_{Fn}}{h_F}$$

h_F , s_{Fn} – see paragraph 6.2.5.4.

To determine ρ_F , the guidelines specified in standard ISO 6336 may be applied.

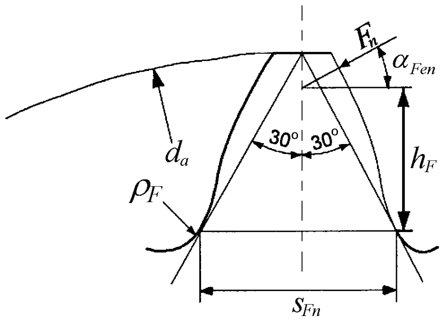


Fig. 6.2.5.5

6.2.5.6 Helix Angle Factor Y_β

Helix angle factor Y_β takes into account the difference between the helix gears and virtual spur gears at normal section for which the calculations are performed. As the contact lines are helical and along the tooth side surface, more favourable stress conditions in the tooth root are taken into account.

The helix angle factor depends on ε_β as well as β, and shall be determined in accordance with the formula below:

$$Y_\beta = 1 - \varepsilon_\beta \frac{\beta}{120} \tag{6.2.5.6}$$

It shall be taken that:

ε_β = 1, if ε_β > 1 and

β = 30°, if β > 30°.

6.2.5.7 Endurance Limit For Bending Stress σ_{FE}

Endurance limit for bending stress σ_{FE} for the particular material represents the value of local tooth root stress limit for long life.

According to standard ISO 6336, the strength determined for 3×10^6 stress cycles is considered as the lowest limit for the bending stress endurance limit.

The quantity of σ_{FE} is defined as non-directional fluctuating load of the minimum value equal to zero (the residual stress due to heat treatment is neglected). Other conditions, such as fluctuating stress, overload etc., are taken into account by design factor Y_d.

The quantity of σ_{FE} corresponds to the probability of damage not exceeding 1%.

The endurance limit depends mainly on:
- material composition, purity and imperfections;
- mechanical properties;
- residual stress;
- hardening procedure, hardened zone depth, hardness gradient;
- material structure (forging, casting, rolled material).

Endurance limit for bending stress σ_{FE} shall be determined in accordance with the results of the tests of actual materials applied. Where such test results are unavailable, the value of the endurance limit for bending stress σ_{FE} shall be determined in accordance with the requirements specified in standard ISO 6336/5 – quality grade MQ.

6.2.5.8 Design Factor Y_d

Design factor Y_d takes into account the effect of load while the ship is going astern and overload due to shrink fit on the tooth root strength compared to the strength of tooth root loaded non-directionally as determined for σ_{FE}.

Design factor Y_d for the load while the ship is going astern shall be determined in accordance with Table 6.2.5.8:

Table 6.2.5.8

	Y_d
In general	1
For gear wheels sporadically loaded with partial power output while the ship is going astern, such as main wheels in reversing gears	0.9
For idle running gear wheels	0.7

6.2.5.9 Life Factor for Tooth Root Y_N

Life factor for tooth root Y_N takes into account the possibility of increased allowable bending stress where the gear box limited life (number of stress cycles) is permitted.

This factor depends mainly on:
- material and hardening;
- number of stress cycles;
- factors $Y_{\delta relT}, Y_{RrelT}, Y_X$.

The life factor for tooth root shall be determined in accordance with the requirements specified in standard ISO 6336/5 – method B.

6.2.5.10 Relative Notch Sensitivity Factor $Y_{\delta relT}$

Relative notch sensitivity factor $Y_{\delta relT}$ indicates the range where theoretical stress concentration is greater than the endurance limit.

This factor depends mainly on the material and relative gradient of stress.

The factor shall be taken as follows:
- for notch parameters (see paragraph 6.2.5.5) within $1{,}5 \le qS < 4$, $Y\delta relT = 1$;
- for notch parameters beyond that interval, in accordance with the requirements specified in standard ISO 6336.

6.2.5.11 Relative Surface Finish Factor Y_{RrelT}

Relative surface finish factor Y_{RrelT} takes account of the relation between the tooth root strength and the surface finish of the tooth root fillet, mainly the roughness amplitude.

Relative surface finish factor Y_{RrelT} shall be determined in accordance with Table 6.2.5.11:

Table 6.2.5.11

	$R_Z < 1$	$1 \le R_Z \le 40$	Material
Y_{RrelT}	1.120	$1.675 - 0.53 \cdot (R_Z+1)^{0,1}$	carburized steels, through hardened steels ($\sigma_B \ge 800$ N/mm^2)
	1.070	$5.3 - 4.2 \cdot (R_Z+1)^{0,01}$	normalized steels ($\sigma_B < 800$ N/mm^2)
	1.025	$4.3 - 3.26 \cdot (R_Z+1)^{0,005}$	nitrided steels

Note:

1. R_Z – average maximum height of the roughness profile of the tooth root fillet.
2. Where the roughness is defined as the arithmetical mean deviation of the profile (R_a), the following formula applies:

$$R_Z = 6R_a$$

This method is applicable only where scratches and similar surface defects are not greater than $2R_Z$.

6.2.5.12 Size Factor Y_X

Size factor Y_X takes account of the reduction in the strength as the tooth size grows.

This factor depends mainly on:
– material and heat treatment;
– tooth module and dimensions of gear wheels;
– case depth to tooth size ratio.

Size factor Y_X shall be determined in accordance with Table 6.2.5.12.

Table 6.2.5.12
Size factor Y_X

$Y_X = 1.00$	for $m_n \le 5$	In general
$Y_X = 1.03 - 0.006\ m_n$	for $5 < m_n < 30$	normalized steels and through hardened steels
$Y_X = 0.85$	for $m_n \ge 30$	
$Y_X = 1.05 - 0.010\ m_n$	for $5 < m_n < 25$	skin hardened steels
$Y_X = 0.80$	for $m_n \ge 25$	

6.2.5.13 Safety Factor for Tooth Root Bending Stress S_F

The quantity of safety factor for tooth root bending stress S_F depends on the gear box intended service and also on whether it is applied in a single unit, or in two or more units.

Safety factor for tooth root bending stress S_F shall be determined in accordance with Table 6.2.5.13.

Table 6.2.5.13

	S_F	
Drive type	Two and more units	Single unit
Main drive	1.55	2
Auxiliary drive	1.4	1.45

For independent duplicated main propulsion gears and gears of auxiliary machinery installed on board the ship in the number greater than required by the *Rules*, the value of S_H may be reduced subject to PRS consent in each particular case.

6.2.6 Shafts

Shafts which are not subjected to variable bending loads shall fulfil, to the applicable extent, the requirements specified in sub-chapters 4.2, 4.3, 4.4 and 4.6.

6.2.7 Gear Wheel Manufacturing – General Notes

6.2.7.1 Welded gear wheels shall be in the stress-relieved condition.

6.2.7.2 Shrink fitted toothed wheel rims shall be so designed to transmit double maximum dynamic torque.

Friction factors for the calculation of shrink fit shall be taken in accordance with Table 6.2.7.2.

Table 6.2.7.2

Fitting method	Steel/steel	Steel/cast iron, including nodular cast iron
Oil heated rim	0.13	0.10
Rim heated in gas furnace (not protected against oil penetration to the rim-wheel contact surface)	0.15	0.12
Contact surfaces degreased and protected against oil penetration	0.18	0.14

Instead of the shrink fit calculations, the results of shrink fit tests with the proof load (in the full range); the testing procedure and proof load selection are subject to PRS agreement in each particular case.

6.2.8 Bearing System

6.2.8.1 Thrust bearing and its foundation shall have sufficient stiffness to prevent adverse deflection and longitudinal vibration of shaft.

6.2.8.2 In general, roller bearings of the main propulsion gear shall be calculated to life time L_{10} equal to:
– 40 000 hours for propeller thrust bearings;
– 30 000 hours for other bearings.

Shorter lifetime may be considered where bearing condition monitoring equipment is provided or operating instructions require inspection of bearings with adequate frequency.

The required lifetime of astern propulsion bearings shall be taken as 5% of the above specified values.

6.2.9 Gearcases

6.2.9.1 Gearcases and their supports shall be designed sufficiently stiff so that movements of the external foundations and the thermal effects under all conditions of service do not disturb the overall tooth contact.

It is recommended that inspection openings be provided in gearcases to enable the teeth of pinions and of wheels to be readily examined.

6.2.9.2 Gearcases fabricated by fusion welding or casting shall be stress relieved before machining operations.

6.2.10 Lubrication

6.2.10.1 Lubrication system shall ensure proper supply of oil to the bearings, teeth and other parts which need lubrication. The relevant requirements specified in Chapter 21 shall also be fulfilled.

6.2.10.2 In gears with medium loads and speeds provided with roller bearings, splash lubrication is permitted.

6.2.10.3 In pressure oil systems, adequate filtering arrangements shall be provided.

Filters in lubrication systems of single main gears shall be so designed as to enable their cleaning without stopping the propulsion system.

6.2.10.4 In pressure oil systems, arrangements for measurement of input and output pressure and temperature as well as alarms giving warning of reaching low oil pressure shall be provided.

In splash lubrication systems, arrangements shall be provided for measurement of oil level in the gearcase.

6.3 Disengaging and Flexible Couplings

6.3.1 General Requirements

6.3.1.1 The requirements specified in this sub-chapter apply to disengaging and flexible couplings.

6.3.1.2 Documentation concerning flexible couplings (see paragraph 1.3.7.9) shall include the following characteristics:
T_{KN} – rated torque for continuous operation;
$T_{K\max}$ – maximum torque for operation in transient conditions;
T_{KW} – allowable dynamic torque for the full range of torques from 0 to T_{KN};
$C_{T\ DYN}$ – dynamic stiffness for the full ranges of torques T_{KN} and T_{KW};
– rotational speed limit;
– allowable torque transmitted by the angular displacement limiter (where provided).

Additionally the following data shall be provided for reference:
- damping coefficient for the full variation ranges of torques T_{KN} and T_{KW};
- allowable power loss P_{KV} in coupling;
- allowable axial and radial displacements as well as angular misalignment;
- allowable service time of flexible components until compulsory replacement.

6.3.1.3 Rigid elements transmitting torque (except for bolts) shall be made from a material with tensile strength $400 < Rm \leq 800$ MPa.

6.3.1.4 Flange connections and connecting bolts shall fulfil the requirements specified in sub-chapter 4.6.

6.3.2 Flexible Couplings

6.3.2.1 Flexible couplings intended for shafting of the ships with one main engine shall be provided with proper arrangements to enable maintaining sufficient speed of ship to ensure its steering qualities when flexible elements have been damaged.

6.3.2.2 If the requirement specified in paragraph 6.3.2.1 is not fulfilled, the static torque breaking elements made from rubber or other synthetic materials shall not be less than eightfold value of the coupling rated torque.

6.3.2.3 Static torque breaking flexible elements in generating sets shall not be less than the torque resulting from the short-circuit current.

Where relevant data are unavailable, the breaking torque shall not be less than 4.5 times as much as the coupling rated torque.

6.3.2.4 Flexible couplings shall endure long-lasting continuous load with the rated torque within the range of temperatures from 5 °C to 60°C.

6.3.3 Disengaging Couplings

6.3.3.1 Disengaging couplings of main engines shall be controlled from the main engine control stand, and shall be fitted also with the arrangements for local control. The control devices shall ensure so smooth engagement of the coupling that the momentary dynamic load does not exceed the maximum torque specified by the manufacturer or double rated engine torque.

6.3.3.2 Where two or more main reversible engines drive one propeller shaft via disengaging couplings, the control arrangements of the couplings shall be so designed as to exclude the possibility of simultaneous engagement when the engine directions of rotation do not provide for the same direction of ship motion.

6.3.4 Emergency Means

Where the propeller shaft is driven through:
– hydraulic or electromagnetic transmission,
– hydraulic or electromagnetic clutch,

provision shall be made to maintain the ship motion with a speed enabling its steering qualities in case of failure of the above-mentioned couplings.

7 THRUSTERS

7.1 Application

7.1.1 The requirements specified in Chapter 7 apply to the ship propulsion, steering or manoeuvring devices which in this Chapter are also referred to as ”devices”. In particular, these requirements cover:
– azimuth thrusters,
– cycloidal propellers,
– retractable and foldable devices,
– devices for dynamic positioning of the ship,
– water-jet propulsion,
– tunnel thrusters.

7.1.2 Devices intended for the main propulsion and steering and for dynamic positioning of the ship are considered as main thrusters and are also referred to as ”main devices”.

Other thrusters are considered as auxiliary ones.

7.2 General Requirements

7.2.1 Where the ship is propelled solely by thrusters, at least two separate devices with independent power supply shall be applied. This requirement does not apply to water-jet propulsion.

The possibility of application of a single device or devices with common power supply is subject to PRS consideration in each particular case.

7.2.2 The devices shall withstand the loads occurring in stationary and transient operating conditions.

7.2.3 Components of thrusters with turning columns which transmit a torque or revolving force shall be calculated taking into account the maximum torque caused by the hydraulic motor turning the column at the maximum difference in pressure of the hydraulic liquid or taking into account the starting torque of the electric motor turning the column. These components shall withstand stoppage of the column turning.

7.2.4 Effective means to prevent outboard water penetration into both the device and hull shall be provided.

7.2.5 Dynamic seals preventing outboard water penetration into the device or hull shall be type-approved by PRS.

7.2.6 Inspection holes shall be provided to enable the necessary periodical survey of the main parts of thrusters.

7.2.7 Thrusters, which are so installed inside the ship hull as to enable their stretching out or turning, shall be located in a separate watertight compartment unless double seals are arranged in accordance with the requirement specified in paragraph 7.2.5. An alarm system warning of water ingress between the seals as well as the possibility of inspection of the seals during dry-docking shall be provided.

7.2.8 Construction of nozzles is subject to PRS consideration in each particular case.

7.2.9 In the case of azimuth thrusters where reverse manoeuvre is effected through the column turning by 180°, the time for such turning shall not exceed 30 s.

7.2.10 Main thrusters shall enable the thrust vector to be controlled from all the main propulsion remote control stands and from the thruster compartment. In each of these locations, indication of the propeller pitch and thrust vector direction, and also means to stop the propeller immediately as well as communications with all other control stands shall be provided. The means for immediate stopping of the propeller shall be independent of the thruster remote control system.

7.3 Drive

7.3.1 Internal combustion engines which drive thrusters directly shall fulfil the requirements specified in Chapter 3. Installations serving engines shall fulfil the relevant requirements specified in *Part VI – Machinery Installations and Refrigerating Plants*, except for the requirement for application of stand-by and spare pumps and other similar appliances.

7.3.2 Hydraulic motors, pumps and other hydraulic components shall be type-approved by PRS.

7.3.3 For main thrusters, a permanently connected spare hydraulic oil storage tank of the capacity sufficient for full oil exchange in at least one thruster shall be provided.

7.3.4 Electric motors, irrespective of their power output, used for powering the main thrusters are subject to PRS survey during their production.

7.4 Gearing and Bearing

7.4.1 Gearings applied in thrusters shall fulfil the relevant requirements specified in Chapter 6.

7.4.2 Gearings of auxiliary devices intended for short-time operation may be calculated for a limited number of operating hours. Calculations of these gears, performed in accordance with the standards in force, are subject to PRS consideration in each particular case.

7.4.3 Basic rating life L10 of rolling-element bearings in main thrusters shall be at least 20 000 hours.

7.4.4 Basic rating life L10 of rolling-element bearings in auxiliary devices shall be at least 5 000 hours.

7.4.5 Bearing of the turning column shall ensure compensation of axial forces in both directions.

7.5 Propulsion Shafting

7.5.1 Shafts shall fulfil the relevant requirements specified in Chapter 4, including the requirements for ice class where applicable.

7.5.2 With respect to torsional vibrations, the relevant requirements specified in Chapter 8 apply.

7.6 Propellers

7.6.1 Fixed pitch propellers and controllable pitch propellers shall fulfil the relevant requirements specified in Chapter 5.

7.6.2 Screw propellers of non-conventional shape and propellers of other types are subject to PRS consideration in each particular case.

7.7 Control Systems

7.7.1 Remote control systems of the thrusters shall fulfil the relevant requirements specified in Chapter 14 of *Part VII – Electrical Installations and Control Systems*.

7.8 Monitoring

7.8.1 Indication system shall fulfil the requirements specified in sub-chapter 14.4.3 of *Part VII – Electrical Installations and Control Systems*.

7.8.2 Indication system shall clearly display, at every steering position, at least the following data:
– rotating direction and rotational speed for fixed pitch arrangements;
– pitch and rotational speed for controllable pitch arrangements;
– thrust angle.

7.8.3 Alarm system shall fulfil the requirements specified in sub-chapter 14.4.1 of *Part VII – Electrical Installations and Control Systems* and the requirements specified in Table 7.8.3. The alarm system of auxiliary devices with an installed power below 200 kW is subject to PRS consideration in each particular case.

Table 7.8.3
Alarm system of thrusters

Item	Component, installation, system	Parameter under control	Alarm system: Parameter value signalled	Notes
1.	Hydraulic drive of: – propeller, – device rotation, – propeller pitch change	level in spare hydraulic oil tank	minimum	–
2.		hydraulic oil pressure	minimum	–
3.		pressure difference in hydraulic oil filter	maximum	
4.		hydraulic oil temperature	maximum	if cooler is applied
5.	Lubricating oil system	oil pressure or oil level in gravity tank	minimum	
6.	Electrical motor drive of: – propeller, – device rotation, – propeller pitch change	load current short-circuit current	maximum minimum	depending on the adopted solution; drive stopping
7.	Thruster monitoring system	alarm system power supply	minimal	–
8.		control system power supply	minimal	–
9.		emergency stop means acc. to paragraph 7.2.10	emergency stop	–
10.	Thruster compartment	fire detection	fire	–
11.		water level in bilge well*	high level	–

* Alarm system giving warning of water penetration into the device casing shall be used where practicable.

7.9 Survey, Testing and Certificates

7.9.1 Thrusters for ships classed with PRS shall be type-approved by PRS.

7.9.2 After consideration of the technical documentation, PRS may accept application of a device that has approval certificate issued by other classification society or specialised national authority.

7.9.3 In the case of a single delivery of device, PRS may accept, after consideration of the technical documentation, application of a device, without a type-approval certificate.

7.9.4 Thrusters mentioned in paragraphs 7.9.1, 7.9.2 and 7.9.3 are subject to PRS survey both in production and testing in accordance with the requirements specified in paragraphs 7.9.6 and 7.9.7.

7.9.5 The scope of survey of auxiliary devices with the motors having power less than 200 kW is subject to PRS consideration in each particular case.

7.9.6 PRS survey of the production and testing of thrusters covers:
– checking of conformity of the applied materials and manufacturing procedures with the approved documentation,
– checking the conformity of workmanship with the approved documentation,
– testing of thrusters' installations including pressure tests of housings, piping and fittings as well as operating tests at the manufacturer's shop.

Factory operating tests shall be performed in accordance with the approved programme and in the presence of PRS surveyor.

Other tests and check procedures may be conducted by the manufacturer if it has been provided in the approved by PRS technical documentation of product type and the manufacturer's quality assurance system has been approved by PRS.

7.9.7 The survey covers materials used in the production – in accordance with paragraph 1.4.4.4, as well as welding, heat treatment and other procedures which are subject to agreement within the process of the classification documentation approval.

7.9.8 Any changes and nonconformities with the approved documentation of type shall be submitted, together with justification, to PRS for approval. Product testing shall start after the changes and nonconformities have been approved.

7.9.9 Pressure tests of casings shall be performed in accordance with the requirements specified in paragraph 1.5.3.1. In the case of hydrostatic pressure acting inside and/or outside the casing, working pressure p shall be taken for calculation as the highest hydrostatic pressure acting on one side in the lowest point of the casing.

7.9.10 Operating tests at the manufacturer's shop shall be performed on a test stand which allows the test to be performed at the rated rotational speed and full torque load on the shaft and column, if any. PRS may consider performance of some or all operating tests on board the ship.

Operating tests include:

.1 start and stop tests of the drive, and reversing tests;
.2 operation tests of thruster as a steering unit;
.3 tests of control systems.

7.9.11 After the operating tests, a lubricating oil sample shall be checked for traces of metallic and non-metallic particles.

7.9.12 After the operating tests, visual examinations of the whole thruster and, in justified cases, also internal examination shall be performed with particular regard to gearing.

7.9.13 The product testing is considered satisfactory if the test results comply with design data and if all tests acceptance criteria are fulfilled.

7.9.14 The certificate of thruster is issued by PRS after approval of the complete test report of the product. PRS reserves the right to issue the certificate after sea trials.

7.9.15 Sea trials of a thruster shall be performed in accordance with the approved programme.

The ability of the device to provide the propulsion and steering in all considered modes of sailing and manoeuvring shall be demonstrated during the sea trials.

Trials shall be performed at different operational ship speeds, various positions and power settings of the device and during rapid manoeuvring which starts of the most inconvenient combinations of ship speeds and position of the device.

7.9.16 In the case of devices installed on board the particular ship for the first time, PRS may request that measurements of the linear vibrations be performed.

7.9.17 During the trials of the monitoring systems, fulfilment of the requirements specified in sub-chapter 7.8 shall be demonstrated.

7.9.18 After the sea trials, PRS may request examination of the device in open condition.

7.9.19 After the sea trials, a sample of lubricating oil shall be checked for content of solid metallic and non-metallic particles.

7.9.20 PRS may require submission of the sea trials report for consideration.

8 TORSIONAL VIBRATIONS

8.1 General Requirements

8.1.1 The requirements specified in this Chapter apply to internal combustion engines intended for the ship main propulsion and for the drive of power generating sets. For diesel engines with the rated power below 55 kW, PRS may consider the requirement of submission of the calculations and/or measurements of torsional vibrations to be waived.

8.1.2 The scope and methodology of calculating the torsional vibrations of propulsion systems shall be such as to enable a complete analysis of the torsional vibration stresses to be expected in the main engine shafting system including its branches.

PRS shall be submitted, for consideration, with the calculations performed for both:
– normal operation,
– departures from normal operation due to irregularities in ignition. In this respect, the calculations shall assume operation with that cylinder without ignition whose malfunction might cause the most adverse dynamic loads.

An analysis of emergency modes of operation of the system (e.g. damper failure, flexible coupling failure, propeller blade break-off, etc.) which are considered by the design engineer the most likely and significant shall be performed. In justified cases, PRS may require that the results of such an analysis be submitted for examination.

Where modifications are introduced into the system which has a substantial effect on the torsional vibration characteristics, the calculation of the torsional vibration characteristics shall be repeated and submitted to PRS for consideration.

The torsional vibration stresses are the stresses to be added to the torsional stress resulting from mean torque at the considered engine speed and power output.

8.1.3 Calculations of torsional vibration shall include:
.1 input data:
  – mass moments of inertia and rigidity of particular components of a system;
  – logic diagrams of all the applicable modes of system operation;
  – type and rated parameters of the torsional vibration dampers, flexible couplings, transmission gears and generators – where applied;
.2 tables of successive forms of free vibrations with resonance within the range from $0.2n_z$ to $1.2n_z$, with their harmonics as specified in .3;
.3 firing order in the engine cylinders and the values of vector sum of the relative amplitudes of torsion angles of the cranks for all considered modes and harmonic orders within the range from 1 to 16 for two-stroke engines and from 0.5 to 12 for four-stroke engines;
.4 values of stresses caused by all significant harmonic excitation torques within the range from $0.2n_z$ to $1.05n_z$ for main engines, and from $0.5n_z$ to $1.1n_z$ for power generating set engines at the weakest cylindrical cross sections of the shafting;
.5 dynamic torques in flexible couplings and on the pinion of transmission gears within the speed range as specified in .4;
.6 for power generating sets – dynamic torques on the generator's rotor;

.7 vibration amplitudes taken at the assumed point of measurement (on the mass where measurements are taken), corresponding to the calculated values of the synthesised stresses and dynamic torques as specified in .4, .5 and .6;

.8 graphical and tabular presentation of dynamic loads and parameters of the torsional vibrations specified in items from .4 to .7. The graphs and tables shall include both combined values and the most significant harmonic compounds.

8.2 Allowable Stresses

8.2.1 Crankshafts

8.2.1.1 Combined torsional stresses during continuous operation of the engines shall not exceed those determined in accordance with the following formulae:

.1 within the range of crankshaft rpm:

$0.9n_Z \le n \le 1.05n_Z$ – for main engines,

$0.9n_Z \le n \le 1.10n_Z$ – for engines driving generators or other auxiliary machinery,

– where the maximum value of variable torsional stresses $\tau_{N\,max}$ has been determined by the crankshaft calculation method given in *Publication No. 8/P – Calculation of Crankshafts for Diesel Engines*

$$\tau_{1k} \le \pm\tau_{Nmax} \qquad (8.2.1.1.1\text{-}1)$$

– where the above-mentioned method has not been applied:

$$\tau_{2k} \le \pm 30.36 C_D \qquad (8.2.1.1.1\text{-}2)$$

.2 within the rpm ranges of crankshaft lower than those mentioned in .1, respectively:

$$\tau_{3k} \le \pm\frac{\tau_k[3-2\left(\frac{n}{n_Z}\right)^2]}{1.38} \qquad (8.2.1.1.2\text{-}1)$$

or

$$\tau_{4k} \le \pm 22C_D[3-2\left(\frac{n}{n_Z}\right)^2] \qquad (8.2.1.1.2\text{-}2)$$

where:

τ_{1k}, τ_{2k}, τ_{3k}, τ_{4k} – allowable stresses, [MPa];

C_D – size factor determined using the formula below:

$$C_D = 0.35 + 0.93d^{-0,2};$$

d – shaft diameter at the weakest section, [mm] $d = \min\,[d_{journal}, d_{crankpin}]$;

n – speed under consideration, [rpm];

n_z – rated speed, [rpm].

In the propulsion systems operated for prolonged periods of time with rated torque in the range of operational speed below the rated one (e.g. tug-boats and pushers), the stresses shall not exceed those determined in accordance with formula 8.2.1.1.1-1 or 8.2.1.1.1-2.

8.2.1.2 The combined torsional stresses for the barred speed ranges, which shall be passed quickly, shall not exceed the values determined in accordance with the following formula depending on the calculation method applied:

$$\tau_{1kz} = \pm 1.9\tau_{3k} \qquad (8.2.1.2\text{-}1)$$

or

$$\tau_{2kz} = \pm 1.9\tau_{4k} \qquad (8.2.1.2\text{-}2)$$

where:

τ_{1kz} and τ_{2kz} – permissible stress for quick passing through the barred range, [MPa];

τ_{3k} and τ_{4k} – see paragraph 8.2.1.1.

8.2.2 Intermediate, Thrust, Propeller and Generator Shafts

8.2.2.1 The combined torsional stresses for continuous operation shall not exceed, in any part of the shaft, the values determined in accordance with the following formulae:

.1 within the range of shaft rpm:

$0.9n_Z \le n \le 1.05n_Z$ – for intermediate, thrust, propeller and shafts,

$0.9n_Z \le n \le 1.10n_Z$ – for generator shafts

$$\tau_{1w} = \pm 1.38 C_w C_k C_D \tag{8.2.2.1.1}$$

.2 within the rpm ranges lower than specified in .1:

$$\tau_{2w} = C_w C_k C_D \left[3 - 2\left(\frac{n}{n_Z}\right)^2\right] \tag{8.2.2.1.2}$$

where:

τ_{1w}, τ_{2w} – allowable stresses, [MPa];

C_w – material coefficient, $C_w = \frac{R_m+160}{18} \le 42.2$ (R_m > 600 MPa shall not be taken into account);

R_m – shaft material tensile strength, [MPa];

C_k – shaft design coefficient,
= 1.0 for intermediate shafts and generator shafts with flanges forged together with shaft,
= 0.6 for intermediate shafts and generator shafts in way of keyways,
= 0.85 for the thrust shaft parts specified in sub-chapter 4.4,
= 0.55 for the propeller shaft parts for which the coefficient value 1.22 or 1.26 shall be taken, in accordance with paragraph 4.5.1;

C_D, n, n_z – see paragraph 8.2.1.1.2.

In the propulsion systems operated for prolonged periods of time with the rated torque at speeds below the rated one (e.g. tugboats, fishing trawlers, etc.), the stresses shall not exceed those determined in accordance with formula 8.2.2.1.1.

8.2.2.2 The combined torsional stresses for the barred speed ranges, which shall be passed quickly, shall not exceed the value determined in accordance with the following formula:

$$\tau_{wz} = \pm \frac{1.7\tau_{2w}}{\sqrt{C_k}} \tag{8.2.2.2}$$

where:

τ_{wz} – allowable stress for quick transition through the barred range, [MPa].

For other symbols – see paragraph 8.2.2.1.

8.2.2.3 The stress values defined in paragraphs 8.2.2.1 and 8.2.2.2 refer to the shafts with diameters equal to those specified in 4.2, 4.4 or 4.5. Where actual diameters of the shafts are greater than required, PRS may accept higher values of the torsional vibration stresses.

PRS may accept stresses exceeding those specified in 8.2.2.1 and 8.2.2.2 where justified by calculation.

8.2.3 Allowable Dynamic Torques

8.2.3.1 Dynamic moments in flexible couplings and vibration dampers shall not exceed the values specified by the manufacturer.

8.2.3.2 It is recommended that the dynamic torques occurring in any stage of a transmission gear do not exceed 1/3 of the rated torque within the rpm range from $0.9n_z$ to $1.05n_z$.

8.2.3.3 Dynamic moments occurring in generator rotor shall not exceed the values specified by the manufacturer – depending on the employed construction of connection with the generator shaft.

8.3 Measurements of Torsional Vibration Parameters

8.3.1 The results of calculation of combined torsional vibration stresses shall be confirmed by measurements taken on the first ship of the series. When estimating these stresses, their harmonic analysis shall be done.

8.3.2 The measured frequencies of free vibrations shall not differ from the calculated values by more than 5%. Where this requirement is not fulfilled – the calculations shall be corrected accordingly.

8.3.3 Where, as a result of calculations, it is not necessary to apply barred speed ranges, or in other justified cases, PRS may allow taking measurements to be waived.

8.4 Barred Speed Ranges

8.4.1 Where the combined actual torsional stresses exceed the permissible values for continuous operation, the barred speed ranges shall be determined. The barred speed ranges shall not occur within the following ranges:
– $n \geq 0.8n_z$ – for ship propulsion system,
– $n \geq 0.85n_z$ – for power generating sets.

8.4.2 In the case of exceeding the permissible stresses due to resonance, the barred speed range shall be determined in accordance with the following formula:

$$\frac{16n_k}{18-\frac{n_k}{n_z}} \leq n \leq \frac{(18-\frac{n_k}{n_z})n_k}{16} \qquad (8.4.2)$$

where:

n – barred speed range, [rpm];
n_k – resonance speed, [rpm];
n_z – rated speed, [rpm].

8.4.3 The limits of barred speed may also be determined by extending by $0.03n_z$ to both sides the range within which the combined torsional vibration stresses or torques in the flexible couplings or transmission gear exceed the permissible values.

8.4.4 Where normal operation of the engine is accompanied by calculated, and confirmed by measurements, speed ranges in which the combined stresses or dynamic torques in couplings or in transmission gears exceed the permissible values, then the ranges of barred speed shall be marked in accordance with paragraph 1.13.6. Proper warning plates shall be located at the engine control stations.

8.4.5 If during the engine operation with one cylinder without ignition (see paragraph 8.1.2) the stresses and torques defined in paragraph 4.4.4 exceed the allowable values, then:

.1 the engine shall be provided with an automatic alarm system, indicating the lack of ignition in a cylinder, and the engine control stations shall be fitted with the plates indicating the barred speed ranges, determined in accordance with paragraph 8.4.2 or 8.4.3 for such a condition of engine;

.2 where the alarm system defined in .1 is not provided, the additional barred speed ranges for the engine operation with one cylinder without ignition shall be marked on the tachometers and warning plates.

9 AUXILIARY MACHINERY

9.1 Power-driven Air Compressors

9.1.1 General Requirements

9.1.1.1 Compressors shall be so designed that the air temperature at the air cooler outlet does not exceed 90°C.

9.1.1.2 Each compressor stage or stub pipe at the immediate outlet from the compressor stage shall be fitted with safety valve preventing the pressure rise in the stage above 1.1 times the rated pressure when the delivery pipe valve is closed.

The construction of safety valve shall preclude the possibility of its adjustment or disconnection after being fitted on the compressor.

9.1.1.3 Delivery stub pipe or the immediate outlet of compressor shall be fitted with a fuse or an alarm with the activation temperature not exceeding 120°C.

9.1.1.4 Bodies of coolers shall be fitted with safety devices ensuring a free outlet of air in case of the pipes' breakage.

9.1.2 Crankshaft

9.1.2.1 The method of verifying calculations specified in paragraphs 9.1.2.3 and 9.1.2.4 applies to the steel crankshafts of naval air compressors with in-line, and V-shaped arrangement of cylinders with single and multi-stage compression.

9.1.2.2 Crankshafts shall be made of steel having tensile strength R_m ranging from 410 to 780 MPa.

The use of steel having a tensile strength over 780 MPa is subject to PRS consideration in each particular case.

Crankshafts may be made of nodular cast iron with a tensile strength 500 ≤ R_m ≤ 700 MPa as required in Chapter 15 of *Part IX – Materials and Welding* of the *Rules for the Classification and Construction of Sea-going Ships*. Crankshafts with other dimensions than those determined by the formulae given below may be applied subject to PRS consent in each particular case, provided that complete strength calculations are submitted.

9.1.2.3 Crank pin diameter d_k of the compressor shall not be less than determined in accordance with the formula below:

$$d_k = 0.25K\sqrt[3]{D^2 p\sqrt{0.3L^2 f + (S\varphi)^2}} \ \ [\text{mm}] \qquad (9.1.2.3\text{-}1)$$

D – design diameter of cylinder, [mm], equal to:
– for single-stage compression
$D = D_C$ (D_C – cylinder diameter),
– for two- and multi-stage compression in separate cylinders,
$D = D_W$ (D_W – high pressure cylinder diameter),
– for two-stage compression by a differential piston

$D = 1.4\ D_W$,
– for two-stage compression by a differential piston
$D = \sqrt{D_n^2 - D_W^2}$ (D_n – low pressure cylinder diameter);

p – compression pressure in high pressure cylinder, [MPa];
L – design distance between main bearings, [mm], equal to:
– where one crank is arranged between two main bearings $L = L'$
– (L' – actual distance between centres of main bearings);
– where two cranks with 180° angle are arranged between two main bearings L = 1,1 L';
S – piston stroke, [mm];
K, f, φ – coefficients determined in accordance with Tables 9.1.2.3-1, 9.1.2.3-2 and 9.1.2.3-3.

Table 9.1.2.3-1
Values of coefficient *K*

Tensile strength, [MPa]	390	490	590	690	780	880
K	1.43	1.35	1.28	1.23	1.2	1.18

Table 9.1.2.3-2
Values of coefficient *f*

Angle between cylinder axes	0° (in line)	45°	60°	90°
f	1.0	2.9	1.96	1.21

Table 9.1.2.3-3
Values of coefficient φ

Number of cylinders	1	2	4	6	8
φ_1	1.0	1.1	1.2	1.3	1.4

If crankshaft journals have co-axial holes with diameters exceeding 0.4 d_k, then the journal diameters shall be determined in accordance with the formula below:

$$d_{k0} \geq d_k \sqrt[3]{\frac{1}{1-(\frac{d_0}{d_a})^4}} \quad \text{[mm]} \qquad (9.1.2.3\text{-}2)$$

d_k – see formula 9.1.2.3-1;
d_0 – co-axial hole diameter, [mm];
d_a – actual diameter of shaft.

Edges of oil holes on journal surfaces shall be rounded to a radius not less than 0.25 times the hole diameter with a smooth finish.

9.1.2.4 Thickness of the crank web h_k shall not be less than that determined in accordance with the formula below:

$$h_k = 0.105 K_1 D \sqrt{\frac{(\psi_1 \psi_2 + 0.4) P C_1 f_1}{b}} \quad \text{[mm]} \qquad (9.1.2.4\text{-}1)$$

K_1 – coefficient taking into account the effect of shaft material and determined in accordance with the formula below:

$$K_1 = a \sqrt[3]{\frac{R_m}{2R_m - 430}} \qquad (9.1.2.4\text{-}2)$$

a = 0.9 for shafts with entire surface nitrided or subjected to other kind of heat treatment accepted by PRS,

a = 0.95 for die forged shafts with the fibre continuity being maintained,
a = 1 for shafts not subjected to quenching and tempering;
ψ_1 and ψ_2 – coefficients determined in accordance with Tables 9.1.2.4-1 and 9.1.2.4-2;
P – compression pressure taken in accordance with the relevant provisions of paragraph 9.1.2.3;
C_1 – distance from the centre of the main bearing to the midplane of the crank web, [mm]; where two cranks are arranged between two main bearings, the distance to the midplane of the web located further from the support under consideration shall be taken;
b – breadth of crank web, [mm];
f_1 – coefficient taken in accordance with Table 9.1.2.4-3;
R_m – tensile strength, [MPa].

Table 9.1.2.4-1
Values of coefficient ψ_1

r/h_k \ ε/h_k	0	0.2	0.4	0.6	0.8	1.0	1.2
0.07	4.5	4.5	4.28	4.1	3.7	3.3	2.75
0.10	3.5	3.5	3.34	3.18	2.85	2.57	2.18
0.15	2.9	2.9	2.82	2.65	2.4	2.07	1.83
0.20	2.5	2.5	2.41	2.32	2.06	1.79	1.61
0.25	2.3	2.3	2.2	2.1	1.9	1.7	1.4

Note:
r – fillet radius of the transition from crank web to crank pin, [mm];
ε – value of overlap, [mm].
For crankshafts without journals' overlap, coefficient ψ_1 shall be taken as for ε/h_k = 0.

Table 9.1.2.4-2
Values of coefficient ψ_2

b/d_k	1.2	1.4	1.5	1.8	2.0	2.2
ψ_2	0.92	0.95	1.0	1.08	1.15	1.27

d_k – see formula 9.1.2.3-1.

Intermediate values of the coefficients specified in Tables 9.1.2.4-1 and 9.1.2.4-2 shall be determined by linear interpolation.

Table 9.1.2.4-3
Values of coefficient f_1

Angle between cylinder axes	0° (in line)	45°	60°	90°
f_1	1.0	1.7	1.4	1.1

9.1.2.5 The radius of fillet of the crank pin and crank web shall not be less than 0.05 the crank pin diameter.

The radius of fillet of the crank pin and the coupling flange shall not be less than 0.08 the crank pin diameter.

Surface hardening of crank pins and journals shall not be applied to fillets, except when the entire shaft has been subjected to hardening.

9.2 Pumps

9.2.1 General Requirements

9.2.1.1 Unless the pumped liquid is used for the lubrication of bearings, provision shall be made to prevent the pumped liquid from penetration into the bearings.

9.2.1.2 It is recommended that the pump sealing on the suction side be fitted with hydraulic seals.

9.2.1.3 Where the pump construction enables the rise of pressure above the rated value, a safety valve shall be fitted on the pump casing or on the delivery pipe before the first stop valve.

9.2.1.4 In pumps intended for transferring inflammable liquids, an outlet pipe from safety valve shall be connected to the pump suction side.

9.2.1.5 Provision shall be made to prevent water hammer. Application of overflow valves for this purpose is not recommended.

9.2.1.6 Strength Calculation

Critical speed of pump impeller shall not be less than 1.3 of the rated r.p.m.

9.2.1.7 Self-priming pumps

Self-priming pumps shall ensure operation under "dry-suction" conditions and it is recommended that they be fitted with arrangements preventing the self-priming device against being damaged as a result of impure water pumping.

9.2.2 Additional Requirements for Flammable Liquid Pumps

9.2.2.1 Pump seals shall be of such construction and materials, that no vapour/air explosive mixture can be generated in case of leakage.

9.2.2.2 The construction of dynamic seals shall prevent the possibility of overheating and self-ignition of seals due to friction of the moving elements.

9.2.2.3 The construction of pumps made of low electrical conductivity materials (plastics, rubber, etc.), shall prevent accumulation of electrostatic charges, or special means for electric charge neutralisation shall be provided.

9.3 Fans, Air Blowers and Turboblowers

9.3.1 General Requirements

9.3.1.1 The requirements specified in this sub-chapter apply to fans intended for systems covered by requirements of *Part VI*, as well as to internal combustion engine turbo-blowers.

9.3.1.2 Impellers of fans and air blowers, including couplings, as well as the assembled rotors of turbochargers shall be dynamically balanced together with the couplings in accordance with the requirements specified in paragraph 6.1.2.

9.3.1.3 Suction ports shall be protected against the entry of incidental solids.

9.3.1.4 Lubrication system of the turbo-blower bearings shall prevent the possibility of penetration of oil into the supercharging air.

9.3.1.5 Strength Calculations

The impeller parts shall be so designed that the equivalent stress at any section will not exceed 0.95 of the material yield point at rotational speed equal to 1.3 of the rated speed.

For turbo-blowers, other safety factors may be applied subject to PRS consent in each particular case, provided that calculation methods determining the maximum local stress or elastoplastic methods have been used.

9.3.2 Additional Requirements for Fans Intended for Transferring Flammable Vapours and Explosive Air Petrol and Mixtures

9.3.2.1 The air gap between the casing and rotor shall not be less than 0.1 of the rotor shaft bearing journal diameter and not less than 2 mm, but it is not required for the air gap to be greater than 13 mm.

9.3.2.2 Terminals of ventilation ducts shall be protected from entering of foreign matters into the fan casings by means of wire net, with square net mesh of the side length not exceeding 13 mm.

9.3.2.3 Ventilation fans shall be of non-sparking design. The fan is not sparking if in both normal and abnormal conditions there is no risk of spark generation. Casing and rotating parts of fan shall be made of such materials, which do not cause electric charge accumulation, and the fans installed shall be properly earthed to the hull of ship in accordance with the requirements specified in *Part VII – Electrical Installations and Control Systems*.

9.3.2.4 Except the cases specified in paragraph 9.3.2.5, rotors and fan casings in way of rotor shall be made of such materials which do not generate sparks, as confirmed by adequate tests.

9.3.2.5 The tests mentioned in paragraph 9.3.2.4 may be waived for fans made of the following combinations of materials:

- **.1** rotor and/or casing made of non-metallic materials with anti-electrostatic properties,
- **.2** rotor and casing made of non-ferrous metal alloys,
- **.3** rotor made of aluminium or magnesium alloy and steel casing (including stainless austenitic steel), where a ring made of non-ferrous material of adequate thickness is used inside the casing in way of rotor,
- **.4** any combination of steel rotor and casing (including stainless austenitic steel) provided that the radial clearance between them is not less than 13 mm.

9.3.2.6 Rotors and fan casings made of the following materials are considered as sparking and their application is not permitted:

- **.1** rotor made of an aluminium or magnesium alloy and steel casing, irrespective of the radial clearance value,
- **.2** casing made of an aluminium or magnesium alloy and steel rotor, irrespective of the radial clearance value,
- **.3** any combination of rotor and casing made of steel with the design radial clearance less than 13 mm.

10 DECK MACHINERY

10.1 General Requirements

10.1.1 Deck machinery shall be designed for the service in conditions specified in sub-chapter 1.9.

10.1.2 Brake linings and their fixing arrangements shall be resistant to sea water and oil as well as heat resistant at temperatures up to 250°C.

Heat resistance of the brake lining connection to the brake structure shall be greater than for the temperature which may occur in combination of any working conditions of the mechanism.

10.1.3 Machinery items which are both manually-operated and power-driven shall be provided with interlocking arrangements precluding simultaneous operation of these drives.

10.1.4 It is recommended that the deck machinery controls be so arranged that lifting will be performed by rotating the handwheel clockwise or by moving the lever backwards, whereas descending – by rotating the hand wheel counter clockwise or by moving the lever forwards. Braking shall be performed by rotating the hand wheel clockwise, whereas brake releasing – anti-clockwise.

10.1.5 Measurement and control instruments and gauges shall be so located as to be capable of being watched from the control station.

10.1.6 The machinery with hydraulic drive or control shall also fulfil the requirements specified in Chapter 11.

10.1.7 Winch drums on which ropes are put in several layers and subjected to load shall have flanges extending beyond the external layer of winding by not less than 2.5 times the rope diameter.

10.2 Steering Gears and Their Installation on Board Ship

10.2.1 General Requirements

10.2.1.1 The main steering gear[*)] shall enable putting the rudder by 35° to each side and putting the rudder over from 35° on either side to 30° on the other side in not more than 28 seconds, with the steering gear rated torque applied to the rudder stock.

The main steering gear design shall be capable of taking the load resulting from the ship motion ”full astern”, this however need not be confirmed by the sea trials.

10.2.1.2 The auxiliary steering gear[*)] shall enable putting the rudder by 15° to each side and putting the rudder over within this range in a time not exceeding 60 seconds with rated torque of this gear applied to the rudder stock.

The auxiliary steering gear shall be so arranged as to be capable of being brought into action within no more than 2 minutes in case of failure of the main steering gear.

10.2.1.3 The main steering gear and auxiliary steering gear shall be so arranged that a failure in one of them will not render the other one inoperative.

In steering gears with single actuator, the shut-off valves of hydraulic tubing shall be fitted directly on the actuator.

10.2.1.4 Rated torque M_{ZN} of steering gear is the rudder stock torque at the following rudder angle:

35° – for main steering gear,

[*)] For definition of the main steering gear – see 1.2 of *Part III – Hull Equipment*.

15° – for auxiliary steering gear,

at rated parameters of steering gear power units*).

10.2.1.5 Where at least two identical power units are applied for the rudder stock, the steering gear shall be so arranged that in the event of failure of a single component of the piping system or a power unit means shall be provided to isolate the damaged system or unit for quick regaining of control of the steering system.

10.2.1.6 Hydraulic steering gear with mechanical drive shall be provided with:

.1 device for keeping the hydraulic oil clean adequate to type and design of the hydraulic system;

.2 low level alarm of hydraulic oil in each circulating tank (visual and audible alarm signals shall be received in the wheelhouse and engine room).

A spare tank of hydraulic oil shall be provided, the capacity of which shall be sufficient for filling at least one of the power units, including the circulation tank. The spare oil tank shall be equipped with a gauge for measuring the tank content and permanently connected to hydraulic pipe in a way which allows for easy filling of the hydraulic system from a post in steering gear compartment.

10.2.1.7 Each part of hydraulic power system, that can be separated from the system and subjected to load from the drive source or by external forces (caused by water pressure exerted on the rudder blade) shall be fitted with relief valves set to a pressure not exceeding the design pressure, but not less than 1.25 times the rated pressure of the system. The minimum output of relief valve(s) shall not be less than 1.1 of total capacity of pumps connected to it. In no case, the pressure rise shall exceed 1.1 times the setting of the relief valves, the change of oil viscosity in extreme ambient conditions shall be taken into account. Means of sealing with lead shall be provided for the relief valves.

It is recommended that the following tests of the relief valves be performed:

– output (throughput),
– resistance to water (hydraulic) hammer.

10.2.1.8 Oil tight seals separating spaces under pressure shall be:

– made with metallic contact or equivalent between parts reciprocally fixed,
– doubled between reciprocally movable parts so as to prevent abrupt drop of pressure in the system in case of one of the seals being damaged; PRS may approve an alternative solution ensuring equivalent protection against leakage.

10.2.1.9 Rudder Position Indicators

Rudder gear parts rigidly coupled with the rudder stock (tiller, quadrant, etc.) shall be fitted with a dial, calibrated for accuracy not less than 1°, to indicate the position of the rudder related to ship's centre line.

10.2.1.10 Limit Switches

Each steering gear shall be provided with an arrangement for stopping its operation before the rudder reaches its limit switches, permanently fixed to the ship hull; the steering gear capability to move the rudder immediately in the opposite direction shall be maintained.

10.2.1.11 The steering gear shall be fitted with a brake or any other device providing to keep the rudder steady at any position when the latter exerts the design torque (without taking into account the friction in the rudder stock bearings).

For hydraulic steering gears, which can be kept steady by closing valves on oil lines, the special braking device may be omitted.

10.2.1.12 The requirements concerning the electric drive and signalling are specified in sub-chapter 5.2 *Part VII – Electrical Installations and Control Systems*.

10.2.1.13 Operating instructions, including the block diagram and switching-over procedures for control systems, power units and hydraulic cylinders of steering gear, shall be posted in permanence and at well visible places in the wheelhouse and steering gear compartment.

Where applicable the below given standard warning label shall be fitted at steering post in the wheelhouse or the statement ought to be included into ship "Procedures manual".

Warning label in Polish:

UWAGA:
Gdy oba zespoły energetyczne maszyny sterowej pracują jednocześnie, w pewnych warunkach ster może nie reagować na zadane polecenie. Należy wówczas wyłączać kolejno pompy, aż kontrola nad sterem zostanie przywrócona.

Warning label in English

CAUTION:
In some circumstances when 2 power units are running simultaneously, the rudder may not respond to helm. In that case, stop each pump in turn until the control is resumed.

The above mentioned label concerns the steering gears equipped with two identical power generators prepared for simultaneous operation and generally fitted with own, separate control system or two separate control circuits able to work simultaneously.

10.2.2 Materials and Manufacturing of Hydraulic Systems

10.2.2.1 Hydraulic cylinder pressure casings, power hydraulics valves, flanges and fittings of pipelines, as well as all parts transmitting forces to the rudder stock (rudder quadrant, tiller, etc.) shall be made of steel or other PRS approved ductile material. The ultimate elongation A_5 of such materials shall be, as a rule, not less than 12%, while their tensile strength shall not exceed 650 MPa. Upon special agreement with PRS, grey cast iron may be used for doubled, slightly loaded parts.

10.2.2.2 Pipes of the hydraulic steering systems shall fulfil the requirements relevant to class I piping and flexible joints specified in sub-chapter 15.1.

10.2.2.3 Piping shall be so made as to enable easy switching on and off individual cylinders and units and shall additionally fulfil the requirements specified in Chapter 11.

The possibility of bleeding air from the pipelines shall be provided, where necessary.

10.2.2.4 Hydraulic steering gear pumps shall be provided with protective means to prevent reverse rotation of an inoperative pump or with automatic arrangements to shut off the flow of liquid through the inoperative pump.

10.2.2.5 Where simultaneous operation of more than one steering gear or power unit is provided, then the risk of hydraulic lock shall be taken into account in the case of failure of singular power unit or control system.

If such a risk cannot be eliminated, a visual and audible alarm shall be provided in the wheelhouse for warning against the loss of steering capability and for identification of damaged system.

Appropriate instructions for switching off the damaged system shall also be displayed in the wheelhouse.

Such an alarm shall be activated (for instance) if:
– setting of variable capacity pump is different than the value set by the control system,
– three-way, full flow valve or a similar device of fixed capacity pump is in wrong position.

10.2.3 Construction and Strength Calculation

10.2.3.1 Steering devices shall be so designed as to reduce the local concentration of stress as far as possible.

Welded parts and welding procedure are subject to PRS approval. All welded joints within hydraulic cylinders or interconnected parts situated in line with the force flux, shall, as a rule, be made with full penetration of weld.

10.2.3.2 The parts of main and auxiliary steering gear situated in line with the force flux shall be checked by calculations for strength when affected by loads corresponding to design torque M_s (see sub-chapter 2.5.4 of *Part III – Hull Equipment*); pipelines and other parts submitted to the inner pressure shall be checked for the load corresponding to the design pressure.

The design pressure shall not be less than the greater of the below values:
– 1.25 of rated pressure (i.e. this corresponding to M_{ZN} torque), or
– assumed safety valve setting.

10.2.3.3 Casings of steering gear actuators and hydraulic batteries shall fulfil the requirements for pressure vessels of Class I specified in Chapter 14.

10.2.3.4 The stresses in a considered part shall not exceed the following values, whichever less:

$$R_m/A \text{ or } R_e/B$$

where:

R_m – tensile strength, [MPa];
R_e – physical yield point or proof stress ($R_{0.2}$), [MPa].

The values of safety factors A and B are specified in Table 10.2.3.4.

Table 10.2.3.4

Factor	Steel	Cast steel	Nodular iron
A	3.5	4	5
B	1.7	2	3

PRS may require fatigue strength calculations accounting for fatigue of materials caused by pressure pulsation in hydraulic system.

10.2.3.5 The parts of steering gear which are situated in line of the force flux not protected against overload by means of limiters fastened to the ship hull (see sub-chapter 2.13 of *Part III – Hull Equipment*) shall have strength not lower than that of the rudder stock.

10.2.4 Connection to Rudder Stock

10.2.4.1 Connection of the steering gear to the elements rigidly fixed to the rudder stock shall be such as to preclude the steering gear damage due to axial displacement of the rudder stock.

10.2.4.2 The connection of the tiller, quadrant or yoke with the rudderstock shall be calculated for transmission of torque not less than $2M_s$ (see sub-chapter 2.5.4 of *Part III – Hull Equipment).*

For one-piece hubs, fastened by shrink fitting to the rudderstock, the friction factor not exceeding 0.13 shall be taken. The split hubs shall be fastened by at least two bolts at each side and shall have:

– two keys designed for transmission of torque not less than $2M_s$, if the friction is not taken into account;
– single key, if the bolt tension is designed for friction transmission of torque not less than $2M_s$.

10.2.5 Hand Operated Steering Gear

10.2.5.1 Main steering gear shall be of self-locking type. The auxiliary steering gear shall also be of self-locking type or may be fitted with device locking the gear in desired position, provided that there is a possibility to change this position.

10.2.5.2 Hand-operated main steering gear shall fulfil the requirements specified in paragraph 10.2.1.1 – when handled by one man with a force not exceeding 120 N applied to the steering wheel handles and with the number of revolutions not greater than $9/R$ when putting the rudder from hard over to hard over (R – radius of steering wheel handle measured from the wheel axis of rotation to the mid-length of the handle, [m]).

10.2.5.3 Hand-operated auxiliary steering gear shall fulfil the requirements specified in paragraph 10.2.1.2 when handled by not more than two men with a force not exceeding 150 N per helmsman, applied to steering gear handles.

10.2.5.4 For hand-operated main steering gear it is sufficient to provide the gear with buffer springs instead of protection against overload required in paragraph 10.2.1.7.

For the hand-operated auxiliary steering gear, the requirements specified in paragraph 10.2.1.7 need not be fulfilled.

10.2.6 Pump Type Test

The pumps of hydraulic power units shall be subjected to a type test. The test duration shall be 100 hours at least. The test stand shall be arranged for idle running of the pump, as well as for the pump operation with maximum capacity at maximum working pressure. The idle running test periods shall be performed alternately with the periods of full load operation. The transition from one operating condition into the other one shall be performed at least as quickly as during the operation on board the ship. No abnormal heating, vibration or other irregularities of pump operation can occur throughout the test duration. Upon completion of the test the pump shall be dismantled and its parts subjected to inspection.

This test may be omitted for the power units, for which the reliability has been confirmed by operational trials during service of the ship.

10.2.7 Tests on Board Ship

10.2.7.1 Steering gear shall be subjected to tightness and operating tests after its installation on board the ship.

10.2.7.2 The scope of sea trials witnessed by PRS surveyor shall include:

.1 checking compliance with the requirements specified in paragraphs 10.2.1.1 and 10.2.1.2 regarding the rudder deflection by main and stand-by steering gear. In the case of controlled pitch propeller, the pitch shall be set to the maximum value for nominal engine rotation speed full ahead.

Where trials are not possible with maximum ship draught, PRS may permit other trial conditions;

.2 power units of the steering gear testing and their switching on/off;

.3 switching off and cutting off of the working power unit, check of time to recover the steering abilities;

.4 hydraulic oil filling-in system check;

.5 control system operation, including control command transfer and local steering;

.6 checking the means of communication between wheelhouse, machinery room and steering gear compartment;

.7 operation test of the alarm system and indicators in accordance with the requirements specified in paragraphs 10.2.2.5 as well as in sub-chapter 8.2.8 of *Part VII – Electrical Installations and Control Systems*;

.8 checking, where applicable, that no hydraulic interlocking (hydraulic lock) occurs and signalling system check.

10.3 Windlasses

10.3.1 Drive

10.3.1.1 Power of the windlass motor shall ensure continuous heaving up a chain cable with an anchor of normal holding force for at least 30 minutes with a speed at least 9 m/min (0.15 m/s) and chain cable pull P on the cable lifter not less than that determined in accordance with the formula below:

– for all types of ships, except supply vessels

$$P = 9.81\, ad^2 \quad [\mathrm{N}] \tag{10.3.1.1-1}$$

where:

a – coefficient taking the following values:
3.75 for steel grade 1 chain cables,
4.25 for steel grade 2 chain cables,
4.75 for steel grade 3 chain cables,

(for chain cable steel grades – see the *Rules for the Classification and Construction of Sea-going Ships*, Chapter 11 of *Part IX – Materials and Welding*);

d – chain cable diameter, [mm].

For chain cables of less than 28 mm in diameter, the value of factor a may be reduced subject to PRS consent in each particular case.

Mean speed of the chain cable heaving-in shall be measured over 2 chain lengths, beginning with the moment when 3 chain lengths are freely hanging down.

10.3.1.2 Windlass drive shall provide the speed of pulling the anchor into the hawse pipe not exceeding 0.15 m/s. It is recommended that this speed not exceed 0.12 m/s.

10.3.1.3 To extract the anchor from the bottom, the windlass power unit shall produce, in a rated working cycle, a continuous pull of one cable lifter equal at least 1.5P for a period not less than 2 minutes. However, the requirement specified in paragraph 10.3.1.1 concerning the heave-up speed need not be fulfilled.

10.3.2 Brakes and Disengaging Clutches

10.3.2.1 Windlasses shall be fitted with disengaging clutches between the cable lifter and the drive shaft.

Windlass with a gear mechanism which is not of self-locking type shall be fitted with automatic cable lifter brakes to prevent paying out of the chain in case of the power failure or power unit failure.

The automatic cable lifter brake shall be capable of maintaining the cable lifter pull not less than 1.3*P*.

10.3.2.2 Cable lifters shall be fitted with brakes which are capable of stopping safely paying out of the chain. This brake shall ensure holding the chain cable without slip on the brake when the cable lifter is declutched and the chain cable loaded with a force:

.1 equal to 0.45 times the breaking load of the cable – for anchor gear with a stopper for holding
the chain cable of a ship lying at anchor;

.2 equal to 0.8 times the breaking load of the cable – for anchor gear without the stopper mentioned in .1 .

The force applied to the brake drive handle shall not exceed 740 N.

10.3.3 Cable Lifters

10.3.3.1 Cable lifters shall have at least than five cams. For horizontal axis cable lifters, the wrapping angle shall not be less than 115°, whereas for vertical axis cable lifters – not less than 150°.

10.3.3.2 Cable lifters shall be so designed that the detachable links (Kenter links) can pass both in horizontal and vertical position.

10.3.4 Overload Protection

Where the maximum torque of the windlass motor may cause the (equivalent) stress in the windlass components exceeding 0.95 the yield point of the material used, or a rise to the force on the sprocket exceeding 0.5 the test load, a safety coupling shall be installed between the motor and the windlass to prevent overload.

10.3.5 Strength Calculation

Stress of the windlass parts being in the flux of strain lines shall not exceed:

0.4 R_e – when loaded with rated power of driving motor,
0.95 R_e – when loaded with the maximum torque of driving motor,
0.95 R_e – when subjected to maximum load caused by anchor cable held by brake – in accordance with 10.3.2.2; this requirement applies to those parts of windlass which are subjected to the above mentioned load;

(R_e – yield point of material of the parts in question).

When designing windlasses, special attention shall be paid to:
- notch stress concentration,
- dynamic loads caused by abrupt start or stop of driving motor,
- calculation methods and approximations applied for finding stress value and cycle,
- effective fixing of the windlass to the foundation.

10.3.6 Additional Requirements for Windlasses with Remote Control

10.3.6.1 Windlasses with remote control shall be fitted with an automatic brake so that the speed of chain cable release, with the chain sprocket disengaged from the drive, does not exceed 3 m/s and is not less than 1.33 m/s, except the initial run.

In ships with the equipment number 400 and less, such an automatic brake is not required.

10.3.6.2 Sprocket brake shall ensure smooth stopping of chain cable within not more than 5 s and not less than 2 s from the moment of control station command.

10.3.6.3 Remote control station shall be fitted with the released chain length indicator and releasing speed indicator – maximum permissible speed 3 m/s shall be marked on the indicator.

10.3.6.4 Remotely controlled windlasses shall be fitted with local manual control posts. In any case of remote control failure the possibility of local control shall be maintained.

10.4 Mooring Winches

10.4.1 Drive

10.4.1.1 Mooring winch motor shall ensure uninterrupted heaving-in of a mooring line at a rated pull for a period of not less than 30 minutes.

The heaving-in speed of the mooring line when reeling the first layer on the drum, with the rated pull, shall be at least:
up to 80 kN – 0.25 m/s,
from 81 to 160 kN – 0.20 m/s,
from 161 to 250 kN – 0.16 m/s,
above 250 kN – 0.13 m/s.

The speed of heaving-in the mooring line by mooring head shall not exceed 0.3 m/s at the rated load.

Guidelines for the rated pull selection are contained in *Part III – Hull Equipment*.

10.4.1.2 Mooring winch power transmission system shall ensure, in the rated working cycle, a continuous pull in the reeled first layer line not less than 1.5 times the rated pull being maintained for at least 2 minutes.

The pull in the line designed for the mooring winch caused by the maximum torque shall not exceed 0.8 of the line breaking load.

10.4.1.3 Where the maximum torque may cause a load in the mooring winch components exceeding that specified in sub-chapter 10.4.3, overload protection shall be provided.

10.4.2 Brakes

10.4.2.1 Mooring winch shall be provided with an automatic braking device holding the mooring line under tension of not less than 1.5 times the rated pull in case of the power loss or drive failure.

10.4.2.2 Mooring winch drum shall be provided with a brake, whose braking torque will prevent unreeling of the mooring line under the tension equal to 0.8 times the breaking load of the line reeled first layer.

The force applied to the brake handle to induce such torque shall not exceed 740 N.

Where the winch drum is fitted with a pawl and ratchet or other locking device, the braking device shall be capable of the winch drum smooth release while the mooring line is tense.

10.4.3 Strength Calculation

10.4.3.1 Stresses in the parts fixing the mooring winch to the foundation and in load-bearing parts of the winch under mooring line breaking load exerted on the drum and on the mooring head at its mid-length shall not exceed 0.95 times the yield strength of their material.

Stresses in the winch components shall be determined taking account of all the possible types and geometrical directions of loads likely to occur in the service conditions.

10.4.3.2 Strength characteristics of the line intended for working with the mooring equipment shall be indicated on the mechanism.

10.4.4 Additional Requirements for Mooring Winches with Pulling Force Automatic Control

10.4.4.1 Mooring winches with automatic control of the pulling force shall be provided with:
– indicator of the actual pull in the mooring line during the winch operation with the automatic control of pull;
– device for automatic releasing of the mooring line to maintain pulling force between 1.05 and 1.5 times the pre-set value (with first layer reeled on).

Remotely controlled mooring winches shall be provided with an alarm system giving warning in the remote control station when the permissible pulling force has been exceeded. The warning shall be given irrespective of the veered line length.

10.5 Towing Winches

10.5.1 Where automatic devices are used for governing the tension of the towline, provision shall be made for the continuous control of the tension. The tension indicators shall be fitted at the towing winch and in the wheelhouse.

10.5.2 Alarm system giving warning when the maximum permissible length of the towline is veered out shall be provided.

10.5.3 The drums of towing winches shall fulfil the requirements specified in paragraph 10.1.7 and shall be provided with fairleads. Separate fairleads shall be used when there are two or more drums. The rope drum shall be provided with clutches disengaging the drum from the driving gear.

Geometrical dimensions of the towing winch drums shall ensure the possibility of free veering of the towline.

10.5.4 The towing winch design shall provide for quick release of the rope drum brake for a free veer of the towline.

10.5.5 Towing winch brakes shall fulfil the following requirements:

.1 towing winch shall be provided with an automatic braking device to stop the winch when the pull is at least 1.25 times the rated pull in case of power decay or failure in the driving system;

.2 rope drum shall be fitted with the brake capable of stopping, without slip, the declutched drum loaded with the force not less than the towline breaking load. Power operated drum brakes shall also be provided with a manual control system. The brake design shall provide for quick release of the brake to ensure free heaving-in the towline.

10.5.6 Towline shall be so fixed to the winch drum that in case of the towline full release it will be disconnected from the drum due to the load equal to or slightly greater than the rated pull of the towing winch.

10.5.7 The components shall be calculated for stress occurring when the drum is subjected to loads corresponding to maximum torque of the motor, as well as when the drum is subjected to load equal to the towline breaking load. The equivalent stresses occurring in the components which may be subjected to acting forces caused by the above-mentioned loads shall not exceed 0.95 of this component material yield point.

10.5.8 Strength characteristics of the towline intended for working with the towing gear shall be marked on the towing gear.

11 HYDRAULIC DRIVES

11.1 Application

11.1.1 The requirements specified in this Chapter apply to all hydraulic appliances and systems aboard the ship except for those mentioned in paragraph 11.1.2.

11.1.2 Independent appliances – cased in individual housings – fulfilling the recognised standards which are not associated with the ship propulsion, steering and manoeuvring need not fulfil the requirements specified in this Chapter.

11.2 General Requirements

11.2.1 Hydraulic oil shall not be a source of corrosion in the hydraulic system. Flash point shall not be less than 150°C. Hydraulic oil shall be suitable for working within the range of operating temperatures of the hydraulic arrangement or system. In particular, this regards the range of viscosity change.

11.2.2 Hydraulic arrangements shall be protected with relief valves. Unless provided otherwise in other parts of the *Rules*, the opening pressure of the relief valve shall not exceed 1.1 of the maximum working pressure.

Nominal flow rate of the relief valves shall be so selected that the generated hydraulic oil pressure does not exceed 1.1 of the pre-set pressure of valve opening at the maximum pump output.

11.2.3 In hydraulic systems and appliances working continuously such as hydraulic main propulsion, steering gears, hydrodynamic couplings, the possibility of cleaning oil filters without stopping the system shall be provided.

11.2.4 A failure of the hydraulic system shall not cause damage to the associated piece of machinery or equipment.

11.2.5 Hydraulic systems of steering gears, as well as hydraulic systems actuating variable pitch propellers shall not have any connection with other hydraulic systems.

11.2.6 Where a feed pipe of hydraulic-powered windlasses is connected to other hydraulic systems, it shall be supplied by two independent pump systems, each of which shall ensure continuous operation of the windlass with the requirement specified in sub-chapter 10.3.1 being fulfilled.

11.3 Flammable Hydraulic Oil Tanks

Flammable oil tanks shall fulfil the same requirements as fuel tanks, with the following exceptions:

.1 in the case of tanks not adjacent to vessel shell plating which are situated outside the machinery compartments, in compartments situated above the load waterline where there are not sources of ignition such as internal combustion engines or boilers, the application of cylindrical level indicator glasses is permitted.

.2 in the case of tanks with the capacity less than 100 dm^3, situated in machinery compartments, PRS may consider acceptance of cylindrical level indicator glasses.

11.4 Pipe Connections

Pipe connections shall fulfil the requirements specified in sub-chapter 15.3, and additionally:

.1 pipes installed on board the ship shall have the inside surface as clean as it is required for hydraulic components,

.2 in pipelines with a nominal diameter less than 50 mm, threaded sleeve joints of a type approved by PRS shall be applied; however, the joints with the rubber washer may only be applied for connection of hydraulic components but not for connection of pipe segments,

.3 pipe joints without PRS approval may be applied, subject to PRS consent in each particular case, only where they fulfil the requirements specified in the relevant national standard and are provided with an appropriate inspection certificate,

.4 pipelines shall not have soldered joints,

.5 flexible hoses with connection fittings shall fulfil the requirements specified in paragraph 15.1.9 and shall be type approved by PRS. Subject to PRS consent in each particular case, fireproof hoses without PRS approval may be applied, except in the installations of steering gears and hydraulic control systems of watertight doors, ports and ramps in the shell plating, provided they fulfil the relevant national standard and have an appropriate inspection certificate.

11.5 Hydraulic Components

11.5.1 Hydraulic accumulators shall fulfil the strength requirements for pressure vessels of the particular class. Each accumulator, which may be cut off the hydraulic system shall be provided with an individual relief valve. A safety valve or other protecting device shall be installed on the gas side to prevent overpressure.

11.5.2 Hydraulic cylinders shall fulfil the strength requirements for pressure vessels of the particular class.

11.5.3 Hydraulic cylinders shall be type-approved by PRS. Subject to PRS consent in each particular case, hydraulic cylinders which are not type-approved by PRS may be applied if they fulfil the requirements specified in the relevant national standard and are provided with an appropriate inspection certificate.

11.5.4 Valves, pumps, hydraulic motors and high pressure filters shall be type-approved by PRS.

11.5.5 Hydraulic cylinders which do not fulfil the requirements specified in paragraph 11.5.3 as well as other hydraulic components which do not fulfil the requirement specified in paragraph 11.5.4 may be applied if they have been manufactured under PRS survey in accordance with the approved documentation and have been approved by PRS surveyor on the manufacturer's premises in accordance with the approved testing programme.

11.6 Testing

11.6.1 Tests shall be performed in accordance with the testing programme approved by PRS.

11.6.2 Testing programme shall determine the type and scope of tests, acceptance criteria, test site and – if necessary – testing procedure.

11.6.3 Tests shall include:
- **.1** pressure tests of piping in accordance with the requirements specified in sub-chapter 1.5.5;
- **.2** post-rinsing check of piping cleanness;
- **.3** operating tests;
- **.4** hydraulic oil check for impurities before and after operating tests.

12 HEATING, COOKING AND REFRIGERATING APPLIANCES

12.1 Oil Fuel Firing Appliances

12.1.1 General Requirements

12.1.1.1 In oil fuel firing heating, cooking and refrigerating appliances (for the purposes of this Chapter hereinafter referred to as appliances) only a fuel with the flash point not exceeding 55°C may be used. Such appliances shall be so designed as to be capable of being ignited without another flammable liquid.

12.1.1.2 The requirement to feed such appliances with an oil fuel having the flash point above 55°C does not apply to wick burners using lighting kerosene provided that the capacity of their tanks is not more than 12 litres and such appliances must not be used on the navigating bridge or in the accommodation spaces.

12.1.1.3 The appliances and their equipment shall be so designed as not to pose a fire hazard in case of overheating. The appliances shall be so fixed as to preclude the possibility of their accidental collapse or shifting.

12.1.1.4 The appliances must not be installed in the spaces where flammable liquids having the flash point below 55°C. Lines conveying exhaust gas from the appliances must not be led through such spaces.

12.1.1.5 Influx of outside air sufficient for burning of the oil fuel shall be provided (see also sub-chapters 19.1, 19.2 and 19.3).

12.1.1.6 Cooking and heating appliances shall be connected to the exhaust gas line and ventilators shall be installed over refrigerators. Exhaust lines shall be so arranged as to enable their periodical cleaning off the burning products' deposits. Exhaust lines' terminations on the open deck shall be provided with effective shielding or other means protecting against wind. Exhaust lines shall also fulfil the requirements specified in paragraphs 18.1.3, 18.1.4, 18.1.7 and 18.1.8.

12.1.1.7 Oil fuel tanks may form an integral part of the appliances or be installed separately.

The integral tanks shall be made of metal and have closed fillers whose design shall preclude their unintended opening and emptying. No soft soldered joint shall be below the maximum filling level of such a tank. The integral tanks shall be situated neither in way of the burner flame nor in such locations where the tanks might be exposed to overheating.

Tanks installed separately shall be positioned at the height recommended by the appliance manufacturer. Such tanks shall fulfil the requirements specified in sub-chapter 20.6. Tanks with a volumetric capacity more than 12 litres shall not be installed in accommodation spaces.

12.1.1.8 The appliances shall be so designed as to prevent the oil fuel leak in case the flame fades accidentally.

12.1.2 Galley Ranges, Heating Stoves and other Heating Appliances with Burners Using Fuel Vapours

12.1.2.1 Where such appliances are installed in machinery spaces, influx of outside air shall be supplied to these appliances and I.C. engines to ensure their stable and safe simultaneous operation. If necessary, separate ducts shall be arranged to supply air to these appliances and I.C. engines which shall be so installed as to prevent the burner flame reaching other equipment in the machinery space.

12.1.2.2 Under galley ranges and heating stoves dip-trays shall be installed to cover all the components containing oil fuel. The capacity of such trays shall be at least 2 litres, and the height of their side walls – at least 20 mm. For trays installed in the machinery space the height of their side walls shall be at least 200 mm and the upper edge of the side walls shall be at least 100 mm above the floor. The burner lower edge of the shall be situated above the upper edges of the dip-tray side walls.

12.1.2.3 Galley ranges and heating stoves shall be fitted with a suitable control ensuring, at any setting, virtually constant rate of fuel flow to the burner and prevents the fuel leak in case the flame fades. The control is considered suitable if it works accurately also in the conditions of jolting and heeling up to 12° to any direction and if it is provided, in addition to a float-type fuel level control, with:
- another float working safely and reliably to shut off the fuel flow where the permissible level of fuel has been exceeded, or
- spill pipe if the dip-tray has sufficient capacity for all the fuel oil content.

12.1.2.4 Fuel tanks not forming integral parts of galley ranges and heating stoves shall, in addition to the requirements of paragraph 12.1.1.7 being fulfilled, be provided with a valve operated from the deck to shut off the fuel.

12.1.2.5 Exhaust gas lines from galley ranges and heating stoves shall be fitted with devices preventing the gas reverse flow.

12.1.2.6 Heating appliances other than stoves shall fulfil the following requirements:
.1 fuel supply to the burner shall be possible as late as the burner has been aired;
.2 fuel supply to the burner shall be controlled by thermostat;
.3 fuel shall be ignited in the burner electrically or by pilot flame;
.4 in case the burner flame fades, the fuel supply shall be shut-off;
.5 main switch of the heating appliance shall be situated outside the space where the heating appliance is installed in a readily accessible position close to the entrance to that space.

12.1.3 Forced-draught Heating Appliances

12.1.3.1 Forced-draught heating appliances containing a combustion chamber around which compressed heated air flows on to the distribution system or to the heated space, shall fulfil the following requirements:
.1 if the fuel evaporates under pressure, the air for combustion shall be supplied by blower;

.2 burner ignition shall be possible as late as the combustion chamber has been aired. Airing is considered as sufficient if the air supplying blower continues to supply sir after the flame has been extinguished;

.3 fuel supply to the burner shall be automatically shut off in the following cases:
– flame fading;
– insufficient air supply for the combustion;
– heated air temperature is above the setting;
– power supply to the safety devices has been interrupted,
and the fuel supply must not be resumed automatically after such a shut-off.

.4 blowers supplying the combustion air as well as blowers forcing the heated air draught shall be fitted with switches enabling their stopping from the position outside the heated space;

.5 if the heated air is sucked from outside, inlets of the ventilation ducts shall be located as high above the deck as practicable. The location of inlets shall prevent penetration of rain or mist into the ducts;

.6 heated air distribution lines shall be made of metal;

.7 possibility of complete closing of the heated air outlets shall be precluded;

.8 possibility of penetration of the fuel leakage shall be precluded;

.9 possibility of air suction from the machinery space to the heating system shall be precluded.

12.2 Solid Fuel Firing Heating and Cooking Appliances

12.2.1 The appliances shall be placed on metal; plates having their edges so bent as to prevent the burning fuel or hot ash from falling out of the plate. This requirement does not apply to the heating appliances installed in the dedicated spaces constructed from fireproof materials.

12.2.2 Heating appliances shall be fitted with thermostats to control the combustion air intake.

12.2.3 In way of each solid fuel firing appliance, a container with sand shall be located or other means shall be provided to enable ready extinguishing of glowing ash.

12.2.4 Also the requirements specified in paragraphs 12.1.1.3, 12.1.1.4, 12.1.1.5 and 12.1.1.6 apply to solid fuel firing heating and cooking appliances.

12.3 Liquefied Gas Firing Heating, Cooking and Refrigerating Appliances

12.3.1 General Requirements

12.3.1.1 In liquefied gas firing heating, cooking and refrigerating appliances (hereinafter referred to as gas receivers), the gas having a trade name “propane” shall solely be used.

12.3.1.2 All components of the liquefied gas system (cylinders, pressure reducing valves, gas receivers, pipes, flexible hoses, fittings) shall comply with the relevant technical standards of the Flag State Administration.

12.3.1.3 Only gas receivers approved by a competent body in accordance with the effective provisions of the Flag State Administration may be installed on board the ship. These gas receivers shall be fitted with an automatic device cutting off the gas supply in case the main flame or the pilot flame fades.

12.3.1.4 Gas receivers and the associated installations may be used only in accommodation spaces and on the navigation bridge. No components of the liquefied gas firing installation shall be situated in machinery spaces.

12.3.1.5 Gas receivers may be installed in overnight cabins only where the cabin air is not used for the gas combustion.

12.3.1.6 Gas receivers may be installed in overnight cabins only where the construction of the navigation bridge precludes the possibility of penetration of the leaking gas into the spaces located below, particularly to the machinery spaces through any control system ducts.

12.3.1.7 Gas receivers using the ambient air for the gas combustion may be installed only in the large enough spaces provided with the ventilation in accordance with the requirements specified in sub-chapter 12.3.7.

12.3.1.8 Gas receivers shall be so fixed and connected to the liquefied gas system as to preclude the possibility of their accidental collapse or shifting and to preclude the possibility of twisting the supply pipes.

12.3.1.9 Several separate liquefied gas systems are permitted on board the ship. Accommodation spaces separated by a cargo hold or tanks shall not be served by the same liquefied gas system.

12.3.2 Gas Cylinders

12.3.2.1 Only gas cylinders of a capacity from 5 to 35 kg are permitted on board the ship.

12.3.2.2 Gas cylinders shall fulfil the requirements specified in sub-chapter 13.3.2.

12.3.3 Liquefied Gas Supply Units

12.3.3.1 Liquefied gas supply unit (i.e. a battery of gas cylinders) shall be placed in a free-standing cabinet situated on an open deck outside the accommodation spaces. The cabinet shall be so located as not to obstruct walking on the deck and it shall not be fixed to the bulwark in way of the bow or stern. It is permitted that a superstructure wall recess be used as a cabinet provided that its boundaries are gas-tight and access to the cylinders is possible from the open deck only. The cabinet shall be so located that the pipes led to the gas receivers be as short as practicable.

12.3.3.2 The cabinet shall be made of non-combustible materials and shall have ventilation openings in its upper and lower parts. On the cabinet outside wall, a notice "Liquefied Gas" together with a graphic symbol "No Smoking" not less than 100 mm in diameter shall be arranged. The cabinet shall be so designed and positioned that the inside temperature not exceed 50 °C.

12.3.3.3 Gas cylinders shall be effectively secured in the cabinet in an upright position.

12.3.3.4 Liquefied gas supply units shall be so installed that in case of leakage, the gas will go out of the cabinet to the open air and the possibility of the gas penetration into the ship enclosed spaces or where may be the source of ignition shall be precluded.

12.3.3.5 One liquefied gas system shall not be simultaneously supplied by more cylinders than it is necessary for its proper operation and the number of cylinders shall not exceed four. The cylinders may be open simultaneously only where they are interconnected by an automatic non-return fitting. The number of cylinders dedicated to one liquefied gas system shall not exceed six including the spare ones.

In passenger ships with galleys catering for passengers, the number of gas cylinders connected to one liquefied gas system may be increased to six, and the total of the cylinders dedicated to one liquefied gas system, i.e. inclusive of the spare ones, may be increased to nine items.

12.3.3.6 Pressure reducing valve, and in the case of two-stage pressure reduction – the first stage pressure reducing valve, shall be located in the cabinet containing the cylinders and permanently fixed to the cabinet wall.

12.3.4 Storage of Empty and Spare Gas Cylinders

Empty and spare gas cylinders shall be stored in cabinets in accordance with the requirements specified in paragraphs 12.3.3.1 and 12.3.3.2 located outside the accommodation spaces and outside the navigating bridge.

12.3.5 Pressure Reducing Valves

12.3.5.1 Gas receivers can be connected to the cylinders only by means of the distribution system fitted with one or more pressure reducing valves which decrease the gas pressure to the operation pressure in accordance with the characteristics specified in paragraph 12.3.5.4. Gas pressure may be reduced in one or two stages. The pressure at the outlet from the pressure reducing valve shall be permanently set and the possibility of changing the setting shall be precluded.

12.3.5.2 End pressure reducing valve shall be fitted with a safety device precluding overpressure in case of the pressure reducing valve failure. The device may be installed on or just after the pressure reducing valve. Means shall be provided to allow the gas release to the open air and the possibility of the gas penetration into the ship enclosed spaces, or where may be the source of ignition, shall be precluded.

12.3.5.3 Safety devices and associated vent pipes shall be protected against water penetration into them.

12.3.5.4 Pressure reducing valves shall maintain the gas pressure in the installation as follows:

.1 for two-stage pressure reduction, the intermediate pressure shall not exceed 250 kPa;

.2 after the end pressure reducing valve, the pressure shall not exceed 5 kPa, to a precision of +10%.

12.3.6 Pipelines, Flexible Hose Assemblies and Liquefied Gas Fittings

12.3.6.1 Liquefied gas lines shall be made of steel or copper. Steel lines shall be protected against corrosion. Wall thickness of the lines shall fulfil the requirements specified in sub-chapter 15.2.

12.3.6.2 Liquefied gas cylinders shall be connected to the distribution system by means of high-pressure flexible hose assemblies or corrugated pipes suitable for propane. Gas receivers which are not permanently fixed may be connected by means of flexible hose assemblies having not more than 1 m in length.

12.3.6.3 Gas lines shall have sufficient strength to withstand the stress occurring during the normal operation of the ship. The dimensions and routes of the lines shall be such as to ensure supplying the receivers with gas of the required pressure and flow rate.

12.3.6.4 Gas lines shall have as few joints as practicable. Gas lines and their joints shall remain gas-tight irrespective of the vibrations and strain they may be subjected to during the operation of the ship.

12.3.6.5 Ready access to the gas lines shall be provided. The lines shall be effectively fixed and they shall be effectively protected against damage particularly at penetrations through steel divisions.

12.3.6.6 Flexible hose assemblies and their joints shall have sufficient strength to withstand the stress occurring during the normal operation of the ship. By no means shall flexible hose assemblies be secured – they shall be laid or hung freely. Flexible hose assemblies shall be so positioned as to prevent their overtemperature and make them visible through their length.

12.3.6.7 The possibility shall be provided for the gas distribution system being cut off the supply unit by a readily accessible valve.

12.3.6.8 Each gas receiver shall be provided with a separate connection – fitted with a shut-off valve – to the gas distribution system. Shut-off valves shall situated in such positions where they be protected against damage and the effect of weather. If a gas receiver is connected to the distribution system by means of flexible hose assembly the shut-off valve shall be installed before the flexible hose assembly.

12.3.6.9 After each pressure reducing valve, a fitting shall be mounted to enable pressure tests of the installation. Shut-off valves shall be provided so as the pressure reducing valves are not subjected to the test pressure during the pressure tests of the installation.

12.3.6.10 Fittings of the liquefied gas system shall be made of bronze, brass or other non-sparking material resistant to corrosion.

12.3.7 Ventilation and Fume Extraction

12.3.7.1 In the spaces where are installed gas receivers using the ambient air for the gas combustion natural ventilation system shall be provided. The natural ventilation system shall consist of the supply and exhaust ducts, of the total net area at least 0.015 m^2 each, situated in the lower part of the space.

12.3.7.2 The ventilation ducts shall neither have any closing devices nor terminate in the in overnight cabins.

12.3.7.3 Exhaust gas arrangements shall be so designed as to ensure safe extraction of the exhaust gas. The arrangements shall be reliable and made of incombustible materials. The space ventilation system shall not interfere with the operation of the exhaust gas arrangements.

12.3.8 Operating and Safety Instructions

Liquefied gas system operating instructions shall be displayed in an appropriate position. The instructions shall contain the following information:

- **.1** valves of gas cylinders not connected to the system shall be permanently closed even if the cylinder is considered as empty;
- **.2** flexible hose assemblies shall be replaced immediately if their condition requires so;
- **.3** all gas receivers shall be connected to the system unless their individual branches are plugged.

12.3.9 Approval of Liquefied Gas System

Before a liquefied gas system is delivered for service after any alteration or repair and prior to the issue of the approval certificate mentioned in sub-chapter 12.3.11, the system shall be approved by a surveyor authorised by a competent body. The approval confirmation shall be available on board the ship – see 12.3.11.

12.3.10 Liquefied Gas System Tests

12.3.10.1 Gas lines running from the shut-off valve installed after the first pressure reducing valve (see paragraph 12.3.6.9) to the shut-off valve installed before the end pressure reducing valve shall be subjected to the following tests:

.1 strength test by pressure of 2 MPa exerted by compressed air, inert gas or liquid;

.2 gastightness test by pressure of 350 MPa exerted by compressed air or inert gas.

12.3.10.2 Gas lines running from the shut-off valve, installed after the single-stage pressure reducing valve or end pressure reducing valve, to the gas receivers' shut-off valves (see 12.3.6.8) shall be subjected to the gastightness test by pressure of 350 MPa exerted by compressed air or inert gas.

12.3.10.3 Gas lines running from the shut-off valve, installed after the single-stage pressure reducing valve or end pressure reducing valve, to the shut-off controls shall be subjected to the gastightness test by pressure of 15 kPa.

12.3.10.4 Gas lines subjected to the gastightness tests mentioned in 12.3.10.1.2, 12.3.10.2 and 12.3.10.3 are considered as gastight if no pressure drop has been found for 10 minutes since the time necessary for the pressure stabilising elapsed.

12.3.10.5 Elements exposed to the gas direct working pressure (from the cylinder fittings to connections of the pressure reducing valves with the distribution system) shall be subjected to the gastightness test by the cylinder gas using suds.

12.3.10.6 Each gas receiver shall be started and the gas combustion process shall be checked for different settings of the control cocks at the operation pressure. Also, operating tests of the automatic gas shut-off devices shall be performed after the flame fading.

12.3.10.7 After the tests mentioned in 12.3.10.6 have been performed, each gas receiver shall be connected to the exhaust line and started. During five-minute operation at the gas rated pressure and the ventilation in operation, it shall be checked that all the exhaust gas is going out through the exhaust gas line. During the test, the windows and doors in such a space shall be closed. If exhaust gas penetrates into the space, the cause of this shall be identified and rectified. Gas receiver shall not be approved for service until all the deficiencies have been rectified.

12.3.11 Approval Confirmation

12.3.11.1 Ship owner is obliged to keep a log book issued by a competent body for each liquefied gas system, and submit it for the entry of the approval confirmation.

12.3.11.2 The approval confirmation is entered by an authorised surveyor after the approval procedure and tests have performed in accordance with 12.3.9 and 12.3.10.

12.3.11.3 The approval confirmation is valid for 3 years. After the confirmation expires, the liquefied gas system is subject to the repeated approval procedure and tests.

12.3.11.4 Exceptionally, on the Owner's justified request, the validity of the confirmation may be extended by not more than 3 months. The consent for the extension of the confirmation validity is issued in writing and shall be enclosed to the liquefied gas system log book.

12.4 Electrical Heating and Cooking Appliances

12.4.1 Electrical heating and cooking appliances shall fulfil the requirements specified in sub-chapters 12.1, 12.2 and 12.3 of *Part VII – Electrical Installations and Control Systems*.

12.4.2 Electric radiators shall situated at least 50 mm from the divisions or ships sides. If the divisions or ship sides are lined with wood or other combustible materials, then the part of the lining in way of the radiators shall be shielded with a non-combustible material.

13 PRESSURE VESSELS AND HEAT EXCHANGERS

13.1 Construction of Pressure Vessels and Heat Exchangers

13.1.1 Components of pressure vessels and heat exchangers being in contact with sea water or other possibly corrosive media shall be constructed from corrosion-resistant materials. For other materials, the method of their protection against corrosion is subject to PRS consideration in each particular case.

13.1.2 Construction of pressure vessels and heat exchangers shall ensure their reliability in the conditions specified in sub-chapter 1.9.

13.1.3 Pipe wall thickness decreased due to bending shall not be less than the design thickness.

13.1.4 Construction of pressure vessels and heat exchangers shall take account of the possible thermal expansion of the shell and other components.

13.1.5 Shells of heat exchangers and pressure vessels shall be fixed to their seatings by supports. Upper fixing arrangements shall be provided if necessary (see also paragraph 2.3.1).

13.2 Fittings and Gauges

13.2.1 Pressure vessels and heat exchangers or their inseparable sets shall be fitted with non-disconnectable safety valves. In the case of several non-interconnected spaces, safety valves shall be provided for each space. Hydrophore tanks shall be fitted with safety valves located on the waterside.

In justified cases, PRS may waive the above-mentioned requirements.

13.2.2 In general, safety valves shall be of a spring-loaded type. Safety diaphragms of a type approved by PRS are permitted in fuel and oil heaters provided they are installed on the fuel and oil side.

13.2.3 The discharge capacity of safety valves shall be such that under no conditions the working pressure is exceeded by more than 10%.

13.2.4 Safety valves shall be so designed as to be capable of being sealed or fitted with an equivalent means to prevent their unauthorised adjustment. Materials used for springs and sealing surfaces of valves shall be resistant to corrosive effect of the medium.

13.2.5 Level indicators and sight glasses may only be installed on pressure vessels and heat exchangers where required by the conditions of control and inspection. Level indicators and sight glasses shall be of reliable construction and protected adequately. For oil and refrigerants, flat glass plates shall be used for level indicators and sight glasses.

13.2.6 Pressure vessels and heat exchangers shall be provided with flanges or flanged branch pieces for installation of fittings and mountings.

In hydrophore tanks, threaded branch pieces may also be applied.

13.2.7 Branch pieces installed on pressure vessels and heat exchangers shall be of rigid construction and of the minimum length sufficient for fixing and dismantling of mountings and fittings without their insulation removal. Branch pieces shall not be exposed to excessive bending stresses and shall be reinforced by stiffening fins where necessary.

13.2.8 Flanges intended for installation of mountings, fittings and piping, as well as branch pieces and sleeves passing through the entire thickness of pressure vessels and heat exchangers shall be fixed by welding, preferably with double welds. Branch pieces may be welded from one side, using a temporary backing strip or a different method ensuring full penetration of the wall.

13.2.9 Pressure vessels and heat exchangers shall be provided with adequate blowdown arrangements as well as drain arrangements.

13.2.10 Pressure vessels and heat exchangers shall be provided with manholes for internal examination. The minimum dimensions of the manholes are as follows:
– 300 × 400 mm – for oval manholes,
– 400 mm – for round manholes.

In justified cases PRS may consider the possibility of reduction of the dimensions to 280 × 380 mm – for oval manholes, and to 380 mm – for round manholes.

Oval manholes in shells shall be so situated that the minor axis be parallel with the shell axis.

13.2.11 Where the manholes mentioned in paragraph 13.2.10 are impracticable, adequate sight holes shall be provided. Pressure vessels and heat exchangers with more than 2.5 m in length shall be provided with the inspection holes at both ends.

Where the pressure vessel or heat exchanger is of dismountable construction or where corrosion and contamination of internal surfaces is precluded, manholes or inspection holes are not required.

Manholes or sight holes are not required where the construction of pressure vessel or heat exchanger precludes the possibility of inspection through such holes.

13.2.12 Where non-metallic gaskets are used, the construction of manholes and other holes shall preclude the possibility of gaskets being forced out.

13.2.13 Pressure vessels and heat exchangers, as well as their inseparable units shall be equipped with a pressure gauge or a compound pressure gauge. In heat exchangers divided into several spaces, a pressure gauge or a compound pressure gauge shall provided for each space (see also sub-chapter 1.13).

13.3 Requirements for Particular Types of Pressure Vessels and Heat Exchangers

13.3.1 Air Receivers

13.3.1.1 Safety valves of starting air receivers for main and auxiliary engines, as well as of fire protection systems, after being lifted, shall completely stop the air escape at the pressure inside the receiver not less than 0.85 of the working pressure.

13.3.1.2 Where air compressors, reducing valves or pipes from which air is supplied to the receivers are provided with safety valves so adjusted to prevent the receivers from being supplied with air of the pressure higher than the working pressure, safety valves need not be fitted on such receivers. In that case, fusible plugs shall be fitted on the receivers instead of the safety valves.

13.3.1.3 The fusible plugs shall have a fusion temperature within 100-130 °C. The fusion temperature shall be permanently marked on the fusible plug. Air receivers having a capacity over 0.7 m^3 shall be fitted with plugs not less than 10 mm in diameter.

13.3.1.4 Air receivers shall be equipped with water-draining arrangements. In air receivers positioned horizontally, the water draining arrangements shall be installed at both ends of the receiver.

13.3.2 Cylinders for Compressed Gases

13.3.2.1 Cylinders for compressed gases are portable pressure vessels designed for the storage of compressed gases or an extinguishing medium, which are stored on board the ship for her operational purposes, but are incapable of being filled by means of the ship's equipment.

13.3.2.2 Strength calculations shall be performed taking account of the requirements specified in sub-chapter 14.2.8 and the following:
- design pressure shall not be less than the pressure which may occur at temperature 45°C, at the predetermined filling level;
- allowable stress σ shall be determined in accordance with sub-chapter 14.2.4, whereas the safety factor in accordance with paragraph 14.2.5.1;
- allowance *c* for cylinders being exposed to corrosion shall not be taken less than 0.5 mm.

Cylinders may be made of steel with the yield stress greater than 750 MPa, but not exceeding 850 MPa, subject to PRS consent in each particular case.

13.3.2.3 Non-disconnectable safety devices of approved construction shall be provided to prevent a dangerous overpressure in the cylinder in case of temperature increase. Safety valves or burst disks activated at a pressure exceeding 1.1 times the working pressure, but not higher than 0.9 times the test pressure, are permitted.

13.3.2.4 Cylinders shall be permanently marked to include the following information:
.1 manufacturer name,
.2 serial number,
.3 year of manufacture,
.4 kind of gas,
.5 capacity,
.6 test pressure,
.7 tare,
.8 maximum load (pressure/weight),
.9 stamp and date of testing.

13.3.2.5 Cylinders shall be subjected to a hydraulic test under pressure equal to 1.5 times the working pressure.

13.3.2.6 Cylinders which are designed for the storage of compressed gases, refrigerants or extinguishing agents shall be approved by PRS or shall be manufactured in accordance with the relevant standards under the survey of a competent technical inspection body approved by PRS.

13.3.3 Pressure Vessels for Processing Fishery Products

13.3.3.1 Pressure vessel covers opened periodically shall be fitted with devices preventing a partial closing or spontaneous opening of the covers. Provision shall be made to preclude the possibility of opening the cover in the case of excessive pressure or underpressure, as well as to preclude the possibility of pressurizing the receiver when the cover is partially closed.

13.3.3.2 Internal equipment, such as mixers, coils, disks, partitions, etc., hindering the internal inspection of the vessels shall be readily removable.

13.3.3.3 Sight glasses of not more than 150 mm in diameter, may be used to monitor the working spaces of mixers, provided that the working pressure in such spaces does not exceed 0.25 MPa.

13.3.3.4 In pressure vessels operating at a pressure exceeding 0.25 MPa, the covers of loading openings shall be so designed that, in the case of seal rupture, the hot medium escapes in a safe direction without hazard for the personnel.

13.3.3.5 Pressure vessels operating under vacuum conditions, heated by steam or water of a temperature over 115 °C, shall be fitted with safety valves to prevent the pressure in the vacuum space from rising (due to the heating system leakage) higher than 0.85 times the test pressure.

These vessels shall be designed for such an opening pressure of the safety valve that the design stresses will not exceed 0.8 times the yield stress of the material at the design temperature.

13.3.3.6 For mixers heated by steam or water, as well as for the walls of vessels being in contact with the rotating product, the design wall thickness allowance, c, shall not be taken less than 2 mm.

13.3.4 Filters and Coolers

13.3.4.1 Filters and coolers of the main and auxiliary engines shall fulfil the requirements for heat exchangers and pressure vessels with respect to the materials and construction.

13.3.4.2 Oil fuel filters installed in parallel to enable their cleaning without cutting off the fuel oil supply to engines (duplex filters) shall be provided with arrangements protecting the filter under pressure against being opened inadvertently.

13.3.4.3 Oil filters or filter chambers shall be provided with adequate means for:
– air venting when being put into operation,
– pressure equalisation before being opened.

Valves or cocks with drain pipes leading to a safe location shall be used for this purpose.

14 STRENGTH CALCULATIONS FOR PRESSURE VESSELS AND HEAT EXCHANGERS

14.1 General Requirements

Depending on the design and parameters, pressure vessels and heat exchangers are divided into classes as indicated in Table 14.1.

Table 14.1

Kind of equipment	Class I	Class II	Class III
Pressure vessels and heat exchangers	$p > 4.0$ or $t > 350$ or $s > 35$	$1.6 < p \le 4.0$ or $120 < t \le 350$ or $16 < s \le 35$	$p \le 1.6$ and $t \le 120$ and $s \le 16$
Pressure vessels and heat exchangers containing toxic, inflammable or explosive media	irrespective of parameters	–	–

p – design pressure, i.e. the pressure taken for strength calculations, not less than opening pressure of safety valves or other safety devices, [MPa];
t – design wall temperature, [°C];
s – wall thickness, [mm].

14.2 Strength Calculations

14.2.1 General Requirements

14.2.1.1 Wall thicknesses determined by calculation are the lowest permissible values under normal operating conditions. The formulae and strength calculation methods do not take into account the manufacturer's tolerances for thickness and these shall be added as special allowances to the design thickness values.

Additional stresses due to external loads (axial forces, bending moments, torques) imposed on the calculated parts (particularly loads due to dead mass or the mass of attached parts) shall be taken into account on PRS request.

14.2.1.2 The dimensions of structural components of pressure vessels and heat exchangers for which no strength calculation methods are given in this *Part* of the *Rules* shall be determined on the basis of experimental data and recognized theoretical calculations, and are subject to special consideration by PRS in each particular case.

14.2.2 Design Pressure

14.2.2.1 Where hydrostatic pressure is greater than 0.05 MPa, the design pressure shall be increased by that value.

14.2.2.2 For flat walls subjected to pressure from both sides, the design pressure shall be taken as the greatest of the acting pressures. Walls in the form of curved surfaces which are subjected to pressure from both sides shall be calculated for the greatest outer and inner pressures. If the pressure on one side of the flat wall or the wall in the form of curved surface is lower than the atmospheric pressure, than the maximum pressure on the other side of the wall increased by 0.1 MPa shall be taken as the design pressure.

14.2.3 Design Temperature

14.2.3.1 For the purpose of determining the allowable stresses depending on the temperature of the medium, the design wall temperature shall not be taken lower than indicated in Table 14.2.3.1.

Table 14.2.3.1

Item	Components of boilers, pressure vessels and heat exchangers and operating conditions thereof	Design wall temperature
1	Heated components	T_v
2	Not heated components [1]	T_m

Notes to Table 14.2.3.1:

[1] – see 14.2.3.2;
T_m – maximum temperature of heated medium, [°C];
T_v – maximum temperature of heating medium. [°C].

14.2.3.2 A wall is considered to be non-heated if it is covered with fire-resisting insulation not exposed to radiant heat.

14.2.3.3 Design temperature for tank walls and pressure vessel walls operating under refrigerant pressure shall be taken equal to 20°C, if higher temperatures are not likely to occur.

14.2.4 Strength Characteristics of Materials and Allowable Stresses

14.2.4.1 For steels with $(R_e/R_m) \le 0.6$ the strength characteristics shall be taken equal to physical yield point or proof stress R_e^t or $R_{0.2}^t$, as well as average creep strength $R_{z/100\,000/t}$ after 10^5 h, at design temperature t.

For steels with $(R_e/R_m) > 0.6$, R_m^t, tensile strength at design temperature t shall also be taken into account.

The minimum values of R_e^t, $R_{0.2}^t$ and R_m^t and average values of $R_{1/100\,000/t}$ and $R_{z/100\,000/t}$ shall be taken for calculations.

14.2.4.2 For materials whose stress-strain curve does not show a specific yield stress, the tensile strength at the design temperature shall be taken for calculations.

14.2.4.3 For cast iron and non-ferrous alloys, the minimum value of ultimate tensile strength at normal temperature shall be taken for calculations.

14.2.4.4 When using non-ferrous materials and their alloys, it shall be taken into account that the heating during processing and welding reduces the strengthening effect achieved by cold processing. Therefore the strength characteristics to be used for strength calculations of the components and assemblies made of such materials shall be those applicable to their annealed condition.

14.2.4.5 Allowable stresses σ assumed for strength calculations shall be determined as the minimum out of the following three values:

$$\sigma = \frac{R_m^t}{\eta_m},\ \sigma = \frac{R_e^t}{\eta_e} \quad \text{or} \quad \sigma = \frac{R_{0,2}^t}{\eta_e}$$
$$\sigma = \frac{R_{z/100000/t}}{\eta_z},\ \sigma = \frac{R_{1/100000/t}}{\eta_p}$$

where:

η_m – safety factor for tensile strength R_m^t

η_z – safety factor for tensile strength $R_{z/100\,000/t}$

η_e – safety factor for yield point R_e^t and $R_{0,2}^t$

η_p – safety factor for creep point $R_{1/100\,000/t}$

For values of factors – see sub-chapter 14.2.5.

14.2.5 Safety Factors

14.2.5.1 For components made of steel forgings or rolled steel, subjected to internal pressure, the safety factors shall not be less than:

$$\eta_e = \eta_z = 1.6;\ \ \eta_m = 2.7 \ \text{ and } \ \eta_p = 1.0$$

For components subjected to external pressure, safety factors η_e, η_z and η_m shall be increased by 20%.

14.2.5.2 For components of boilers, heat exchangers and pressure vessels of Class II and Class III, made of steels with $(R_e/R_m) \le 0.6$, the safety factors may be reduced, however they shall not be less than:

$$\eta_e = \eta_z = 1.5; \;\; \eta_m = 2.6$$

14.2.5.3 For components of heat exchangers and pressure vessels made of cast steel and subjected to internal pressure, the safety factors shall not be less than:

$$\eta_e = \eta_z = 2.2; \;\; \eta_m = 3.0 \;\text{ and }\; \eta_p = 1.0$$

For components exposed to outer pressure, safety factors η_e and η_m shall be increased by 20% (η_z remains unchanged).

14.2.5.4 Safety factor η_m for components made of cast iron shall not be taken less than 4.8 for external and internal pressures.

Safety factor η_m for components made of non-ferrous alloys shall not be taken less than 4.6 for internal pressure and 5.5 for external pressure. For conical shells, η_m shall not be taken less than 6.0 for external pressure.

14.2.6 Strength Factors

14.2.6.1 Strength factors φ of welded joints shall be determined in accordance with Table 14.2.6.1-1 depending on the joint type and welding process. For particular classes of boilers, pressure vessels and heat exchangers (see Table 14.1), strength factor φ shall not be less than that specified in Table 14.2.6.1-2.

Table 14.2.6.1-1

Welding process	Joint type	Weld type	φ
Automatic	Butt joint	Double-sided Single-sided with backing Single-sided without backing	1.0 0.9 0.8
	Overlap joint	Double-sided Single-sided	0.8 0.7
Semi-automatic and manual	Butt joint	Double-sided Single-sided with backing Single-sided without backing	0.9 0.8 0.7
	Overlap joint	Double-sided Single-sided	0.7 0.6

Notes to Table 14.2.6.1-1:

1. Full penetration shall be achieved in each case.
2. For welded joints made in electroslag process, $\varphi = 1$ shall be taken.

Table 14.2.6.1-2

Equipment type	Factor φ		
	Class I	Class II	Class III
Pressure vessels and heat exchangers	0.9	0.7	0.6

14.2.6.2 Strength factor of cylindrical walls weakened by holes with uniform diameter shall be taken equal to the lowest of the following three values:

.1 strength factor of cylindrical walls weakened by a longitudinal row or a field of equally spaced holes (Fig. 14.2.6.2-1), as determined using the formula below:

$$\varphi = \frac{a-d}{a} \qquad (14.2.6.2.1)$$

.2 strength factor, reduced to the longitudinal direction, of cylindrical walls weakened by a transverse row or a field of equally spaced holes (Fig. 14.2.6.2-1), as determined using the formula below:

$$\varphi = 2\frac{a_1-d}{a_1} \qquad (14.2.6.2.2)$$

.3 strength factor, reduced to the longitudinal direction, of cylindrical walls weakened by a field of equally spaced staggered holes (Fig. 14.2.6.2-2 and Fig. 14.2.6.2-3), as determined using the formula below:

$$\varphi = k\frac{a_2-d}{a_2} \qquad (14.2.6.2.3\text{-}1)$$

where:

φ – strength factor of walls weakened by holes;
d – diameter of the holes for expanded tubes or inner diameter of welded-on tubes and extruded branch pieces, [mm];
a – spacing between axes of two adjacent holes arranged along the wall, [mm];
a_1 – spacing between axes of two adjacent holes in the transverse (or circumferential) direction, taken as the mean circumference arc length, [mm];
a_2 – spacing between axes of two adjacent holes in staggered rows, taken as mean circumference arc length, [mm], as determined using the formula below:

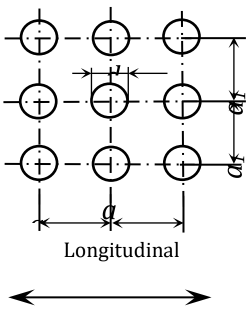
$$a_2 = \sqrt{l^2 + l_1^2}\ \text{[mm]} \qquad (14.2.6.2.3\text{-}2)$$

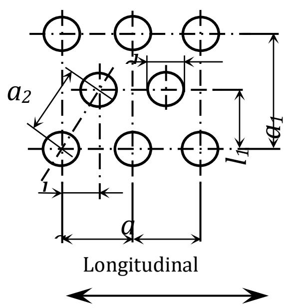
l – spacing between axes of two adjacent holes in the longitudinal direction (Fig. 14.2.6.2.-2 and Fig. 14.2.6.2-3), [mm];
l_1 – spacing between axes of two adjacent holes in the transverse or circumferential direction (Fig. 14.2.6.2.-2 and Fig. 14.2.6.2-3), [mm];
k – factor depending on ratio $\frac{l_1}{l}$, taken from Table 14.2.6.2.3.

Table 14.2.6.2.3

$\frac{l_1}{l}$	5.0	4.5	4.0	3.5	3.0	2.5	2.0	1.5	1.0	0.5
k	1.76	1.73	1.70	1.65	1.60	1.51	1.41	1.27	1.13	1.00

Note:
Intermediate values of k shall be determined by linear interpolation.





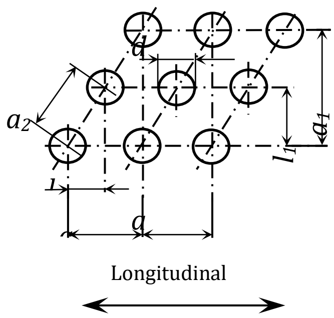


Fig. 14.2.6.2-1Fig. 14.2.6.2-2Fig. 14.2.6.2-3

14.2.6.3 Where rows or fields of equally spaced holes contain holes of different diameters, value d in the formulae for strength factor determination (14.2.6.2.1, 14.2.6.2.2, 14.2.6.2.3-1, 14.2.6.2.3-2) shall be taken as the value equal to the arithmetic mean of the two largest adjacent holes. In the case of uneven spacing between the holes of equal diameters, the lowest values of a, a_1 or a_2 .

14.2.6.4 In the case of weld seams with holes, the strength factor shall be taken as the product of the seam strength factor and the strength factor of the wall weakened by the holes.

14.2.6.5 For seamless cylindrical walls not weakened by a seam or row/field of holes, strength factor φ shall be taken as equal to 1.0. In no case factor φ shall be taken greater than 1.0.

14.2.6.6 Strength factor of walls weakened by holes for expanded tubes, as determined in accordance with formulae 14.2.6.2.1, 14.2.6.2.2, 14.2.6.2.3, shall not be taken less than 0.3. Calculations with the lesser value of the strength factor are subject to PRS consideration in each particular case.

14.2.6.7 For walls of cylindrical components made of sheets with different thickness, joined by longitudinal weld seam, the thickness calculations shall be done separately for each sheet, taking account of the actual weakenings.

14.2.6.8 For tubes with longitudinal weld seam, the strength factor is subject to PRS consideration in each particular case.

14.2.6.9 Strength factors for walls weakened by openings requiring full or partial strengthening shall be determined in accordance with sub-chapter 14.2.17.7.

14.2.6.10 Strength factors for flat flue sheets shall be determined in accordance with formula 14.2.6.2.1 for tangential and radial spacings respectively. The lesser strength factor obtained shall be taken for calculation of the flat flue sheet thickness.

14.2.7 Design Thickness Allowances

14.2.7.1 In each case where the design wall thickness allowance c, is not expressly specified, it shall be taken at least 1 mm. For steel walls with more than 30 mm in thickness, as well as for walls of corrosion-resistant non-ferrous metals or high alloy materials, and for materials adequately protected against corrosion, e.g. by cladding or coating with a protective compound, the design thickness allowance may be waived subject to PRS acceptance in each particular case.

14.2.7.2 For pressure vessels and heat exchangers inaccessible for internal examination and for those whose walls are subjected to heavy corrosion or wear, PRS may require an increased allowance c to the design thickness.

14.2.8 Cylindrical and Spherical Elements and Tubes Subjected to Internal Pressure

14.2.8.1 The requirements specified in this sub-chapter may be applied where the following conditions are fulfilled:

$\frac{D_a}{D} \leq 1.6$ – for cylindrical elements,

$\frac{D_a}{D} \leq 1.7$ – for tubes,

$\frac{D_a}{D} \leq 1{,}2$ – for spherical elements.

Cylindrical elements with a diameter $D_a \leq 200$ mm shall be considered as tubes.

For D_a , D – see paragraph 14.2.8.2.

14.2.8.2 Thickness of cylindrical walls and tubes shall not be less than the values calculated in accordance with the formulae below:

$$s = \frac{D_a p}{2\sigma\varphi + p} + c \quad [\text{mm}] \qquad (14.2.8.2\text{-}1)$$

or

$$s = \frac{D p}{2\sigma\varphi - p} + c \quad [\text{mm}] \qquad (14.2.8.2\text{-}2)$$

s – wall thickness, [mm];
p – design pressure, [MPa];
D_a – outside diameter, [mm];
D – inside diameter, [mm];
φ – strength factor (see sub-chapter 14.2.6);
σ – allowable stress (see paragraph 14.2.4.5), [MPa];
c – design thickness allowance (see sub-chapter 14.2.7), [mm].

14.2.8.3 Thickness of spherical walls shall not be less than the values obtained from the formulae:

$$s = \frac{D_a p}{4\sigma\varphi + p} + c \quad [\text{mm}] \qquad (14.2.8.3\text{-}1)$$

or

$$s = \frac{D p}{4\sigma\varphi - p} + c\varphi \quad [\text{mm}] \qquad (14.2.8.3\text{-}2)$$

For symbols – see paragraph 14.2.8.2.

14.2.8.4 Irrespective of the values obtained in accordance with formulae 14.2.8.2-1, 14.2.8.2-2, 14.2.8.3-1 and 14.2.8.3-2, the thickness of spherical and cylindrical walls as well as tubes shall not be less than:

.1 5 mm – for seamless and welded elements;
.2 12 mm – for tube plates with radial hole arrangement for expanded tubes;
.3 6 mm – for tube plates with welded-on or soldered-on tubes;
.4 specified in Table 14.2.8.4 – for tubes.

Table 14.2.8.4

Tube outside diameter, [mm]	≤20	>20 ≤30	>30 ≤38	>38 ≤51	>51 ≤70	>70 ≤95	>95 ≤102	>102 ≤121	>121 ≤152	>152 ≤191	>191
Minimum wall thickness, [mm]	1.75	2.0	2.2	2.4	2.6	3.0	3.25	3.5	4.0	5.0	5.4

Note:
The decrease in wall thickness due to expanding or bending shall be compensated by allowances.

14.2.8.5 The minimum wall thickness of pipes made of non-ferrous alloys and stainless steel may be less than those specified in paragraph 14.2.8.4 subject to PRS consent in each particular case, however not less than those determined in accordance with the formulae specified in paragraphs 14.2.8.2 and 14.2.8.3.

14.2.9 Elements Subjected to External Pressure

14.2.9.1 The requirements specified in this sub-chapter apply to cylindrical walls with:

$$\frac{D_a}{D} \leq 1.2$$

Wall thickness of pipes with Da ≤ 200 mm in diameter shall be determined in accordance with paragraph 14.2.8.2.

14.2.9.2 Plain wall thickness of cylindrical elements, with or without stiffeners, shall not be less than that determined in accordance with the formula below:

$$s = \frac{50(B+\sqrt{B^2+0.04AC})}{A} + c \quad [\text{mm}] \qquad (14.2.9.2\text{-}1)$$

where:

$$A = 200\frac{\sigma}{D_m}(1+\frac{D_m}{10l})(1+\frac{5D_m}{l}) \qquad (14.2.9.2\text{-}2)$$

$$B = p(1+\frac{5D_m}{l}) \qquad (14.2.9.2\text{-}3)$$

$$C = 0.045 \cdot p \cdot D_m \qquad (14.2.9.2\text{-}4)$$

s – wall thickness, [mm];
p – design pressure (see sub-chapter 14.2.2), [MPa];
D_m – mean diameter, [mm];
σ – allowable stress (see paragraph 14.2.4.5), [MPa];
c – design thickness allowance (see sub-chapter 14.2.7), [mm];
l – design length of cylindrical portion between stiffeners, [mm].

End plates and stiffening rings (Fig. 14.2.9.2) or similar structures may be considered as stiffeners.

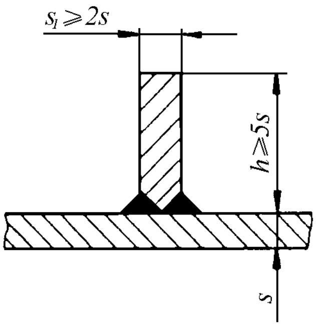


Fig. 14.2.9.2

14.2.9.3 Strengthening means shall be provided in way of holes and openings in cylindrical and spherical walls in accordance with the requirements specified in sub-chapter 14.2.17.

14.2.10 Conical Elements

14.2.10.1 Wall thickness of conical elements subjected to internal pressure shall not be less than:

.1 for $\alpha \le 70°$ – the greater value out of those determined in accordance with the formulae below:

$$s = \frac{D_a p y}{4\sigma\varphi} + c \quad [\text{mm}] \qquad (14.2.10.1.1\text{-}1)$$

and

$$s = \frac{D_c p y}{(4\sigma-p)cos\alpha} + c \quad [\text{mm}] \qquad (14.2.10.1.1\text{-}2)$$

.2 for $\alpha > 70°$ – the value determined in accordance with the formula below:

$$s = 0.3[D_a - (r+s)]\sqrt{\frac{p}{\sigma}\cdot\frac{\alpha}{90°}} + c \quad [\text{mm}] \qquad (14.2.10.1.2)$$

s – wall thickness, [mm];
D_c – design diameter (see Figures. 14.2.10.1.2-1 to 14.2.10.1.2-4), [mm];
D_a – outside diameter (see Figures 14.2.10.1.2-1 to 14.2.10.1.2-4), [mm];
p – design pressure (see sub-chapter 14.2.2), [MPa];
y – shape factor (see Table 14.2.10.1);
α, α_1, α_2, α_3 – angles (see Figures 14.2.10.1.2-1 to 14.2.10.1.2-4), [°];
σ – allowable stress (see paragraph 14.2.4.5), [MPa];
φ – strength factor (see sub-chapter 14.2.6). In formulae 14.2.10.1.1-1 and 14.2.10.1.2, the strength factor for circumferential weld seam shall be applied, whereas in formula 14.2.10.1.1-2 – for longitudinal weld seam. For seamless conical shell segments, and also where circumferential seam is at the distance from the edge exceeding:

$0.5\sqrt{\frac{D_a s}{\cos\alpha}}$ strength factor $\varphi = 1$ shall be taken;

r – edge radius (Figures 14.2.10.1.2-1, 14.2.10.1.2-2 and 14.2.10.1.2-4), [mm];
c – design thickness allowance (see sub-chapter 14.2.7), [mm].

Table 14.2.10.1

α [deg]	Shape factor y as function of r / D_a ratio											
	0.01	0.02	0.03	0.04	0.06	0.08	0.10	0.15	0.20	0.30	0.40	0.50
10	1.4	1.3	1.2	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1
20	2.0	1.8	1.7	1.6	1.4	1.3	1.2	1.1	1.1	1.1	1.1	1.1
30	2.7	2.4	2.2	2.0	1.8	1.7	1.6	1.4	1.3	1.1	1.1	1.1
45	4.1	3.7	3.3	3.0	2.6	2.4	2.2	1.9	1.8	1.4	1.1	1.1
60	6.4	5.7	5.1	4.7	4.0	3.5	3.2	2.8	2.5	2.0	1.4	1.1
75	13.6	11.7	10.7	9.5	7.7	7.0	6.3	5.4	4.8	3.1	2.0	1.1

Note:
For welded joints (see Fig. 14.2.10.1.2-3), shape factor y shall be determined for $r / D_a = 0.01$.

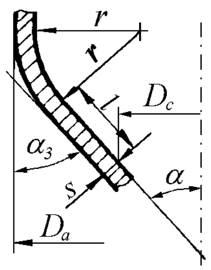


Fig. 14.2.10.1.2-1

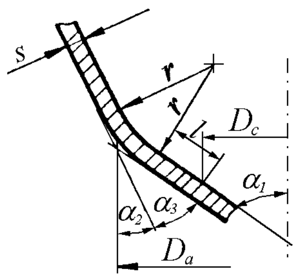


Fig. 14.2.10.1.2-2

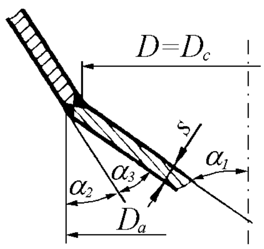


Fig. 14.2.10.1.2-3

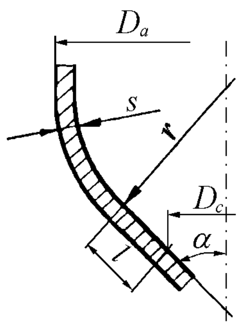


Fig. 14.2.10.1.2-4

l – distance from the edge of the wide end of conical shell, along the generatrix, taken as tenfold wall thickness, however not greater than half the length of the conical shell generatrix segment (Figures 14.2.10.1.2-1, 14.2.10.1.2-2 and 14.2.10.1.2-4), [mm];

14.2.10.2 Wall thickness of conical elements subjected to external pressure shall be determined in accordance with paragraph 14.2.10.1, provided the following conditions are fulfilled:

.1 strength factor of welded joint φ shall be taken equal to 1;

.2 allowance c shall be taken equal to 2 mm;

.3 design diameter D_c shall be determined in accordance with the formula below:

$$D_c = \frac{d_1+d_2}{2\cos\alpha} \quad [\text{mm}] \tag{14.2.10.2.3}$$

d_1, d_2 – the greatest and the smallest diameter of the cone, respectively, [mm];

.4 for $\alpha < 45°$ it shall be demonstrated that the walls are not subjected to plastic strain. Pressure p_1, at which plastic strain occurs, shall be determined in accordance with the formula below:

$$p_1 = 26E10^{-6}\frac{D_c}{l_1}[\frac{100(s-c)}{D_c}]^2\sqrt{\frac{100(s-c)}{D_c}} \quad [\text{MPa}] \tag{14.2.10.2.4}$$

E – modulus of elasticity, [MPa];
l_1 – the maximum length of the cone or distance between its restrains, [mm].

Fulfilment of inequality $p_1 > p$ (p – design pressure, [MPa]) is the condition of absence of plastic strain of the cone walls.

14.2.10.3 Welded joints (see Fig. 14.2.10.1.2-3) are permitted only with the values of angle α3 ≤ 30° and wall thickness s ≤ 20 mm. The joints shall be double-side welded. In conical shell segments with α ≥ 70°, welded joints may be made without edge bevelling provided that the requirements specified in paragraph 14.2.10.2 are fulfilled.

14.2.10.4 In way of holes and openings in conical walls, adequate strengthening shall be provided in accordance with the requirements specified in sub-chapter 14.2.17.

14.2.11 Flat End Plates and Covers

14.2.11.1 Thickness of the flat end plates unsupported by stays, as well as of welded or bolted covers (Figures 14.2.11.1-1 ÷ 14.2.11.1-8 and Fig. 1.2 in the Annex) shall not be less than that determined in accordance with the formula below:

$$s = KD_c\sqrt{\frac{p}{\sigma}}+c \quad [\text{mm}] \tag{14.2.11.1-1}$$

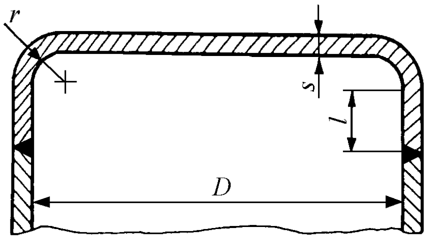
s – wall thickness, [mm];
K – design factor for the design patterns shown in Figures 14.2.11.1-1 to 14.2.11.1-8 and items 1.1 to 1.6 in the Annex;
D_c – design diameter (Figures 14.2.11.1-2 to 14.2.11.1-7 and item 1.2 in the Annex), [mm]; for end plates such as shown in Fig. 14.2.11.1-1 and item 1.1 in the Annex, the design diameter shall be determined in accordance with the formula below:

$$D_c = D - r \quad [\text{mm}] \tag{14.2.11.1-2}$$

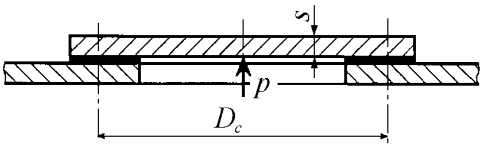
for rectangular or oval covers the design diameter shall be determined in accordance with the formula below:

$$D_c = m\sqrt{\frac{2}{1+(\frac{m}{n})^2}} \quad [\text{mm}] \tag{14.2.11.1-3}$$

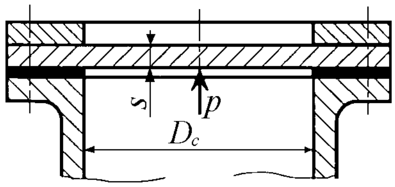
D_b – pitch circle diameter of bolts (Fig. 14.2.11.1-6), [mm];
D – internal diameter, [mm];
n and m – the maximum and minimum length of the axis or the side of the opening respectively, measured to the axis of the packing arrangement, [mm] (Fig. 14.2.11.1-8);
r – inner curvature radius of the dished end plate, [mm];
p – design pressure (see sub-chapter 14.2.2), [MPa];
σ – allowable stress (see paragraph 14.2.4.5), [MPa];
c – design thickness allowance (see sub-chapter 14.2.7), [mm];
l – length of end plate cylindrical portion (see Fig. 14.2.11.1-1 and item 1.1 in the Annex), [mm].



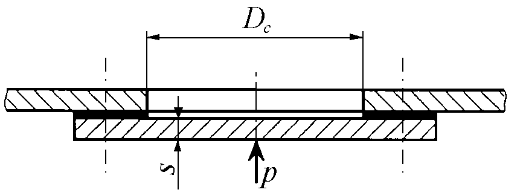
$K = 0.30$
Fig. 14.2.11.1-1



$K = 0.41$
Fig. 14.2.11.1-2

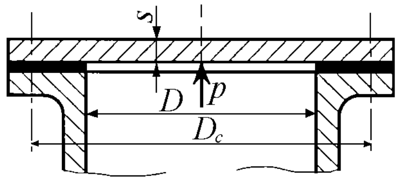


$K = 0.41$
Fig. 14.2.11.1-4



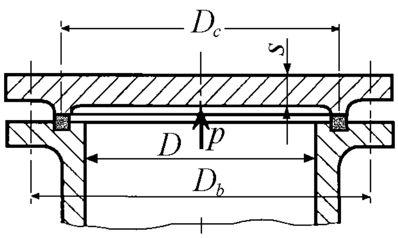
$K = 0.45$

Fig. 14.2.11.1-3



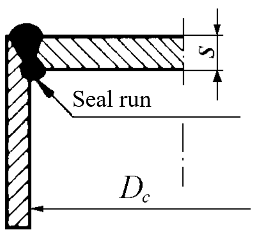
$K = 0.35$

Fig. 14.2.11.1-5



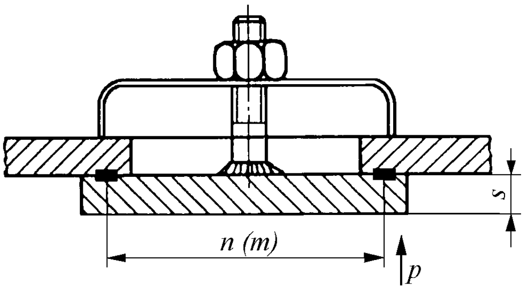
D_b/D	K
1.25	0.6
1.50	0.7
1.75	0.8

Fig. 14.2.11.1-6



$K = 0,50$

Fig. 14.2.11.1-7



$K = 0.53$

Fig. 14.2.11.1-8

14.2.11.2 Thickness of the plates shown in item 1.2 of the Annex shall not be less than that determined in accordance with formula 14.2.11.1-1. Additionally, the following requirements shall be fulfilled:

.1 For circular end plates

$$0.77s_1 \geq s_2 \geq \frac{1.3p}{\sigma}\left(\frac{D_c}{2} - r\right) \qquad (14.2.11.2.1)$$

.2 For rectangular end plates

$$0.55s_1 \geq s_2 \geq \frac{1.3p}{\sigma} \cdot \frac{nm}{(n+m)} \qquad (14.2.11.2.2)$$

s – end plate thickness, [mm];
s_1 – shell thickness, [mm];
s_2 – end plate thickness within the relieving groove, [mm].

For other symbols – see sub-chapter 14.2.11.1.

Thickness s_2 shall never be less than 5 mm.

The above conditions apply to end plates with not more than 200 mm in diameter or side length. The dimensions of relieving grooves in end plates with diameters or side lengths over 200 mm are subject to PRS consideration in each particular case.

14.2.12 Flanging Flat Walls

14.2.12.1 In flat wall and end plate calculations, the flanging can be taken into account only when the inner flanging radius is not less than that specified in Table 14.2.12.1.

Table 14.2.12.1

End plate outer diameter [mm]	Flanging radius [mm]
up to 350	25
from 350 to 500	30
from 500 to 950	35

The inner flanging radius shall not be less than 1.3 times the wall thickness.

14.2.12.2 The length of cylindrical portion of a flanged flat end plate, l, shall not be less than that determined in accordance with the formula below:

$$l = 0.5\sqrt{Ds} \quad \text{[mm]} \qquad (14.2.12.2)$$

for symbols l, D, s – see Fig. 14.2.11.1-1.

14.2.13 Strengthening of Openings in Flat Walls

14.2.13.1 In flat walls, end plates and covers, openings with diameters greater than four times the thickness shall be strengthened by means of welded-on branch pieces or pads, or by increasing the design wall thickness. The openings shall be arranged at a distance not less than 0.125 times the design diameter from the design diameter outline.

14.2.13.2 If the actual wall thickness is greater than that determined in accordance with formula 14.2.11.1-1, the maximum diameter of a not strengthened opening shall be determined in accordance with the formula below:

$$d = 8s_r(1{,}5\frac{s_r^2}{s^2} - 1) \quad [\mathrm{mm}] \qquad (14.2.13.2)$$

d – diameter of not strengthened opening, [mm];
s_r – actual wall thickness, [mm];
s – design wall thickness determined in accordance with formula 14.2.11.1-1, [mm].

14.2.13.3 Edge strengthening shall be provided for openings with diameters greater than those determined in accordance with formulae 14.2.13.1 and 14.2.13.2 to fulfil the condition below:

$$s_k\left(\frac{h^2}{s_r^2} - 0.65\right) \geq 0.65d - 1.4s_r \qquad (14.2.13.3)$$

s_k – branch piece wall thickness, [mm], (see Fig. 14.2.13.3);
d – branch piece inside diameter, [mm];
s_r – see paragraph 14.2.13.2, [mm];
h = $h_1 + h_2$,[mm], (see Fig. 14.2.13.3).

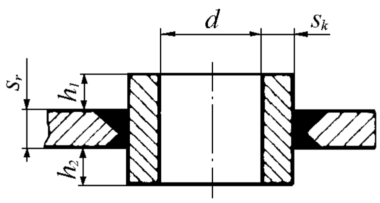


Fig. 14.2.13.3

14.2.14 Tube Plates

14.2.14.1 Thickness s_1 of flat tube plates of heat exchangers shall not be less than that determined in accordance with the formula below:

$$s_1 = 0{,}9KD_W\sqrt{\frac{P}{\sigma}} + c \quad [\mathrm{mm}] \qquad (14.2.14.1)$$

K – factor depending on the ratio of shell wall thickness s to tube plate thickness s_1; for tube plates welded to the shell, K shall be determined in accordance with diagram 14.2.14.1 on the preliminary assumption of s_1 thickness, and the calculation shall be corrected if the difference between assumed value of s_1 and that determined in accordance with formula 14.2.14.1 exceeds 5%;

for the tube plate fixed by bolts or stud-bolts between the shell and cover flanges K = 0.5;

D_W – shell inner diameter, [mm];
P – design pressure (see sub-chapter 14.2.2), [MPa];
σ – allowable stress (see paragraph 14.2.4.5), [MPa];

for heat exchangers of rigid structure where the thermal elongation factors of shell and pipe materials are different, σ shall be reduced by 10%;

φ – strength factor of tube plate weakened by holes for pipes (see paragraph 14.2.14.2);
c – design thickness allowance, [mm] (see sub-chapter 14.2.7).

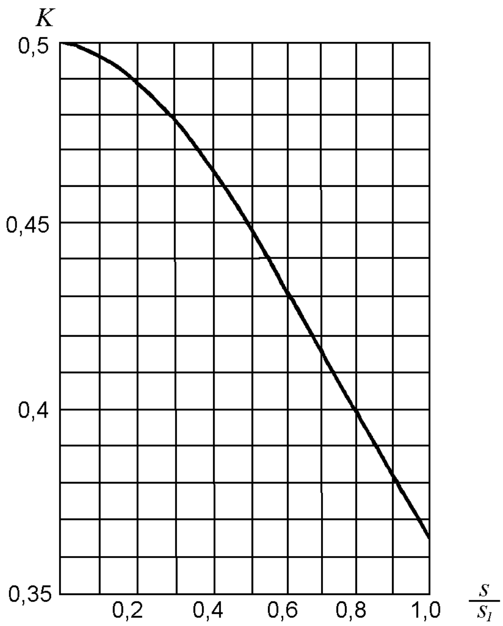


Fig. 14.2.14.1

14.2.14.2 Where $0.75 > \frac{d}{a} > 0.4$ and $\frac{D_w}{s_1} \geq 40$, the strength factor of a tube plate shall be calculated in accordance with the following formulae:

where holes are arranged in an equilateral triangle pattern:

$$\varphi = 0.935 - 0,65\frac{d}{a} \qquad (14.2.14.2\text{-}1)$$

where holes are arranged in a row or in transposition:

$$\varphi = 0.975 - 0.68\frac{d}{a_2} \qquad (14.2.14.2\text{-}2)$$

d – diameter of tube plate holes, [mm];
a – spacing of hole-axes arranged in triangle pattern, [mm];
a_2 – spacing of hole-axes arranged in row or in transposition (as well as arranged concentrically), whichever is lesser, [mm].

14.2.14.3 For quotients $\frac{d}{a} = 0.75 \div 0.80$, tube plate thickness determined in accordance with formula 14.2.14.1 shall fulfil the condition below:

$$f_{min} \geq 5d$$

f_{min} – minimum allowable cross sectional area of bridge in tube plate, [mm^2].

For values of $\frac{d}{a}$ and $\frac{D_w}{s_1}$ other than those specified above, as well as for heat exchangers with rigid structure when the difference in the mean temperatures exceeds 50 °C, the thickness of tube plates is subject to PRS consideration in each particular case.

14.2.14.4 In addition to the requirement specified in paragraph 14.2.14.1, the thickness of tube plates with expanded tubes shall fulfil the condition below:

$$s \geq 10 + 0.125d \tag{14.2.14.4}$$

Expanded connections of tubes to tube plates shall also fulfil the requirements specified in paragraphs 14.2.18.1, 14.2.18.2 and 14.2.18.3.

14.2.15 Dished Ends

14.2.15.1 Thickness of dished ends, whether unpierced or pierced, subjected to internal or external pressure (see Fig. 14.2.15.1) shall not be less than that determined in accordance with the formula below:

$$s = \frac{D_a p y}{4\sigma\varphi} + c \quad [\text{mm}] \tag{14.2.15.1}$$

s – end wall thickness, [mm];
p – design pressure, [MPa];
D_a – end outer diameter, [mm].

The end shall be flanged within the distance not less than 0.1 D_a measured from the outer edge of the end cylindrical portion (see Fig. 14.2.15.1);

φ – strength factor (see sub-chapter 14.2.6);
σ – allowable stress (see paragraph 14.2.4.5), [MPa];
y – shape factor determined in accordance with Table 14.2.15.1 depending on the ratio of the height to outside diameter of the end and on the value of weakening by holes; for intermediate values of $\frac{h_a}{D}$ and $\frac{d}{\sqrt{D_a s}}$, shape factor y may be determined by linear interpolation.

To determine y in accordance with Table 14.2.15.1, the preliminary value of s shall be initially taken from the standardised thickness series. The final value of s shall not be less than that determined in accordance with formula 14.2.15.1.

For elliptical and basket shaped ends, RW is the maximum radius of curvature.

Table 14.2.15.1

End shape	Ratio $\frac{h_a}{D_a}$	Shape factor							
		y – for flanged area and unpierced ends	y_A – for dished part of end with not strengthened holes with respect to $\frac{d}{\sqrt{D_a s}}$						y_c – for dished part of end with strengthened holes
			0.5	1.0	2.0	3.0	4.0	5.0	
Dished elliptical or basket shaped ends with $R_W = D_a$	0.20	2.9	2.9	2.9	3.7	4.6	5.5	6.5	2.4
Dished elliptical or basket shaped ends with $R_W = 0.8\ D_a$	0.25	2.0	2.0	2.3	3.2	4.1	5.0	5.9	1.8
Dished spherical ends with $R_W = 0.5\ D_a$	0.50	1.1	1.2	1.6	2.2	3.0	3.7	4.35	1.1

c – design thickness allowance, to be taken equal to:
2 mm – if subjected to internal pressure,
3 mm – if subjected to external pressure;

for wall thickness exceeding 30 mm, the above values of allowance may be reduced by 1 mm;

d – the largest diameter of not strengthened hole, [mm].

Formula 14.2.15.1 is applicable if the following conditions are fulfilled:

$$\frac{h_a}{D_a} \geq 0.18; \ \frac{s-c}{D_a} \geq 0.0025; \ R_W \leq D_a; \ r \geq 0.1D_a; l \leq 150 \text{ mm},$$

where: $l \geq 25$ mm for $s \leq 10$ mm,
$l \geq 15 + s$, [mm] for $10 < s \leq 20$ mm,
$l \geq 25 + 0.5\,s$, [mm] for $s > 20$ mm.

The symbols for dimensions of dished end elements are shown in Fig. 14.2.15.1.

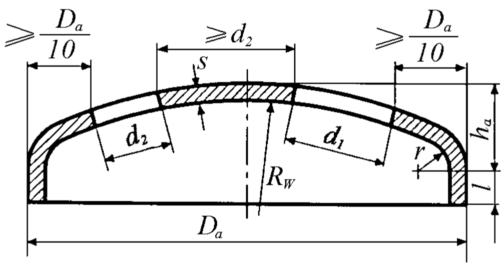


Fig. 14.2.15.1

14.2.15.2 Unpierced ends as well as ends with holes whose diameter is not greater than $4s$ and not greater than 100 mm arranged at a distance not less than $0.2D_a$ from the outer cylindrical portion of the end are also considered as unpierced ends. Not strengthened holes with the diameter less than the wall thickness, however not exceeding 25 mm, are permitted in way of the end curvature.

14.2.15.3 Dished ends subjected to external pressure, except for those made of cast iron, shall be checked for shape stability in accordance with the formula below:

$$\frac{36.6E_T}{R_W^2} \cdot \frac{(s-c)^2}{100p} > 3.3 \qquad (14.2.15.3)$$

E_T – modulus of elasticity at design temperature, [MPa],

for steel modulus of elasticity – see Table 14.2.15.3, for non-ferrous materials the modulus of elasticity value is subject to PRS acceptance;

R_W – maximum inner radius of curvature, [mm].

For other symbols – see paragraph 14.2.15.1.

Table 14.2.15.3

Design temperature T, [°C]	20	250	300	400	500
Modulus of elasticity E_T for steel, [MPa]	206 000	186 000	181 000	172 000	162 000

14.2.15.4 The minimum wall thickness of dished steel ends shall not be less than 5 mm. For ends manufactured of non-ferrous alloys, the minimum wall thickness may be reduced subject to PRS consent.

14.2.15.5 Application of dished ends of welded construction is subject to PRS consideration in each particular case.

14.2.16 Flanged End Plates

Thickness of unpierced flanged end plates (Fig. 14.2.16) subjected to internal pressure shall not be less than that determined in accordance with the formula below:

$$s = \frac{3Dp}{\sigma} + c \quad [\text{mm}] \qquad (14.2.16)$$

s – wall thickness, [mm];
p – design pressure (see sub-chapter 14.2.2), [MPa];
D – inside diameter of end plate, taken equal to shell internal diameter, [mm];
σ – allowable stress (see paragraph 14.2.4.5), [MPa];
c – design thickness allowance (see sub-chapter 14.2.7), [mm].

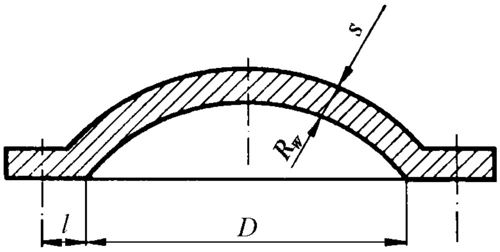


Fig. 14.2.16

Flanged end plates are allowed within a range of diameters D up to 500 mm and for working pressures not higher than 1.5 MPa. The end plate curvature radius R_W shall not be less than 1.2 D, and the distance l shall not exceed 2s.

14.2.17 Openings in Cylindrical, Spherical, Conical Walls and in Dished Ends

14.2.17.1 Strengthening arrangements shall be provided in way of openings. The following strengthening methods are permitted:

.1 wall thickness increased above the design thickness (Fig. 14.2.17.1-1 and Fig. 14.2.17.1-2);
.2 disk-shaped strengthening plates welded on the wall being strengthened (Fig. 14.2.17.1-3 and Fig. 14.2.17.1-4);
.3 welded-on pipe elements, such as branch pieces, sleeves etc. (Figures 14.2.17.1-5 to 14.2.17.1-7).

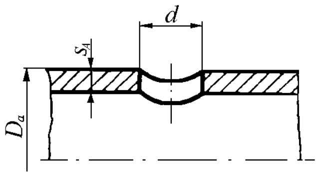


Fig. 14.2.17.1-1

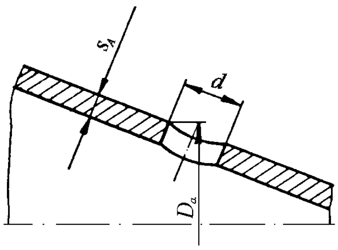


Fig. 14.2.17.1-2

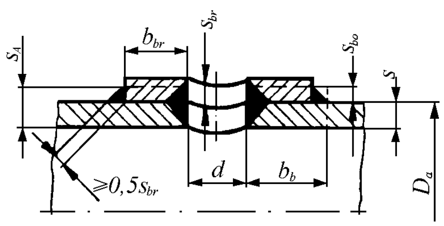


Fig. 14.2.17.1-3

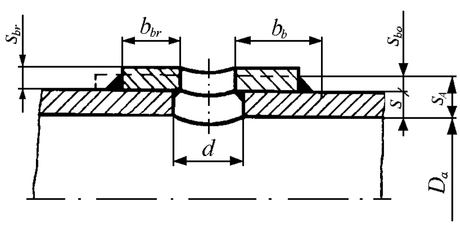


Fig. 14.2.17.1-4

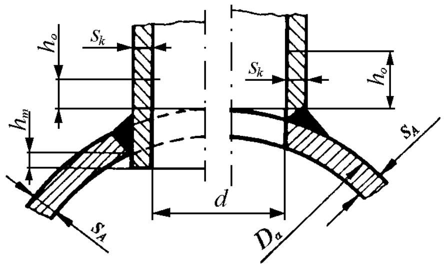


Fig. 14.2.17.1-5

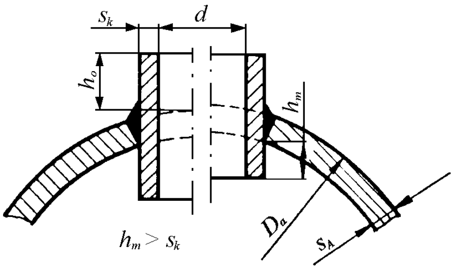


Fig. 14.2.17.1-6

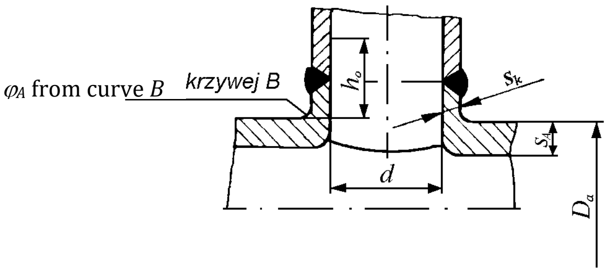


Fig. 14.2.17.1-7

It is recommended that opening strengthening elements, as shown in Figures 14.2.17.1-5 to 14.2.17.1-7, be welded with temporary backing or using other techniques ensuring proper penetration of the welded joint.

14.2.17.2 Thickness of pierced walls shall fulfil the requirements specified in sub-chapters 14.2.8 and 14.2.9 for cylindrical walls, in sub-chapter 14.2.10 – for conical walls and in sub-chapter 14.2.15 – for dished ends.

14.2.17.3 Materials used for the walls being strengthened and for strengthening elements shall have identical strength characteristics, if possible. Where the materials of strengthening elements have worse strength characteristics than the wall material, the cross-sectional area strengthening elements shall be increased respectively.

Strengthening elements shall be properly connected to the wall being strengthened.

14.2.17.4 Openings in walls shall be located at a distance equal at least triple wall thickness, however not less than 50 mm from the welded joints. The arrangement of openings at the distance less than 50 mm from the welded joints is subject to PRS consideration in each particular case.

14.2.17.5 Opening diameter (or the largest dimension of an opening other than circular) shall not exceed 500 mm. Application of openings greater than 500 mm and their strengthening methods are subject to PRS consideration in each particular case.

14.2.17.6 In general, wall thickness of tubular elements (branch pieces, sleeves or nozzles) welded to the walls of pressure vessels and heat exchangers shall not be less than 5 mm. Application of elements less than 5 mm in thickness is subject to PRS consideration in each particular case.

14.2.17.7 Opening may be strengthened by increasing design thickness of the wall. In that case, increased wall thickness s_A shall not be less than the value determined in accordance with the following formulae:

for cylindrical shells

$$s_A = \frac{pD_a}{2\sigma\varphi_A + p} + c \quad [\text{mm}] \tag{14.2.17.7-1}$$

for spherical shells

$$s_A = \frac{pD_a}{4\sigma\varphi_A + p} + c \quad [\text{mm}] \tag{14.2.17.7-2}$$

for conical shells

$$s_A = \frac{pD_a}{(2\sigma\varphi_A - p)\cos\alpha} + c \quad [\text{mm}] \tag{14.2.17.7-3}$$

s_A – required wall thickness without strengthening elements, [mm];

φ_A – strength factor of wall weakened by opening which is being strengthened, determined for the pattern curve *A* (see diagram in Fig. 14.2.17.7) depending on dimensionless parameter $\frac{d}{\sqrt{D_a(s_A - c)}}$, and to determine this parameter, the value of s_A obtained in accordance with formulae 14.2.17.7-1 to 14.2.17.7-3 shall be taken;

d – diameter of the opening (inner diameter of a branch piece, sleeve) or the dimension of an oval or elliptical opening along the longitudinal axis, [mm].

For other symbols – see paragraphs 14.2.8.2 and 14.2.10.1.

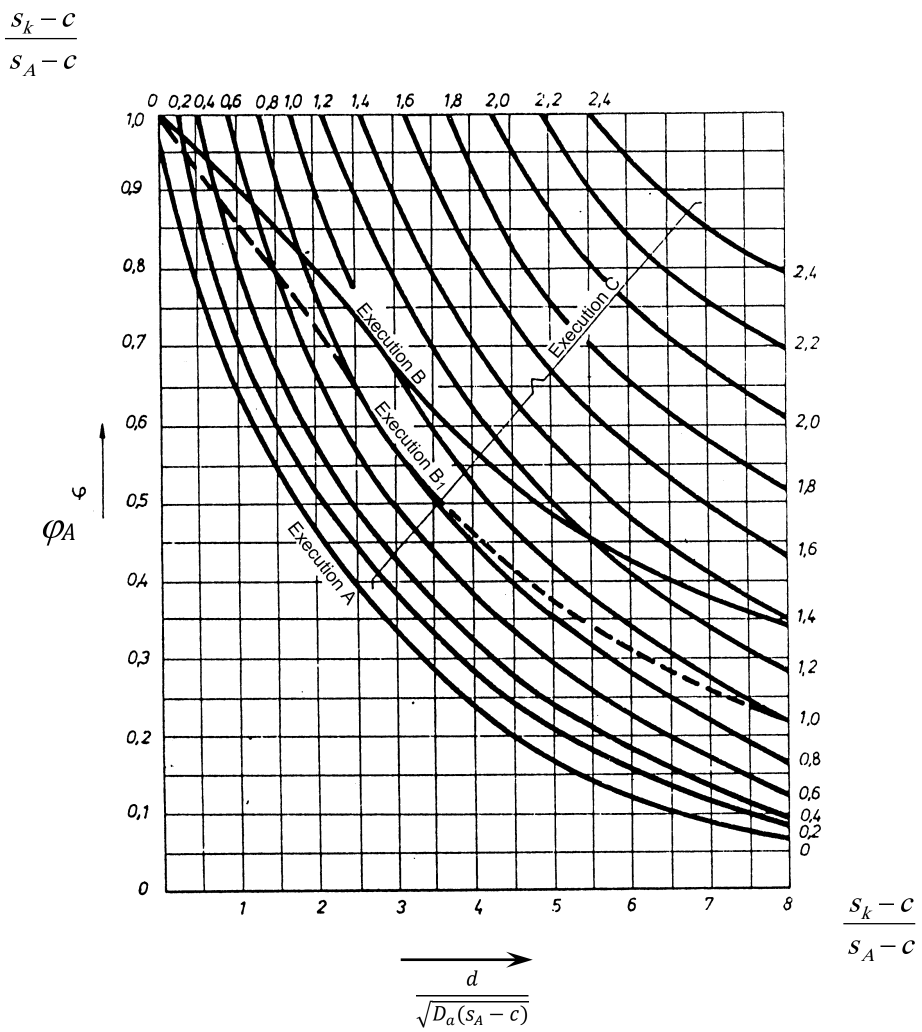


Fig. 14.2.17.7

14.2.17.8 Where disc-shaped plates are used to strengthen openings in cylindrical, spherical or conical walls, the dimensions of the strengthening plates shall be determined in accordance with the following formulae:

$$b_b = \sqrt{D_a(s_A - c)} \text{ [mm]} \qquad (14.2.17.8\text{-}1)$$

$$s_{bo} \geq s_A - s_r \qquad (14.2.17.8\text{-}2)$$

b_b – maximum effective width of plate (see Figures 14.2.17.1-3 and 14.2.17.1-4), [mm];
s_{bo} – plate thickness (see Figures 14.2.17.1-3 and 14.2.17.1-4), [mm];
s_A – total thickness of wall being strengthened and strengthening plate, determined in accordance with the requirements specified in paragraph 14.2.17.7, [mm];
s_r – actual thickness of wall being strengthened, [mm].

For other symbols – see paragraph 14.2.17.7.

Where the actual width of strengthening plate is less than that resulting from formula 14.2.17.8-1, the plate thickness shall be increased respectively, in accordance with the formula below:

$$s_{br} \geq s_{bo} \frac{1+\frac{b_b}{b_{br}}}{2} \qquad (14.2.17.8\text{-}3)$$

s_{br} – actual thickness of plate, [mm];
b_{br} – actual width of plate, [mm].

Thickness of weld seam connecting the strengthening plate to the wall shall not be less than 0.5 s_{br} (Fig. 14.2.17.1-3).

14.2.17.9 Dimensions of welded tubular elements used to strengthen openings in cylindrical, spherical and conical walls shall not be less than those determined as follows:

.1 Wall thickness s_k of a tubular element (branch piece, sleeve, etc.), [mm], shall be determined as a function of the following dimensionless parameter

$$\frac{d}{\sqrt{D_a(s_A - c)}}$$

and the strength factor φ_A , from curve *C* shown in Fig. 14.2.17.7. Quantities φ_r and s_r shall be substituted for φ_A and s_A shown in Fig. 14.2.17.7, where:
s_r – actual wall thickness, [mm];
φ_r – actual strength factor of wall with thickness s_r , as determined in accordance with formulae 14.2.8.2-1, 14.2.8.2-2, 14.2.8.3-1, 14.2.8.3-2 and 14.2.10.1.2 by re-arranging the said formulae to determine φ.

Ratio 14.2.17.7:

$$\frac{s_k - c}{s_A - c}$$

obtained from the diagram in Fig. 14.2.17.7 shall be used to determine the minimum thickness s_k , [mm] of a branch piece or sleeve. In this ratio, actual thickness s_r shall be substituted for s_A.

.2 The minimum design height, h_0, [mm] of a tubular strengthening element shall be determined in accordance with the formula below:

$$h_0 = \sqrt{d(s_k - c)} \qquad (14.2.17.9.2\text{-}1)$$

If actual height, h_r, of a tubular strengthening element is less than that determined in accordance with formula 14.2.17.9.2-1, thickness s_k shall be increased respectively as follows:

$$s_{kr} = s_k \frac{h_0}{h_r} \quad [\text{mm}] \qquad (14.2.17.9.2\text{-}2)$$

14.2.17.10 Dimensions of the elements strengthening openings in dished ends shall be determined as follows:

.1 For openings strengthened by increasing the dished end wall thickness, factor y_A obtained from Table 14.2.15.1 shall be substituted for factor y in formula 14.2.15.1.

.2 For openings strengthened by means of disk-shaped strengthening plates, the plate dimensions shall be determined in accordance with paragraph 14.2.17.8, and the total thickness of the strengthened end wall, s_A, shall be determined in accordance with the formula below:

$$s_A = \frac{p(R_W+s)y_0}{2\sigma\varphi_A} + c \quad [\text{mm}] \qquad (14.2.17.10.2)$$

R_W – inner radius of curvature in way of opening, [mm];
y_0 – shape factor determined in accordance with Table 14.2.15.1.

For other symbols – see paragraphs 14.2.15.1 and 14.2.17.7.

.3 The dimensions of tubular elements strengthening openings shall be determined in accordance with paragraph 14.2.17.9, except that the expression $2(0.5D_a + s)$ shall be substituted for D_a in the following dimensionless parameter

$$\frac{d}{\sqrt{D_a(s-c)}}$$

and the actual strength factor φ for the dished end wall thickness, s, shall be determined in accordance with formula 14.2.15.1, assuming $\varphi = \varphi_A$, $y = y_0$ and $s = s_A$ (see paragraph 14.2.15.1).

14.2.17.11 For through tubular strengthening elements with the inward projecting portion hm ≥ sr (Figures 14.2.17.1-5 and 14.2.17.1-6), thickness of the tubular element may be reduced by 20%, however its thickness shall not be less than that required for the design pressure.

14.2.17.12 The ratio of a tubular strengthening element thickness, s_k, to the thickness of wall being strengthened, s , shall not be greater than 2.4. If this ratio is taken as more than 2.4, for construction reasons, tubular strengthening element thickness s_k shall be assumed not greater than 2.4 times the thickness of the wall being strengthened in the calculation.

14.2.17.13 Disk-shaped strengthening plates and tubular strengthening elements may also be used in combination (Fig.. 14.2.17.13). In that case, the dimensions of strengthening elements shall be determined taking account of the requirements for both the disk-shaped and tubular strengthening element.

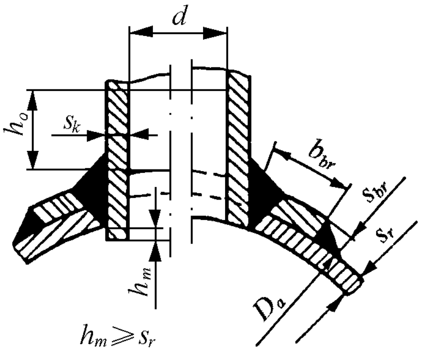


Fig. 14.2.17.13

14.2.17.14 For branch pieces drawn from the wall being strengthened (Fig. 14.2.17.1-7), thickness s_A shall not be less than that determined in accordance with formulae 14.2.17.7-1 to 14.2.17.10.2.

Strength factor φ_A the wall weakened due to a drawn branch piece shall be obtained from diagram 14.2.17.7 as follows:

for $\frac{d}{D_a} \leq 0.4$ – from curve B,

for $\frac{d}{D_a} = 1.0$ – from curve B_1,

for $0.4 < \frac{d}{D_a} < 1.0$ – by interpolation of curves B and B_1.

Thickness of a drawn branch shoulder, s_k, shall not be less than that determined in accordance with the formula below:

$$s_k \geq s_A \frac{d}{D_a} \quad \text{[mm]} \qquad (14.2.17.14)$$

however not less than that required for the design pressure.

14.2.17.15 The effect of adjacent openings may be disregarded provided that:

$$(l + s_{kr1} + s_{kr2}) \geq 2\sqrt{D_a(s_r - c)} \qquad (14.2.17.15\text{-}1)$$

$(l + s_{kr1} + s_{kr2})$ –distance between two adjacent openings (Figures 14.2.17.15-1 and 14.2.17.15-2), [mm];
D_a – outside diameter of wall being reinforced, [mm];
s_r – actual thickness of wall being reinforced, [mm];
c – design thickness allowance, [mm], (see sub-chapter 14.2.7), [mm].

Where $(l + s_{kr1} + s_{kr2}) < \sqrt{D_a(s_r - c)}$, the stress occurring in the wall cross section between the openings due to design pressure shall be chaecked. Both longitudinal and lateral stresses in that section shall not exceed the allowable values determined in accordance with the formula below:

$$\frac{F}{f_c} \leq \sigma \qquad (14.2.17.15\text{-}2)$$

σ – allowable stress (see paragraph 14.2.4.5), [MPa];
F – load exerted by the design pressure upon the cross-section between openings (see paragraph 14.2.17.16), [N];
f_c – cross sectional area between openings (see paragraph 14.2.17.17), [mm[2]].

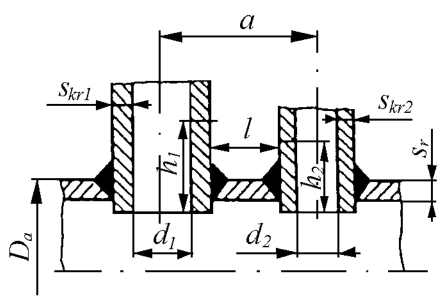


Fig. 14.2.17.15-1

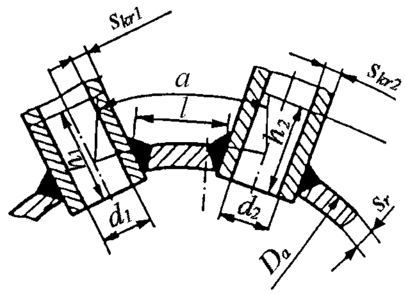


Fig. 14.2.17.15-2

14.2.17.16 Load exerted by the design pressure on the cross sectional area between two openings shall be determined as follows:

.1 for openings arranged longitudinally along a cylindrical wall:

$$F_a = \frac{Dpa}{2} \quad [\text{N}] \tag{14.2.17.16.1}$$

.2 for openings arranged circumferentially in cylindrical or conical walls, as well as in spherical walls:

$$F_b = \frac{Dpa}{4} \quad [\text{N}] \tag{14.2.17.16.2}$$

.3 for openings in dished ends:

$$F_b = \frac{R_B pay}{2} \quad [\text{N}] \tag{14.2.17.16.3-1}$$

a – spacing between two adjacent openings, measured at the outside circumference, as shown in Fig. 14.2.17.15-2, [mm];
D – inside diameter (for conical walls measured at the centre of the opening), [mm];
p – design pressure, [MPa];
R_B – inner radius of curvature (see paragraph 14.2.17.10), [mm];
y – shape factor (see paragraph 14.2.15.1).

Where openings are arranged in cylindrical walls with a diagonal pitch, the load in question shall be determined in accordance with formula 14.2.17.16.2, and the obtained results shall be multiplied by the following factor:

$$K = 1 + \cos^2\alpha \tag{14.2.17.16.3-2}$$

α – angle between the line of a row of openings and longitudinal axis.

14.2.17.17 For tubular strengthening elements, cross sectional area, f_c, [mm^2] between two adjacent openings shall be determined in accordance with the formula below:

$$f_c = l(s - c) + 0{,}5[h_1(s_{kr1} - c) + h_2(s_{kr2} - c)] \ [\text{mm}^2] \tag{14.2.17.17-1}$$

h_1 and h_2 – height of strengthening elements, [mm], determined in accordance with the following formulae:

for blind strengthening elements:

$$h_{1,2} = h_0 + s \tag{14.2.17.17-2}$$

for through strengthening elements:

$$h_{1,2} = h_0 + s + h_m \tag{14.2.17.17-3}$$

l – width of bridge between two adjacent openings (Figures 14.2.17.15-1 and 14.2.17.15-2), [mm];
s – thickness of wall being reinforced, [mm];
s_{kr1} and s_{kr2} – thicknesses of tubular strengthening elements (Figures 14.2.17.15-1 and 14.2.17.15-2), [mm];
c – design thickness allowance, [mm], (see sub-chapter 14.2.7);
h_0 – design height of tubular strengthening element (see formula 14.2.17.9.2-1), [mm];
h_m – design height of tubular strengthening element projecting inwards (see Figures 14.2.17.1-5, 14.2.17.1-6 and 14.2.17.13), [mm].

For openings to be strengthened by other means (combined or disc-shaped strengthening elements, etc.), the values of f_c shall be determined in accordance with the same procedure.

14.2.17.18 For drawn branch pieces arranged in a row, strength factor φ, determined for this row in accordance with formula 14.2.6.2.1, shall not be less than strength factor φ_A, obtained from curves B and B1 in Fig. 14.2.17.7. For $\varphi < \varphi_A$, the value of φ shall be used to determine the wall thickness in accordance with paragraph 14.2.17.14.

This requirement also applies to welded branch pieces arranged in a row, whose thickness is determined only for the internal pressure effect.

14.2.18 Flared Tube Joints in Tube Plates

14.2.18.1 For flared tubes, the flared belt length in the tube plate shall not be less than 12 mm. The flared joints for working pressures above 1.6 MPa shall be made with sealing grooves.

14.2.18.2 Flared tube joints shall be checked for secure connection of the tubes in tube plates due to axial loads. The tubes are considered as securely connected, if the value obtained in accordance with the following formula:

$$\frac{pf_s}{20sl} \qquad (14.2.18.2)$$

is not greater than:

15 – for plain tube joints,
30 – for joints with sealing grooves,
40 – for joints with tube flanging;
p – design pressure (see paragraph 14.2.2), [MPa];
f_s – maximum sector of the area of wall being strengthened per tube, [mm^2]. This sector is bounded by lines passing at right angles through the centres of the lines connecting the axis of tube in question with the adjacent tubes.
s – wall thickness of tube, [mm];
l – expansion belt length, [mm].

Length of the flared belt of tubes l shall not be taken greater than 40 mm.

14.2.18.3 Length of the flared belt of plain tubes shall not be less than that determined in accordance with the formula below:

$$l = \frac{pf_sK_r}{q} \quad [\text{mm}] \qquad (14.2.18.3\text{-}1)$$

where:

K_r = 5.0 – safety factor of flared joint;
p, f_s – see paragraph 14.2.18.2;
q – strength of pipe joint per l mm of flared belt, determined experimentally in accordance with the formula below, [N/mm]:

$$q = \frac{F}{l_1} \quad [\text{N/mm}] \qquad (14.2.18.3\text{-}2)$$

where:

F – axial force necessary to extract the flared tube from the tube plate, [N];
l_1 – length of flared belt used for experimental determining of the value of q [mm].

15 PIPING SYSTEMS

15.1 Class, Material, Manufacture and Application of Piping

15.1.1 The requirements specified in this sub-chapter apply to piping systems, normally employed in ships and made of carbon steel, carbon-manganese steel, alloy steel or non-ferrous materials, specified in the scope of the considered documentation (see also paragraph 15.1.8).

The requirements do not cover open-ended exhaust gas lines from internal combustion engines.

15.1.2 For the purpose of determining the scope of tests, joint type, heat treatment and welding procedure, piping systems – depending on their service and the conveyed medium parameters – are subdivided into classes as specified in Table 15.1.2.

Table 15.1.2
Piping classes

Piping for:	Class I	Class II	Class III
Toxic [2] or strongly corrosive media	Without special safeguards [1]	With special safeguards [1]	
Flammable media with service temperature above the flashpoint or with the flashpoint below 60 °C, liquefied gases	Without special safeguards [1]	With special safeguards [1]	
Oil fuel, lubricating oil flammable hydraulic oil, oil cargo [3]	p >1.6 or t >150	Any combination of pressure p and temperature t beyond the scope of class I or III – see Fig. 15.1.2	$p \le 0.7$ and $t \le 60$
Other media [3], [4], [5]	p >4.0 or t >300		$p \le 1.6$ and $t \le 200$

Notes to Table 15.1.2

1) Special safeguards are intended to reduce the possibility of leakage and prevent damage in the immediate vicinity or potential risk of ignition sources, such safeguards may include pipe ducts, shielding, screening etc.
2) Pipelines conveying toxic media belong to Class I.
3) p – design pressure, [MPa], (see paragraph 15.2.1).
t – design temperature, [^{o} C], (see paragraph 15.2.1).
4) Including water, air, gas, lubricating oil and non-combustible hydraulic oil.
5) Open-ended pipes (drains, overflow pipes, air pipes and exhaust gas lines from safety valves) belong to Class III.

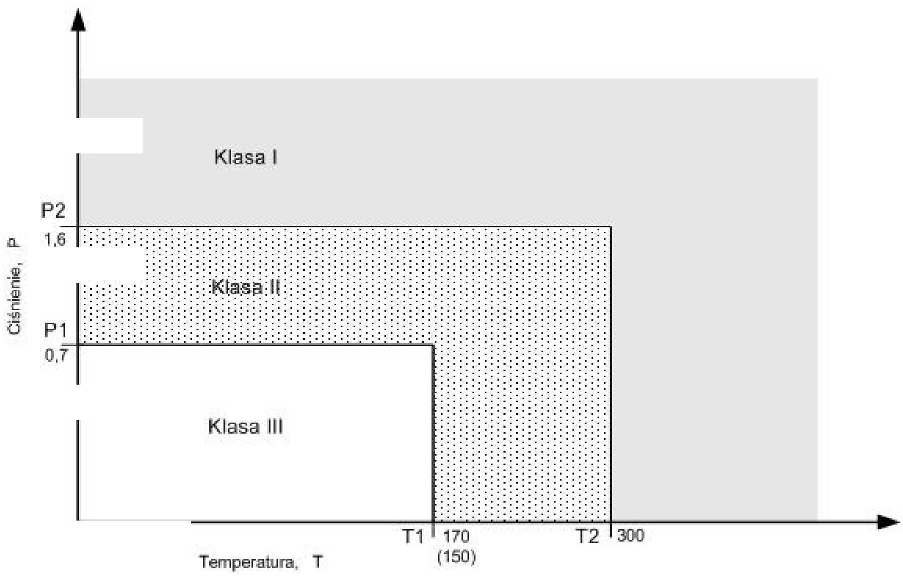


Fig. 15.1.2

15.1.3 Materials intended for pipes, valves and fittings as well as the methods of their testing shall fulfil the requirements specified in *Part IX – Materials and Welding* of the *Rules for the Construction and Classification of Sea-going Ships*.

Materials for pipes, valves and fittings intended to be exposed to strongly corrosive media are subject to PRS consideration in each particular case.

Fabrication of piping systems shall fulfil the requirements specified in *Publication No. 23/P – Pipelines Prefabrication*.

15.1.4 Steel pipes intended for Class I or Class II piping systems shall be seamless, hot or cold drawn pipes. Welded pipes, approved by PRS as equivalent to seamless pipes, may also be used.

Pipes, valves and fittings made of carbon steel and carbon-manganese steel shall be used only for media with temperature not exceeding 400 °C and made of low alloy steel – for media with temperature not exceeding 500 °C.

Such steel may also be used for media with temperature higher than those stated above, provided that at such temperatures their mechanical properties and creep strength limit within 100 000 hours are in accordance with the standards in force and that such characteristics are guaranteed by the manufacturer.

Pipes, valves and fittings for media with temperature exceeding 500 °C shall be made of alloy steel.

15.1.5 Copper and copper alloy pipes shall be seamless or other type approved by PRS. These pipes for class I and class II piping systems shall be seamless.

Copper and copper alloy pipes, valves and fittings shall not be used for media with temperature exceeding:
200 °C – for copper and copper-aluminium alloys,
260 °C – for bronze,
300 °C – for copper-nickel alloys.

15.1.6 Pipes, vales and fittings of nodular ferritic cast iron may be accepted for media with temperatures not exceeding 350 °C for the purposes including:
– bilge, ballast and cargo pipes within double bottom or cargo tanks,
– side valves, fittings and flanged branch pieces as well as valves and fittings installed on collision bulkhead and on fuel and oil tanks.

Application of nodular ferritic cast iron for other valves, fittings and pipes as well as for Class II or Class III piping is subject to PRS consideration in each particular case.

15.1.7 Grey cast iron may be used for class III piping systems.

The use of grey cast-iron pipes, valves and fittings in other service piping systems is subject to special consideration by PRS.

Grey cast-iron shall not be used for:
– pipes, valves and fittings for media with temperature exceeding 220°C,
– pipes, valves and fittings subjected to hydraulic shock or excessive strains and vibrations,
– pipes, valves and fittings of fire-extinguishing systems,
– pipes connected directly to the shell plating,
– valves and fittings installed on the shell plating or collision bulkhead,
– valves and fittings installed directly on oil fuel, lubricating oil or other flammable oil tanks under hydrostatic pressure, unless proper means have been provided to protect them against damage.

15.1.8 The requirements for plastic pipes as well as conditions for their application in ships are specified in *Publication No. 53/P – Plastic Pipelines on Ships*.

15.1.9 Type and construction of non-metallic flexible joints used in systems whose documentation is required to be submitted to PRS are subject to PRS approval.

Flexible joints shall be fabricated as assemblies complete with flange or screw connection pieces ready to be inserted in a pipeline. Installation of flexible joints in pipelines by means of hose clips is not permitted. The joints shall be located in conspicuous and readily accessible positions. Arrangement of cut-off valves shall be such as to allow replacement of flexible joints without stopping the machinery other than that served by the joint.

Flexible joints shall be fire-resistant when used in piping:
– conveying oil fuel or lubricating oil,
– serving watertight doors,
– leading to openings in shell plating (including bilge system),
– conveying other flammable oil, if the joint damage may cause hazard to the ship or crew and passengers.

Flexible joint is considered as fire-resistant if it endures exposition to a fire of temperature 800 °C for 30 minutes with flowing water at the maximum service pressure. The outlet temperature shall not be less than 80 °C and shall be recorded throughout the test [*)].

Material for flexible hoses shall be selected taking account of the hose intended use for certain type of fluid, its pressure, temperature and ambient conditions.

The hose bursting pressure shall be at least 4-times the design pressure.

The length of hoses shall be such as to ensure flexibility of joints and normal operation of the machinery.

15.2 Pipe Wall Thickness

15.2.1 The formulae given below are applicable when the ratio of the pipe outside diameter to its inside diameter is not greater than 1.7.

Wall thickness s for straight or bent metal pipe exposed to internal pressure (considering the requirements specified in paragraph 15.2.2) shall not be less than that determined in accordance with the formula below:

$$s = s_0 + b + c \quad \text{[mm]} \tag{15.2.1-1}$$

and in no case lesser than the value specified in Table 15.2.1-1.

$$s_0 = \frac{dp}{2\sigma_d \varphi + p} \quad \text{[mm]} \tag{15.2.1-2}$$

where:

d – outside diameter of pipe, [mm];
p – design pressure, [MPa], however:
  – for piping for oil fuel heated up to a temperature exceeding 60 °C – not less than 1.4 MPa,
  – for piping of CO_2 fire extinguishing systems – the design pressure shall be taken in accordance with the notes to Table 3.6.1 in *Part V – Fire Protection*;
φ – safety factor equal to 1.0 for seamless pipes and for welded pipes, considered as equivalent to seamless pipes; for all other welded pipes, the value of safety factor will be subject to PRS consideration in each particular case;
b – allowance for a reduction of pipe wall thickness due to bending; the value b shall be so determined that the calculated stress in the bend, due to the internal pressure only, does

[*)] Fire test of the flexible joint with flowing water at a pressure of at least 0.5 MPa and subsequent hydraulic pressure test with twice the design pressure is an alternative.

not exceed the permissible stress; where the exact value of thickness reduction at the bend is not available, the value b may be determined by the following formula:

$$b = 0{,}4(d/R)s_0 \ \text{[mm]} \quad (15.2.1\text{-}3)$$

R – mean inside bend radius, [mm];
c – corrosion allowance, [mm], to be taken:
- for steel pipes – in accordance with Table 15.2.1-2,
- for non-ferrous metal pipes – in accordance with Table 15.2.1-3;

σ_d – allowable stress, [MPa], to be taken as follows:
- for steel pipes – the lowest out of the following value:

$R_m/2.7$; $R_e^t/1.8$ or $R_{0.2}^t/1.8$; $R_{z/100000/t}/1.8$ and $R_{l/100000/t}/1.0$

where:

R_m – minimum tensile strength, [MPa],
R_e^t, $R_{0,2}^t$ – minimum yield point or 0.2% proof stress, [MPa], at the design temperature t, [°C],
$R_{z/100000/t}$ – average creep stress, [MPa], to produce rupture in 10^5 hours, at design temperature t, [°C],
$R_{l/100000/t}$ – average stress, [MPa], to produce 1% creep in 10^5 hours, at design temperature t, [°C].

Notes:
1. The above defined safety factor 1.8 may be reduced to 1.6 subject to PRS consent in each particular case.
2. PRS may require $R_{l/100000/t}$ value to be taken into account, if necessary.

- for high alloy steel pipes σ_d is subject to PRS consideration in each particular case;
- for copper and copper alloy pipes, σ_d shall be determined in accordance with Table 15.2.1-4;

t – design temperature [°C], to be considered for determining the allowable stress, is the maximum temperature of the medium inside the pipe; in special cases, the design temperature is always subject to PRS consideration.

Table 15.2.1-1
Minimum wall thickness of pipes, *s* [mm]

Nominal diameter [mm]	External diameter [mm]	Steel pipes						Austenitic stainless steel pipes	Copper pipes	Copper alloy pipes
		A	**B**	**C**	**D**	**E**	**F**			
	< 8	–	–	–	–	–	–	–	1,0	0.8
6	8.0	–	–	–	–	–	–	–	1.0	0.8
	10.2	1.6	–	–	–	–	–	1.0	1.0	0.8
	12	1.6	–	–	–	–	–	1.0	1.2	1.0
8	13.5	1.8	–	–	–	–	–	1.0	1.2	1.0
10	16	1.8	–	–	–	–	–	1.0	1.2	1.0
	17.2	1.8	–	–	–	–	–	1.0	1.2	1.0
	19.3	1.8	–	–	–	–	–	–	1.2	1.0
15	20	2	–	–	–	–	–	–	1.2	1.0
	21.3	2	–	3.2	–	3.2	2.6	1.6	1.2	1.0
	25	2	–	3.2	–	3.2	2.6	1.6	1.5	1.2
20	26.9	2	–	3.2	–	3.2	2.6	1.6	1.5	1.2

Nominal diameter [mm]	External diameter [mm]	Steel pipes						Austenitic stainless steel pipes	Copper pipes	Copper alloy pipes
		A	**B**	**C**	**D**	**E**	**F**			
	30	2	–	3.2	–	4	3.2	1.6	1.5	1.2
25	33.7	2	–	3.2	–	4	3.2	1.6	1.5	1.2
	38	2	4.5	3.6	6.3	4	3.2	1.6	1.5	1.2
32	42.4	2	4.5	3.6	6.,3	4	3.2	1.6	1.5	1.2
	44.5	2	4.5	3.6	6.3	4	3.2	1.6	1.5	1.2
40	48.3	2.3	4.5	3.6	6.3	4	3.2	1.6	2.0	1.5
	51	2.3	4.5	4	6.3	4.5	3.6	–	2.0	1.5
	54	2.3	4.5	4	6.3	4.5	3.6	–	2.0	1.5
50	57	2.3	4.5	4	6.3	4.5	3.6	–	2.0	1.5
	60.3	2.3	4.5	4	6.3	4.5	3.6	2.0	2.0	1.5
	63.5	2.3	4.5	4	6.3	5	3.6	2.0	2.0	1.5
	70	2.6	4.5	4	6.3	5	3.6	2.0	2.0	1.5
65	76.1	2.6	4.5	4.5	6.3	5	3.6	2,0	2.0	1.5
	82.5	2.6	4.5	4.5	6.3	5.6	4	2,0	2.0	1.5
80	88.9	2.9	4.5	4.5	7.1	5.6	4	2,0	2.5	2.0
90	101.6	2.9	4.5	4.5	7.1	6.3	4	–	2.5	2.0
100	108	2.9	4.5	4.5	7.1	7.1	4.5	-	2.5	2.0
	114.3	3.2	4.5	4.5	8	7.1	4.5	2.3	2.5	2.0
	127	3.2	4.5	4.5	8	8	4.5	2.3	2.5	2.0
	133	3.6	4.5	4.5	8	8	5.0	2.3	3.0	2.5
125	139.7	3.6	4.5	4.5	8	8	5.0	2.3	3.0	2.5
	152.4	4	4.5	4.5	8.8	8.8	5.6	2.3	3.0	2.5
150	159	4	4.5	4.5	8.8	8.8	5.6	2.3	3.0	2.5
	168.3	4	4.5	4.5	8.8	8.8	5.6	2.3	3.0	2.5
	177.8	4.5	5	5	8.8	–	–	–	3.0	2.5
175	193.7	4.5	5.4	5.4	8.8	–	–	–	3.5	3.0
200	219.1	4.5	5.9	5.9	8.8	–	–	2.6	3.5	3.0

A – Piping systems other than those mentioned under B, C, D, E, F, G or in note 8.
B – Air, overflow and sounding pipes of structural tanks, except those mentioned under D, and drain pipes covered by 15.9.4.
C – Sea-water pipes (bilge, ballast, cooling water, fire systems, etc.) – except those mentioned under D.
D – Bilge, ballast, air, overflow and sounding pipes in way of oil fuel tanks and bilge, air, overflow, sounding and oil fuel pipes in way of ballast tanks as well as air pipes above open deck.
E – Carbon dioxide fire extinguishing piping – from cylinders to distribution valves (see notes 2, 3, 6, 7).
F – Carbon dioxide fire extinguishing piping – from distribution valves to discharge nozzles (see notes 2, 3, 6, 7).

Notes to Table 15.2.1-1:

1) Wall thickness and pipe diameters listed in the Table are determined in accordance with ISO Recommendations R 336. Minor variations resulting from the use of other standards may be accepted.
2) For the values listed in the Table no allowances need to be made for negative manufacturing tolerance or reduction in thickness due to bending.
3) For diameters greater than those listed in the Table, the minimum wall thickness is subject to PRS consideration in each particular case.
4) For pipes effectively protected against corrosion, upon agreement with PRS, the wall thickness values specified in columns 4, 5 and 6 may be reduced, but not more than by 1 mm.
5) For sounding pipes the wall thickness values specified in columns 4 and 6 apply to these parts located outside the tanks for which the pipes are intended.
6) For threaded pipes, the wall thickness listed is the minimum thickness in the threaded part of the pipe.
7) The pipes listed under F and G shall be galvanized at least inside. Subject to PRS consent, short section of pipes installed in engine room need not be galvanized.
8) The Table does not cover exhaust gas lines. Minimum wall thickness values for these lines are subject to PRS consideration in each particular case. The recommended minimum wall thickness of exhaust gas pipes is 4 mm.

Table 15.2.1-2
Corrosion allowance for steel pipes, *c* [mm]

Piping service	*c*
Compressed air systems	1.0
Hydraulic oil systems	0.3
Lubricating oil systems	0.3
Oil fuel systems	1.0
Fresh water systems	0.8
Sea-water systems	3.0

Notes to Table 15.2.1-2:

1) If the pipes are effectively protected against corrosion then – subject to PRS consent in each particular case – the corrosion allowance may be reduced, however not more than by 50%.
2) For special alloy steel pipes with sufficient corrosion resistance, corrosion allowance *c* may be reduced down to zero.
3) For pipes passing through tanks, the values specified in the Table for inside medium shall be increased by corrosion allowance taking account of the ambient conditions – in accordance with the Table.

Table 15.2.1-3
Corrosion allowance for copper and copper alloy pipes, *c* [mm]

Pipe material	*c* [mm]
Copper and copper alloys except those with lead content	0.8
Copper-nickel alloys (with nickel content 10% and more)	0.5

Note to Table 15.2.1-3:

For special alloy pipes having sufficient resistance to corrosion, corrosion allowance *c* may be reduced to zero.

Table 15.2.1-4
Allowable stress σ_d [MPa] for copper and copper alloys depending on temperature of medium

Pipe material	Material condition	R_m [MPa]	Temperature of medium, [°C]										
			50	75	100	125	150	175	200	225	250	275	300
Copper	Annealed	215	41	41	40	40	34	27.5	18.5	–	–	–	–
Aluminium brass	Annealed	325	78	78	78	78	78	51	24.5	–	–	–	–
Copper-nickel alloys 95/5 and 90/10	Annealed	275	68	68	67	65,5	64	62	59	56	52	48	44
Copper-nickel alloy 70/30	Annealed	365	81	79	77	75	73	71	69	67	65.5	64	62

Notes to Table 15.2.1-4:

1) Intermediate values shall be determined by linear interpolation.
2) For materials not included in the Table, the allowable stress is subject to PRS consideration in each particular case.

15.2.2 For pipes with fabricated with negative tolerance of thickness, the wall thickness shall be determined in accordance with the formula below:

$$s_1 = \frac{s}{1-0.01a} \tag{15.2.2}$$

where:

s – wall thickness determined in accordance with formula 15.2.1-1, [mm];
a – negative tolerance of pipe thickness, [%].

15.3 Pipe Connections

The following pipe connections of pipe lengths may be used:

– direct welding,
– flanges,
– threaded joints,
– mechanical joints.

Each of the above mentioned connections shall be made in accordance with a recognised standard or of a proven construction for the intended application and shall be approved by PRS.

15.3.1 Welded Connections

15.3.1.1 Welding and non-destructive testing of welds shall be performed in accordance with the requirements specified in Publication No. 23/P – Pipelines Prefabrication and Part IX – Materials and Welding of the Rules for Classification and Construction of Sea-going Ships.

15.3.1.2 Butt welded joints shall be of full penetration type. Such joints made with special provision for a high quality of root side*) – for all classes, irrespective of outside diameter. Butt welded joints without special provision for a high quality of root side may be used for piping systems of class II and III, irrespective of outside diameter.

15.3.1.3 Slip-on sleeve and socket welded joints shall have sleeves, sockets and weldments conformant to a recognised standard. Acceptable applications of pipe connections in the relevant class of piping are specified in Table 15.3.1.3.

Table 15.3.1.3
Application of slip-on and socket welded joints

<table>
<tr><th rowspan="2">Class of piping</th><th rowspan="2">Pipe outside diameter [mm]</th><th colspan="2">Type of connection</th></tr>
<tr><th>Sleep-on sleeve</th><th>Socket welded joint</th></tr>
<tr><td>I</td><td rowspan="2">≤ 88.9</td><td colspan="2" rowspan="2">Both types are permitted except piping systems:
– conveying toxic media,
– subjected to fatigue loads,
– where severe corrosion is expected to occur</td></tr>
<tr><td>II</td></tr>
<tr><td>III</td><td>Irrespective of pipe diameter</td><td colspan="2">Both types are permitted without limitation</td></tr>
</table>

15.3.2 Flange Connections

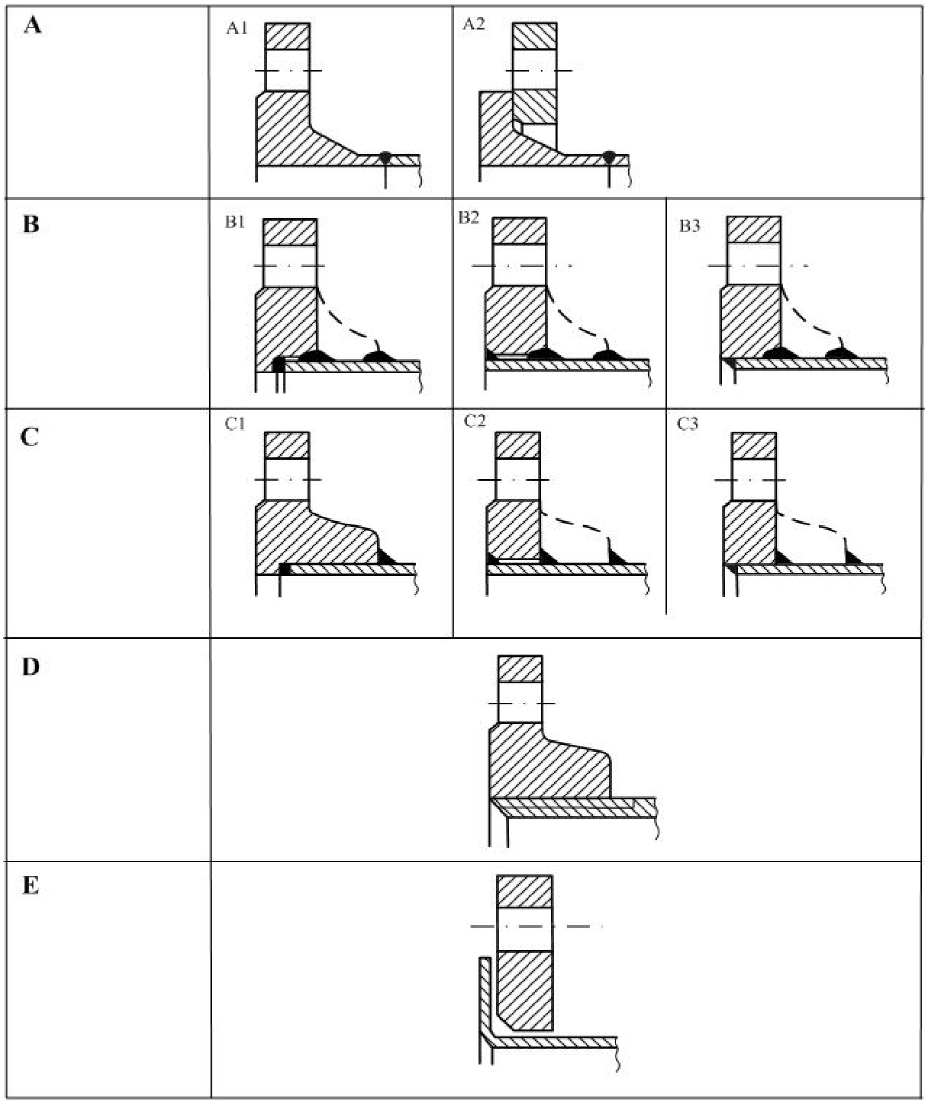
15.3.2.1 Dimensions and type of flanges as well as bolts used to connect them shall conform to a recognised standard. Non-standard flanges and bolts may be used subject to PRS consent in each particular case.

15.3.2.2 The material of gaskets shall be resistant to the effect of the medium conveyed and to the surrounding environment. The construction of gaskets shall correspond to the design pressure and temperature whereas their dimensions and shape shall conform to a recognised standard. Gaskets of connections in fuel oil piping shall ensure tightness at the temperature of the conveyed medium not less than 120 °C.

15.3.2.3 Acceptable types of flanges for piping connections are shown in Table 15.3.2.3. Division of the flanges results from their design and method of their connection to the pipes. Other flanges may be used for piping connections subject to PRS consent in each particular case.

*) The expression "special provision for a high quality of root side" means that butt welds were accomplished as double welded or by use of a backing ring or inert gas back-up on first pass. Subject to PRS consent, other similar methods may be accepted.

Table 15.3.2.3
Acceptable types of flanges for piping connections



Note to Table 15.3.2.3:

In flanges of type D taper pipe thread shall be used. The minor diameter of pipe thread shall not be appreciably less than the pipe outside diameter. For certain types of thread, after the flange has been screwed hard home, the pipe shall be expanded into the flange.

15.3.2.4 Depending on the pipeline class and type of the conveyed medium, flange connections shown in Table 15.3.2.4.

Table 15.3.2.4
Required types of flange connection for particular classes of piping and types of media

Class of piping	Toxic, strong corrosive, flammable media [4]	Lubricating and fuel oil	Other media [1], [2], [3], [4], [5]
I	A, B [6]	A, B	A, B
II	A, B, C	A, B, C	A, B, C, D [5]
III	Other media	A, B, C, E	A, B, C, D, E

Notes to Table 15.3.2.4:

1) Including water, air, gas and hydraulic oil.
2) Type E flanges shall be used for water pipes and open-ended lines only.
3) Only type A where design temperature exceeds 400 °C.
4) Only type A where design pressure exceeds 1.0 MPa.
5) Types D and E shall not be used for pipes where design temperature exceeds 250 °C.
6) Type B flanges may be used for pipes with outside diameter not greater than 150 mm only.

When selecting flange type for pipe connections, outside loads and cyclic loads imposed on pipelines as well as location of pipelines on board the ship shall be taken into account.

15.3.3 Slip-on Threaded Joints

15.3.3.1 Slip-on threaded joints having threads where pressure-tight joints are made on pipes with parallel or tapered threads shall conform to a recognised standard.

15.3.3.2 Screwed joints may be used in piping of CO_2 fire extinguishing systems within the spaces covered only.

15.3.3.3 Screwed joints shall not be used in pipelines conveying flammable or toxic media or in those pipelines where crevice corrosion, appreciable erosion or changing loads are expected to occur.

15.3.3.4 Slip-on threaded joints acceptable for piping connections with regard to the pipe outside diameter and thread type are shown in Table 15.3.3.4. Slip-on threaded joints conformant to a recognised standard may be used for greater pipe diameters than those specified in Table 15.3.3.4 subject to PRS consent in each particular case.

Table 15.3.3.4
Acceptable applications of slip-on threaded joints

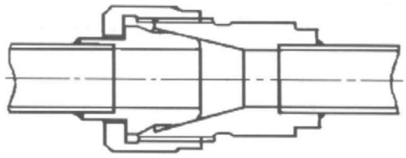
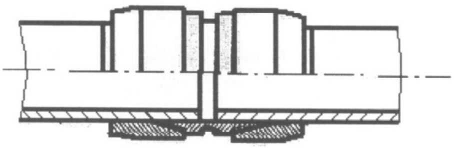
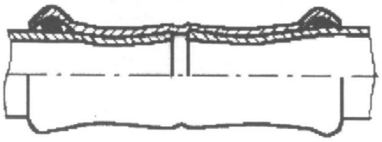
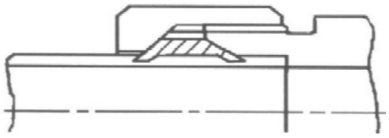
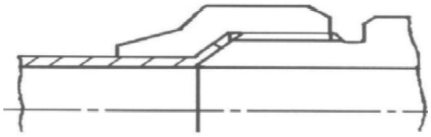
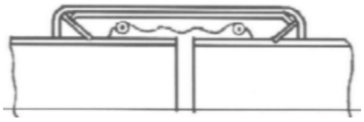
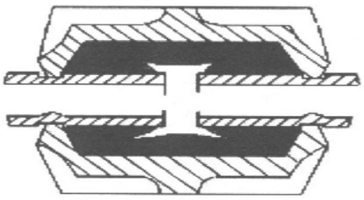
Class of piping	Pipe outside diameter [mm]	Type of thread	
		Parallel thread	Tapered thread
I	≤ 33.7	No	Yes
II	≤ 33.7	No	Yes
III	≤ 60.3	Yes	Yes

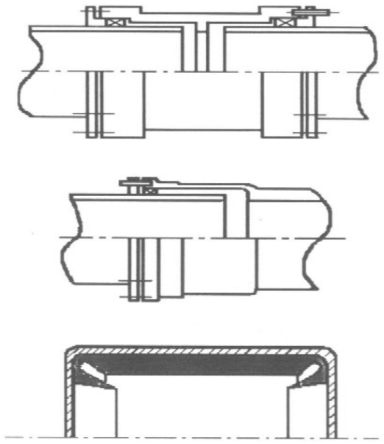
15.3.4 Mechanical Joints

15.3.4.1 Due to the great variations in design and configuration of mechanical joints, no specific recommendation regarding a calculation method for theoretical calculations is given in these requirements. The type approval is based on the results of testing of the actual joints. The type approval procedure is specified in *Publication No. 57/P – Type Approval of Mechanical Joints*.

15.3.4.2 The requirements specified in this sub-chapter apply to pipe unions, compression couplings, and skip-on joints as shown in Table 15.3.4.2. Similar joints complying with the requirements specified in this sub-chapter may be accepted by PRS.

Table 15.3.4.2
Examples of mechanical joints

PIPE UNIONS	
Welded or brazed types	
COMPRESSION COUPLINGS	
Swage type	
Press type	
Bite type	
Flared type	
SLIP-ON JOINTS	
Grip type	
Machine grooved type	

Slip type	

15.3.4.3 Mechanical joints shall be type approved by PRS for the intended application and service conditions.

15.3.4.4 Where the application of mechanical joints results in reduction of the pipe wall thickness due to the use of bite type rings or other structural elements, this shall be taken into account in determining the minimum wall thickness of the pipe (see sub-chapter 15.2).

15.3.4.5 Construction of mechanical joints shall prevent the possibility of tightness failure affected by pressure pulsation, piping vibration, temperature variation and other similar adverse effects occurring during the operation on board.

15.3.4.6 Material of mechanical joints shall be compatible with the piping material as well as internal and external media.

15.3.4.7 Mechanical joints in which tightness failure may occur in the event of damage shall not be used in piping sections directly connected to the hull shell or tanks containing flammable liquids.

15.3.4.8 Mechanical joints shall withstand external and internal pressure as well as vacuum, as applicable.

15.3.4.9 The number of mechanical joints shall be kept to a minimum. In general, flanged joints shall be used.

15.3.4.10 Piping in which a mechanical joint is fitted shall be adequately adjusted in accordance with the joint manufacturer's specifications. Supports or hangers shall not be used to force alignment at the point of connection.

15.3.4.11 Slip-on joints shall not be used in pipelines in cargo holds, tanks and other spaces which are not easily accessible, unless accepted by PRS. Application of these joints inside tanks may be permitted only for the same media that are in the tanks.

15.3.4.12 Unrestrained slip-on joints may be used only in the cases where compensation of lateral pipe deformation is necessary. Application of these joints as the main means of pipe connection is not permitted.

15.3.4.13 Application of mechanical joints and their acceptable use for each service, depending on the class of piping and outside d_Z, is indicated in Table 15.3.4.13.

Table 15.3.4.13
Application of mechanical joints depending upon piping class

Types of joints	Class of piping		
	I	**II**	**III**
PIPE UNIONS			
Welded and brazed type	Yes (for $d_Z \le 60.3$ mm)	Yes (for $d_Z \le 60.3$ mm)	Yes
COMPRESSION COUPLINGS			
Swage type	Yes	Yes	Yes
Press type	No	No	Yes
Bite type	Yes (for $d_Z \le 60.3$ mm)	Yes (for $d_Z \le 60.3$ mm)	Yes
Flared type	Yes (for $d_Z \le 60.3$ mm)	Yes (for $d_Z \le 60.3$ mm)	Yes
SLIP-ON JOINTS			
Grip type	No	Yes	Yes
Machine grooved type	Yes	Yes	Yes
Slip type	No	Yes	Yes

15.3.4.14 Acceptable applications of specific mechanical joints for particular piping systems are indicated in Table 15.3.4.14.

Table 15.3.4.14
Acceptable applications of mechanical joints

Systems		Kind of connections		
		Pipe unions	Compression couplings 5)	Slip-on joints
Flammable fluids with flash point > 60°C				
1	Fuel oil lines	Yes	Yes	Yes 2), 3)
2	Lubricating oil lines	Yes	Yes	Yes 2), 3)
3	Hydraulic oil	Yes	Yes	Yes 2), 3)
Sea water				
4	Bilge lines	Yes	Yes	Yes 1)
5	Fire main and water spray	Yes	Yes	Yes 3)
6	Foam system	Yes	Yes	Yes 3)
7	Sprinkler system	Yes	Yes	Yes 3)
8	Ballast system	Yes	Yes	Yes 1)
9	Cooling water system	Yes	Yes	Yes 1)
10	Non-essential systems	Yes	Yes	Yes
Fresh water				
11	Cooling water systems	Yes	Yes	Yes 1)
12	Sanitary and drinking water systems	Yes	Yes	Yes
Miscellaneous				
13	Deck drains (internal)	Yes	Yes	Yes 4)
14	Sanitary drains	Yes	Yes	Yes
15	Scuppers and discharge (overboard)	Yes	Yes	No
16	Sounding/vent pipes for water tanks and void spaces	Yes	Yes	Yes
17	Sounding/vent pipes for flammable liquid tanks with flash point > 60°C	Yes	Yes	Yes 2), 3)

18	Starting/control air 1)	Yes	Yes	No
19	Service air (other than those specified in 18)	Yes	Yes	Yes
20	CO_2 system 1)	Yes	Yes	No

Notes to Table 15.3.4.14:

1) Inside machinery spaces of category A – only approved fire resistant types.

2) Not inside machinery spaces of category A or accommodation spaces. May be accepted in other machinery spaces, provided the joints are located in easily visible and accessible positions.

3) Approved fire resistant types.

4) Above free board deck only.

5) If compression couplings include any components which readily deteriorate in case of fire, they shall be of approved fire resistant type as required for slip-on joints.

15.3.4.15 The installation of mechanical joints shall be in accordance with the manufacturer's assembly instructions. Where special tools and gauges are required for installation of the joints, these shall be supplied by the manufacturer.

15.4 Tube Bend Radii

The mean radius of bend of the steel and copper pipes subjected to a pressure exceeding 0.5 MPa or to a temperature of the internal medium exceeding 60 °C, as well as the radius of bend of the pipes intended for self-expansion shall not be less than 2.5*d*.

If, during the bending, no reduction of the pipe wall thickness occurs, then – subject to PRS acceptance of the bending process in each particular case – the specified radius may be reduced.

15.5 Protection against Overpressure

15.5.1 Where the pressure is likely to develop in excess of the working pressure, the piping shall be provided with means preventing the pressure in the pipeline to rise above the working pressure.

Open escape of oil fuel, lubricating oil and other flammable oil from the safety valves is not permitted.

15.5.2 Where provision is made for a reducing valve on the pipeline, a pressure gauge and safety valve shall be installed thereafter. An arrangement for bypassing the reducing valves is recommended.

15.6 Corrosion Protection

15.6.1 Upon completion of bending and welding steel pipes of bilge, ballast and sea-water systems, air, sounding and overflow pipes of water tanks and ballast/fuel tanks, gas freeing and vent pipes of cargo tanks and cofferdams in tankers shall be protected against corrosion by a method accepted by PRS in each particular case.

15.6.2 Where bottom and side fittings or their parts are made of copper alloys, provision shall be made for protection of the shell plating and all other elements being in contact with the said fittings against electrolytic corrosion.

15.6.3 Where galvanized sea-water pipes are connected to copper alloy casings of pumps, units, heat exchangers and elements of fittings, provision shall be made for protection against electrolytic corrosion.

15.7 Insulation of Pipes

15.7.1 Insulation of pipes shall fulfil the requirements specified in paragraph 2.1.7.

15.7.2 Insulation of refrigerating pipes shall be protected against absorption of moisture. At bulkhead and deck penetrations the pipes shall not be in direct contact with these divisions, to avoid the formation of heat leakage bridges.

15.7.3 Antiperspiration materials and glues applied with insulation as well as insulation of fittings need not fulfil the requirements specified in paragraph 15.7.1, provided these materials are used in as small quantity as possible, and their uncovered surfaces have the characteristics of LFS material (see definitions in sub-chapter 1.2 of *Part V – Fire Protection*).

15.8 Valves and Fittings

15.8.1 Covers of valves with internal diameter of more than 32 mm, equipped with turning spindles, shall be secured to the bodies by bolts or studs.

Screwed-on covers of valves shall be effectively secured against loosening.

The nut of cock plug shall be secured against unscrewing from the taper.

15.8.2 Remote controlled valves, operating with auxiliary source of power shall have local manual control, the operation of which shall be independent of the remote control. Manual control of the valves shall not render any failure in the remote control system.

Construction of the remote controlled valves shall be such as to ensure that in the case of failure of remote control system the valves remain in a position that will not render any state of emergency to the ship or they automatically set to such position.

15.8.3 Shut-off devices shall be fitted with nameplates clearly specifying their purpose.

15.8.4 For remote controlled valves, nameplates specifying their purpose, as well as the indications (valve open/valve closed), shall be provided in the control stations. Where the remote control is intended for closing the valves only, such indicators need not be provided.

15.8.5 Valves and fittings installed on watertight bulkheads shall be secured by studs screwed into pads fitted to the bulkhead, or they may be attached to bulkhead penetration pieces.

The stud holes shall not be through holes.

15.8.6 Valve chests and manually controlled valves shall be situated in positions always accessible during the normal operation of the vessel.

15.9 Bottom and Side Sea Chests, Bottom and Side Valves and Fittings and Openings in Hull Shell

15.9.1 Bottom and side valves and fittings as well as fittings installed on bottom and side sea chests shall be made of steel, cast steel or bronze.

No components of the valves and fittings of the side sea chests installed below the deck or of the bottom valves and fittings shall be made of materials which may readily deteriorate in the event of fire.

15.9.2 Spindles and closing parts of the bottom and side valves and fittings shall be made of materials resistant to the corrosive effect of sea-water.

15.9.3 Bottom and side inlet valves and fittings shall be installed directly on the bottom or side sea chests or on welded pads.

15.9.4 It is recommended that discharge lines having open inlet openings which penetrate the side shell plating be made of pipes and provided with fittings in accordance with the requirements specified in Table 15.9.4**.**

As a general rule it shall be adopted that the lines having open inlets in enclosed spaces and led through the side shell plating below the deepest waterline shall be fitted with shut-off non-return valves installed directly on the shell plating. If such valves are not installed directly on the shall plating, the line between the plating and valve shall be made of pipe with an increased wall thickness.

15.9.5 Side inlets of piping systems serving the main and auxiliary machinery, located in machinery spaces shall be fitted with readily accessible valves or gate valves with a local control. The valve controls shall be fitted with indicators (valve open/valve closed).

Side discharge valves shall be of a spring-loaded non-return type. Lower edge of the discharge opening shall be situated above the deepest waterline as high as practicable.

If the lower edge of the opening is situated higher than 200 mm above the deepest waterline, then the side valve may be of a non-spring-loaded non-return type.

If the lower edge of the opening is situated higher than 300 mm above the deepest waterline and on the line inside the ship a loop having a lower edge above the deck and at least 500 mm above the deepest waterline is arranged, then the side valve may be of a shut-off type.

15.9.6 The means for operating the bottom sea inlet valves shall be situated in readily accessible positions and fitted with indicators (valve open/valve closed). It is recommended that these means be located above the floor plating of the machinery space.

15.9.7 The bottom and side valves and fittings shall be installed on welded pads.

The valves and fittings may be installed on the welded distance pieces, provided the latter are of rigid construction and of a minimum length.

The wall thickness of a distance piece shall not be less than the minimum thickness of the shell plating at the ship ends.

The holes for the fastening bolts or studs shall not be of through type and shall end in welded pads. Gaskets shall not be made of materials which readily deteriorate in the event of fire.

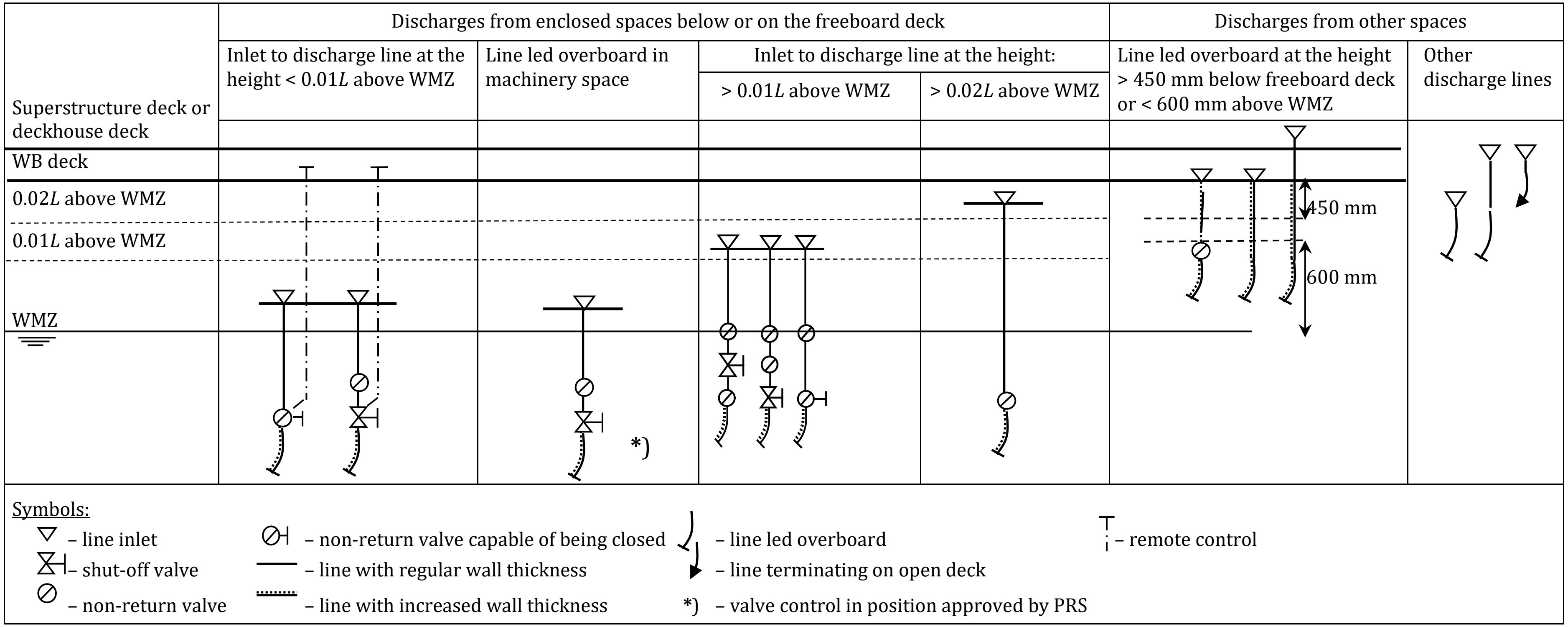
15.9.8 Bottom and side valves and fittings of ships with wooden or plastic hull may have flanges fastened with bolts to galvanised or bronze flanges which form internal hull strengthening of the shell. The holes for the bolts fastening the flanges of valves and fittings to the strengthening flanges shall not be through holes but they shall terminate in such a flange.

15.9.9 The number of through holes in the hull shell for the fastening bolts shall be kept to a minimum as necessary. For this purpose, pipelines of similar service shall be connected to the common discharges as practicable.

15.9.10 Arrangement of inlets and discharges in the hull shell shall preclude the possibility for the drainage or other wastes being sucked by sea-water pumps.

15.9.11 Openings in the ship shell plating for the bottom and side sea chests shall be fitted with protective gratings; alternatively, holes or slots may be made in the ship's hull. The total area of the holes or slots shall not be less than 2.5 times the total cross-sectional area of the installed sea-water inlet valves. The diameter of holes or width of slots in the gratings or shell plating shall be about 20 mm.

Table 15.9.4
Recommendations for discharge lines



Notes to Table 15.9.4

1. WMZ – deepest waterline.
2. WB – freeboard deck.
3. *L* – length of ship as defined in paragraph 1.2.3.2 of *Part II – Hull*.
4. Line with regular wall thickness has wall thickness not less than that specified in column 4 of Table 15.2.1-1, whereas line with increased wall thickness has wall thickness not less than that specified in column 6 of the same table.

Where compressed air is used to blow through the sea chests, screw-down non-return valves shall be fitted on the compressed air pipes and the pressure shall not exceed 0.5 MPa.

15.9.12 Where it does not result from the requirements specified in other paragraphs, screw-down non-return valves shall be installed on each line before and after each pump having a connection to the bottom and side valves.

15.10 Piping Arrangement

15.10.1 Piping Arrangement in Watertight Constructions and Fire Divisions

15.10.1.1 The number of pipes led through watertight bulkheads shall be kept to a minimum.

15.10.1.2 Only one pipeline is permitted to be led below the upper deck through the collision bulkhead for handling the liquid contained in the forepeak. Where the forepeak is divided by a longitudinal bulkhead into two watertight compartments, one suction branch pipe serving each compartment may be provided.

On each pipeline mentioned above, a shut-off valve shall be provided being installed on the welded pad on the fore side of the bulkhead. Such valves shall be controllable from a ready accessible position on the deck. Such valves need not be installed unless the pipeline passes through tanks and accommodation spaces.

15.10.1.3 Where the pipes pierce watertight bulkheads, decks or other watertight structures, provision shall be made for penetration pieces, bulkhead flanges or other arrangements ensuring watertight integrity of the structure concerned.

Holes for bolts and studs shall not pierce the watertight structures, but shall terminate in the pads.

Gaskets shall not be made of materials which readily deteriorate in case of fire.

Penetration pieces attached by welding to watertight decks and bulkheads shall be thicker by 1.5 to 3 mm than the wall thickness of a pipe to be connected, depending on its diameter.

15.10.1.4 Penetration of pipes through fire-resisting divisions shall be so made that fire resistance of the division does not deteriorate.

15.10.2 Piping Arrangement in Tanks

15.10.2.1 Drinking water pipes may be led through oil tanks and oil pipes may be led through drinking water tanks only in tight tunnels, forming an integral part of the tank structure.

Sea-water pipes, oil pipe, vent pipes, overflow pipes and sounding pipes need not be led inside tunnels provided that within the tanks pipes are seamless with wall thickness not less than 3 mm and connected by means of permanent joints. Where the use of detachable joints is indispensable, they shall be of a flange type with oil resistant gaskets.

15.10.2.2 Where no tunnels are used in leading the pipes through tanks, strain compensation shall be ensured by means of bends within the tanks.

Where pipes are led in tunnels, it is recommended that the compensation bends be situated outside the tunnels.

15.10.3 Piping Arrangement in Cargo Holds and Other Spaces

15.10.3.1 Means used to secure pipes shall not cause stresses therein due to thermal expansion, deformation of ship structure or vibration.

15.10.3.2 Pipes passing through cargo holds, chain lockers and other spaces where they are liable to mechanical damage shall be protected effectively.

15.10.3.3 Oil fuel pipes, hydraulic pipes and water pipes, except bilge pipes, shall not be led through cargo holds.

In special cases, which are subject to PRS consideration in each particular case, such pipes may be led through cargo holds, however only in tunnels or using pipes with increased wall thickness and protecting them with robust steel shielding.

15.10.3.4 In no case shall considerably heated pipes penetrate divisions made of flammable materials.

15.10.3.5 Oil fuel pipes shall not be led through accommodation and service spaces except bunkering pipes led through sanitary spaces provided that the wall thickness is not less than 5 mm and the pipes have no detachable joints within those spaces.

15.10.3.6 Pipes intended to convey hot media as well as long pipes led along the ship shall be fitted with expansion pieces or sufficient number of bends securing free compensation of strain.

15.10.4 Piping Arrangement near Electrical Appliances

15.10.4.1 In no case shall pipes subjected to pressure be led above and behind the main or emergency switchboards, or the control panels of important arrangements and machinery.

In front of and alongside the switchboards and control panels such pipes may be led at a distance of at least 500 mm provided that the pipes have no detachable joints or special shielding is provided.

15.10.4.2 Pipes shall not be led through the space where the main gyrocompass is installed, with the exception of pipes used for cooling it.

16 BILGE SYSTEM

16.1 Pumps

16.1.1 Self-propelled ships fitted with the main or auxiliary engine shall be provided with at least two power bilge pumps.

One of the bilge pumps may be a main engine-driven pump.

Independent drive water pumps of adequate capacity may be used as bilge pumps. Where it is required by *Part V – Fire Protection* to provide a ship with a separate fire pump, such a pump may be used in the bilge system only for emergency drainage of the machinery space.

16.1.2 In self-propelled ships without auxiliary engines, one of the pumps may be driven by the main engine, and the other may be a hand pump. In that case, the capacity of the main engine-driven pump shall not be less than the capacity of the sea water pump intended for cooling such an engine, whereas the capacity of the hand pump – not less than 7.5 m^3/h.

16.1.3 Bilge pumps shall be of self-priming type. It is recommended that one of the bilge pumps be of piston type.

16.1.4 Capacity Q of each bilge pump required in paragraphs 16.1.1 and 16.1.2 shall not be less than that determined in accordance with the formula below:

$$Q = \frac{5.65}{1000} D^2 \ \ [\mathrm{m^3/h}] \qquad (16.1.4)$$

where:

D – internal diameter of the bilge main, determined in accordance with formula 16.2.1, [mm]

Two pumps of combined capacity not less than that calculated from the above-mentioned formula may replace one of the bilge pumps.

16.1.5 In ships having a length of up to 12 m, one bilge hand pump or piston pump or membrane pump having the capacity not less than 4.5 m^3/h may be provided for drainage of each watertight compartment outside the machinery space.

Suction lines of such pumps shall be made of steel or copper. Strum boxes need not be installed for membrane pumps which are capable of being quickly open and closed without any tools.

16.1.6 For bilge drainage in non-propelled ships not fitted with power-driven auxiliary machinery at least two hand piston pumps of a combined capacity not less than 4.5 m^3/h shall be installed.

The pumps shall be situated above the bulkhead deck and shall have a sufficient suction head. In non-propelled ships having a power source, it is recommended that power pumps be installed in the number and with the capacity in accordance with the requirements for hand pumps.

16.1.7 Bilge pumps may be used to serve the ballast pipelines and to deck washing.

16.2 Pipe Diameters

16.2.1 Internal diameter D of the bilge main and of the branch suctions connected directly to the pump shall not be less than that determined in accordance with the formula below:

$$D = 1.68\sqrt{L(B+H)} + 25 \ \ [\mathrm{mm}] \qquad (16.2.1)$$

where:
L – length of ship, [m];
B – breadth of ship, [m];
H – moulded depth, [m];
and not less than the value obtained in accordance with the requirements specified in paragraph 16.2.3.

16.2.2 Internal diameter d of the branch suctions connected to the bilge main and the diameter of suction pipes of hand pumps shall not be less than that determined in accordance with the formula below:

$$d = 2.15\sqrt{l(B+H)} + 25 \quad [\mathrm{mm}] \qquad (16.2.2)$$

where:

l – length of the compartment to be drained, measured over its bottom, [m];
B, H – see paragraph 16.2.1.

16.2.3 The internal diameter of the bilge main and branch suctions shall not be less than 50 mm. In no case the internal diameter of the bilge main and of the branch suctions connected directly to the pump shall be less than that of the suction branch of the pump.

16.2.4 Cross-sectional area of the pipe connecting the distribution chest with the bilge main shall not be less than the total cross-sectional area of the two largest branch bilge suctions connected to this chest, however not greater than the cross-sectional area of the bilge main**.**

16.3 Arrangement and Joints of Pipes

16.3.1 Arrangement of bilge pipes and branch suctions shall be such as to enable any watertight compartment to be drained by any of the pumps required in paragraph 16.1.1. This requirement does not apply to fore- and afterpeaks as well as tanks intended for the carriage of liquids.

Each space or group of spaces which are not drained by means of the bilge system pipes shall be provided with other means to remove water.

16.3.2 Arrangement of bilge pipes shall preclude the possibility of passage of sea-water into the ship or passage of water from one watertight compartment into another. For this purpose the suction valves of bilge piping distribution chests as well as the valves on branch suctions connected directly to the bilge main shall be of shut-off non-return type. Check valves that are not spring loaded shall not be applied.

Where a bilge pump is intended to be also used for pumping water over the ship's side, the suction lines shall be separated by three-way valves with “L” plugs or an additional shut-off non-return valve shall be installed on the suction branch between the pump and the bilge main. Other equivalent solutions are also permitted.

16.3.3 Arrangement of bilge suctions and location of the valves and fittings shall be such that the flow resistance be as low as possible and – if practicable – that self-priming of the pumps be ensured (e.g. by creating an effective loop water seal).

16.3.4 Arrangement of bilge pipes shall be such as to enable draining the engine room through the suctions connected directly to the pump, with other compartments being simultaneously drained by other pumps.

16.3.5 Arrangement of bilge pipes shall be such as to enable one of the pumps to be operated while the remaining pumps are being used for other services.

16.3.6 Arrangement of bilge pipes shall be such as to enable the compartment drainage while the water fire extinguishing system is being used in this compartment.

16.3.7 In general, bilge pipes shall be led outside the double bottom space. Where it is necessary to lead bilge pipes through oil or drinking water tanks, the pipes shall fulfil the requirements specified in paragraph 15.10.2.1.

Where pipes are led through the double bottom tanks, open ends of suction pipes shall be fitted with non-return valves.

16.4 Drainage of Machinery Spaces

16.4.1 Machinery spaces whose bottom rise towards the sides is 5° or more, shall have at least two suction branches situated close to the ship’s plane of symmetry and so connected directly to the pumps required in paragraphs 16.1.1 and 16.1.2 that to provide the possibility for each bilge pump suction from each suction branch or the possibility for any bilge pump suction from both suction branches.

Where the bottom rise is less than 5°, additional bilge suctions connected to the bilge main shall be provided at each side.

16.4.2 Where tanks extend under the machinery space floor from side to side of the ship or are arranged otherwise, the arrangement of branch suctions is subject to PRS consideration in each particular case.

16.4.3 Where the machinery space whose bottom has a rise less than 5° is located abaft, the suction branches mentioned in paragraph 16.4.1 shall be positioned at the sides in the forward part of the machinery space, the possibility shall be provided for each bilge pump suction from each suction branch or the possibility for any bilge pump simultaneous suction from both suction branches. Additionally, one or two suction branches (which is subject to PRS consideration in each particular case), depending on the shape of the after part of the machinery space, shall be provided in the after part of the machinery space.

16.4.4 In suction branches for the machinery space drainage, easily accessible settling tanks shall be provided. The pipes connecting settling tanks with bilges shall be as straight as practicable. Lower ends of these pipes shall not be fitted with strum boxes.

16.4.5 Settling tanks shall be provided with readily openable covers. In ships of the length not exceeding 17 m, meshes may be used instead of settling tanks to separate impurities if access is provided for their cleaning.

16.4.6 The number and arrangement of the suction branches in other machinery spaces shall be the same as in cargo holds (see sub-chapter 16.5).

16.5 Drainage of Cargo Holds

16.5.1 Cargo holds whose bottom rise towards the sides is 5° or more, shall be provided with at least one suction branch situated close to the ship's plane of symmetry.

Where the bottom rise is less than 5°, additional bilge suctions connected to the bilge main shall be provided at each side.

16.5.2 At the narrow ends of cargo holds one bilge suction may be fitted.

16.5.3 Where tight wooden panels or removable covers are provided over bilges in cargo holds, provision shall be made to enable free draining the water accumulated in the hold into bilges.

16.5.4 Where an access manhole to the bilge well shall be provided, it shall be arranged as close to the suction strum box as practicable.

16.5.5 Where tanks extend under the cargo hold floor from side to side of the ship or are arranged otherwise, the arrangement of branch suctions is subject to PRS consideration in each particular case.

16.5.6 Branch suctions from cargo holds and other compartments shall be fitted with strum boxes or strainers with perforations from 8 to 10 mm in diameter. The combined area of such perforations shall not be less than twice the area of the relevant suction pipe.

Strum boxes shall be so constructed that they can be cleared without dismantling any joint on the suction branch.

16.6 Drainage of Fore-and Afterpeaks

The peaks which are not used as tanks may be drained by means of separate hand pumps water ejectors or directly drained by means of pump through a self-closing valve.

16.7 Drainage of Other Spaces

16.7.1 Chain locker and boatswain's store may be drained by means of hand pumps, water ejectors or other arrangements subject to PRS consent in each particular case.

16.7.2 Drainage of steering gear rooms and other small compartments situated above the afterpeak may be carried out by means of hand pumps or water ejectors or by means of drain pipes led into the shaft tunnel or machinery space bilges. The drain pipes shall be fitted with self-closing cocks located in readily accessible places. The internal diameter of the drain pipes shall not be less than 39 mm.

17 AIR, OVERFLOW AND SOUNDING PIPES

17.1 Air Pipes

17.1.1 Each ship's tank intended for the storage of liquid, each cofferdam, as well as the side and bottom sea chests and boxes of shell coolers, shall be fitted with air pipes.

Air pipes of side and bottom sea chests and boxes of shell coolers shall have not less than 30 mm in diameter and be fitted with shut-off valves installed directly on those chests and boxes (see also sub-chapter 15.9).

Air pipes of tanks adjacent to the shell plating, side and bottom sea chests and boxes of shell coolers shall terminate above the open deck.

17.1.2 Tank air pipes shall be led from the upper part of the tank from a place situated at the maximum distance from the filing pipe. The number and arrangement of the air pipes shall be such as to preclude the formation of air pockets.

17.1.3 Tanks extending from ship's side to side shall be fitted with air pipes at both sides. Air pipes shall not be used as filling pipes unless the tank is fitted with more than one air pipe.

Air pipes of tanks intended for different kinds of liquids shall not be interconnected.

17.1.4 The height of air pipes measured from the open deck to the uppermost level of the liquid in the filled up air pipe shall be at least:
– 600 mm –on the freeboard deck,
– 380 mm –on the first tier superstructure deck superstructure deck.

17.1.5 Open ends of air pipes situated on the freeboard open decks and on the first tier superstructure decks (see definitions in subchapter 1.2 of *Part III – Hull Equipment*), as well as those situated above these decks within the zone limited by the angle of downflooding (see definitions in subchapter 1.2 of *Part IV – Stability and Freeboard*) shall be fitted with fixed, self-acting closing appliances – the so called vent heads – of a type approved by PRS preventing the entry of sea-water into the tanks. This requirement does not apply to the compartments permanently filled with sea-water such as side and bottom sea chests and boxes of shell coolers. The ends of air pipes not fitted with the vent heads shall be made as an elbow, with its opening facing downwards or otherwise – subject to PRS consent in each particular case.

Requirements concerning construction and testing of air pipes closing appliances are specified in *Publication No. 33/P – Air Pipe Closing Devices*.

17.1.6 Open ends of air pipes of tanks containing oil fuel and other readily ignitable liquids shall be led to the places on the open deck where the escaping vapours will not cause any fire risk and shall be fitted with devices preventing flame passage of the construction accepted by PRS in each particular case. Free area of the cross-section of such devices shall not be less than the cross-sectional area of the air pipe.

17.1.7 Air pipes of lubricating oil storage tanks and not heated tanks of oil residues not forming an integral part of the ship's structure may terminate in the spaces where the tanks are located. It is necessary to ensure that no leaking oil may spread onto electrical equipment or heated surfaces in case of tank overfilling.

17.1.8 Total cross-sectional area of air pipes of the tanks filled by gravity shall not be less than the total cross-sectional area of all pipes by which the liquid may be simultaneously delivered into the tank.

17.1.9 Total cross-sectional area of air pipes of tanks filled by ship pumps or shore pumps shall be at least 1.25 times the cross-sectional area of the tank filling pipe. Cross-sectional area of an air pipe/manifold serving several tanks, if applicable, shall be at least 1.25 times the cross-sectional area of the common filling pipeline where these are independent filling pipes (or 1.25 times the combined cross-sectional area of the pipelines filling such tanks where these are independent filling pipes and simultaneous filling of such tanks is possible). In that case connections of the air pipes of individual tanks to such pipeline/manifold shall be located above the deepest waterline and for ships assigned a subdivision mark in the symbol of class – above the damage waterline.

17.1.10 Where a tanks filled by ship pumps or shore pumps is fitted with an overflow pipe, the combined cross-sectional area of the tank air pipes shall not be less than 1/3 of the filling pipeline cross-sectional area and in no case shall the diameter of an air pipe be less than 50 mm. Where air pipes serving several tanks are interconnected, the cross-sectional area of such a common air pipe shall not be less than 1/3 of the cross-sectional area of common pipeline filling such tanks; however the requirement specified in paragraph 17.2.3 shall also be fulfilled.

If the tanks filled only by gravity, the filling pipeline shall not have a diameter less than 38 mm for tanks of a capacity more than 200 *l* and not less than 12 mm for tanks of lesser capacity; however, in no case shall the air pipeline diameter be lesser than that of the filling pipeline.

Pipes passing through the refrigerated spaces shall not have an inside diameter less than 32 mm.

17.1.11 Arrangement of air pipes shall be such that under normal list and trim conditions no hydraulic seals may occur in the pipes.

17.1.12 Air pipes of oil fuel tanks shall have no detachable joints in way of accommodation and refrigerated spaces.

17.1.13 Nameplates shall be affixed to the upper ends of air pipes.

17.1.14 Air pipes of internal combustion engine crankcases shall fulfil the requirements specified in paragraph 3.2.1.

17.2 Overflow Pipes

17.2.1 Oil fuel tanks shall be fitted with overflow pipes. Overflow pipes need not be provided if the fuel system has been so arranged that the possibility for the liquid overflowing overboard during bunkering or transferring the oil fuel is precluded**.**

17.2.2 Cross-sectional area of the overflow pipes shall fulfil the requirements for air pipes specified in paragraph 17.1.9.

17.2.3 Where overflow pipes from several tanks forming integral part of the hull structure and situated in different watertight compartments are led to a common line (manifold), the connections and the common line itself shall be situated above the deepest damage load waterline in ships having a mark of subdivision in their symbol of class and above the deepest load waterline in other ships.

17.2.4 Air pipes being also overflow pipes shall not be connected to the air pipe of the overflow tank, but directly to that tank or to other overflow pipe, of a sufficient diameter, connected to that tank**.**

17.2.5 Overflow pipes of oil fuel daily service and settling tanks shall be led to the tanks situated below the above-mentioned tanks.

17.2.6 A heat-proof sight glass or an alarm giving warning of oil fuel overflow shall be fitted on the overflow tank or a vertical segment of the overflow pipe**.**

17.3 Overflow Tanks

17.3.1 Capacity of oil fuel overflow tanks shall not be less than 10-minute capacity of the fuel transfer pump.

17.3.2 Overflow tank shall be provided with visual and audible alarms giving warning of the tank being filled above 75% of its capacity.

17.4 Sounding Pipes and Arrangements

17.4.1 Tanks and cofferdams, as well as bilges and bilge wells which are not readily accessible, shall be fitted with sounding pipes led to the open deck or other level indicating devices approved by PRS. Sounding pipes shall be led straight or with slight curvature to permit easy passage of the sounding rod to readily accessible places.

Sounding pipes for cofferdams and tanks not forming part of the hull structure need not be led to the open deck.

17.4.2 Other level indicators, accepted by PRS, may be used for oil fuel instead of sounding pipes. The indicators shall fulfil the following requirements:

- **.1** failure of the indicator or overfilling of the tank shall not cause any leakage of oil fuel;
- **.2** level indicators with a transparent insert are permitted; this insert shall be made of unbreakable fireproof flat glass or plastic not losing its transparency in contact with oil;
- **.3** between the level indicator and tank a self-closing cock shall be installed being situated at the bottom end of the indicator. Such a cock shall also be fitted at the top end of the indicator if the indicator has plastic inserts or if the indicator is connected to the tank below the maximum level of liquid.

In the case of oil tanks with a capacity less than 50 l, the self-closing cocks are not required.

17.4.3 Where the ship has a flat bottom and the bilge or tank extends from side to side of the ship, sounding pipes shall be installed at each side of the ship.

17.4.4 Sounding pipes for bottom oil tanks and bottom oil fuel tanks may terminate in machinery spaces or shaft tunnels, provided that:

.1 top ends of sounding pipes are led to the places distant from the places of high risk of ignition or are fitted with shielding effectively preventing oil fuel from accidental leakage onto heated surfaces of boilers, engines, exhaust pipes, etc. as well as onto electric machinery and switchboards;

.2 additionally, a self-closing test cock of a small diameter is fitted under the above-mentioned cock to enable the check that there is no oil fuel in the pipe prior to opening the sounding cock;

.3 sounding pipes shall not be used for tank filling or tank venting.

17.4.5 Sounding pipes of the double bottom water tanks may terminate in the spaces above the tanks if they are readily accessible. These pipes shall not be used as air pipes and shall be fitted with self-closing cocks.

17.4.6 Striking plate or an equivalent arrangement protecting the bottom plating against damage shall be fitted under each open ended sounding pipe, e.g. sounding rod foot or pipe closer.

Where slotted sounding pipes having closed ends are employed the lower ends of the pipes shall be adequately strengthened.

17.4.7 Internal diameter of sounding pipes shall not be less than 32 mm.

Internal diameter of sounding pipes led through refrigerated spaces where the temperature may drop to 0°C or below, as well as of the pipes of tanks fitted with heating installation, shall not be less than 50 mm. Within refrigerated spaces the pipes shall be insulated.

17.4.8 Nameplates shall be affixed to the upper ends of sounding pipes.

17.4.9 Plugs and threaded parts of the sounding pipe deck sockets fitted in open decks shall be made of bronze, brass or stainless steel. Application of other materials is subject to PRS consideration in each particular case.

17.4.10 Self-closing fittings of sounding pipes of the double bottom oil fuel shall be resistant to corrosion tanks and so manufactured as to preclude sparking.

18 EXHAUST GAS SYSTEM

18.1 Exhaust Gas Lines

18.1.1 Exhaust gas lines shall terminate on the open deck and they shall not be led through the accommodation spaces.

18.1.2 Where exhaust gas lines are led through the shell plating, near or below the load waterline, means shall be provided or the line shall be properly shaped to prevent overboard water ingress into the engine. Inside the ship, the exhaust gas line shall form a sort of loop whose lower edge is situated as high above the deepest waterline as practicable. The wall thickness of the pipeline segment between the loop lower edge and shell plating shall not be less than the shell plating thickness where the exhaust line terminates.

18.1.3 In supply ships intended to serve tankers carrying cargoes with a flash point below 60 °C and to serve ships carrying flammable or dangerous goods all exhaust gas lines shall be fitted with the devices for smothering sparks – so called spark arresters of a design agreed with PRS.

Alternatively the pipes may be led overboard through the shell plating (provided that it is permitted by the manufacturer of the equipment from which the exhaust gas is being extracted), on the condition that the outlet is arranged at least 0.3 m below the light waterline.

18.1.4 Exhaust gas lines and oil fuel tanks shall not be less than 450 mm apart (measured from the exhaust gas line insulation. The temperature of the insulation surface shall not exceed 60°C.

18.1.5 Each main I.C. engine shall be fitted with an individual exhaust line and silencer. In justified cases, PRS may grant exemptions from the requirement.

18.1.6 Exhaust gas pipes of auxiliary engines may be connected to a common exhaust line, provided reliable measures are taken to prevent:
– exhaust gas from the common exhaust line from entering the engines which are not in operation,
– damage to any of the engines when starting.

18.1.7 Exhaust lines shall be so secured as to provide for their thermal deformation eliminating the stress transfer on the hull structure.

18.1.8 Exhaust gas system shall constitute no fire risk. If the exhaust lines are led through the spaces where crates containing fish, fishing nets etc. are stored, the lines shall shielded to provide a minimum distance of 10 cm for free air circulation around the pipe.

18.1.9 Where exhaust gas boilers and wet type spark arresters are installed, measures shall be taken to prevent water from entering the engine in the case of boiler pipes leakage or due to any other damage. The drain pipes shall be led to the engine room bilges and fitted with hydraulic seals.

18.1.10 The requirements for exhaust lines of heating and cooking appliances firing oil fuel, solid fuel or liquefied gas are specified in Chapter 12.

19 VENTILATION SYSTEM

19.1 Ventilation Ducts

19.1.1 Ventilation ducts shall not be led through the watertight bulkheads below the bulkhead deck.

19.1.2 Trunks and vertical ventilation ducts led through the watertight decks within one watertight compartment, shall be watertight and shall have the strength equivalent to the strength of the hull local structures between those watertight decks.

19.1.3 Ventilation ducts shall be protected against corrosion or made of corrosion resistant material.

19.1.4 Ventilation ducts leading to cargo spaces, machinery spaces or other spaces protected against fire by smothering system shall be provided with closing arrangements operable from the deck.

19.2 Arrangement of Ventilator Heads

Supply ventilator heads shall be located in such places on open decks where the possibility of drawing in the air contaminated with oil vapours and the possibility of the entry of outboard water into ventilation ducts are reduced to a minimum.

Height of ventilator head coamings shall fulfil the requirements specified in *Part II – Hull.*

19.3 Ventilation of Machinery Spaces

Ventilation system of machinery spaces shall ensure sufficient influx of air necessary for the operation of engines and oil-fired boilers of heating appliances at all weather conditions. Provision shall be made for the extraction of gases heavier than air from the lower parts of those spaces, from the places below the floor plates where the gases may accumulate. The system shall consist of at least two ducts so arranged that the change of air is as efficient as practicable. At least one of these ducts shall terminate under the floor.

19.4 Ventilation of Battery Rooms and Lockers

19.4.1 Ventilation system (both inlet and outlet) of battery rooms and battery lockers shall be independent and shall ensure the removal of air from the upper parts of the ventilated rooms and lockers.

Ventilation ducts (both inlet and outlet) shall be gastight.

19.4.2 The supplied air shall be led to the lower zone of the battery rooms and lockers.

19.4.3 Outlets to the open of ventilation ducts shall be so arranged as to prevent water, precipitation and solids from entering into the ducts. Flame arresters shall not be fitted on these ducts. The outlets of the exhaust ventilation ducts shall be situated in such positions where the discharged gases will not cause any fire risk.

19.4.4 Ventilation of accumulator battery lockers containing batteries with charging capacity not exceeding 0.2 kW, may be effected through holes in the lower and upper parts of the locker. Battery charging current shall be determined in accordance with the guidelines specified in Chapter 10 of *Part VII – Electrical Installations and Control Systems*.

19.4.5 The rate of air flow Q for ventilation of an accumulator battery room or accumulator battery locker shall not be less than that determined in accordance with the formula below:

$$Q = 0.11In \quad [\mathrm{m^3/h}] \tag{19.4.5}$$

where:

I – charging current during gas evolution, however not less than 0.25 times the maximum charging current, [A];
n – number of battery cells.

19.4.6 Cross-sectional area F of a vent duct in the case of natural ventilation of accumulator battery rooms or accumulator battery lockers shall not be less than that determined in accordance with the formula below:

$$F = 2.9Q \quad [\mathrm{cm^2}] \tag{19.4.6}$$

however not less than 80 cm^2 for lead-acid batteries and 120 cm^2 for alkaline batteries, where:

Q – air flow rate, [m^3/h], determined in accordance with formula 19.4.5.

19.4.7 Natural ventilation of the rooms may be applied provided that:

.1 required amount of air, calculated in accordance with formula 19.4.5 is less than 85 m^3/h;
.2 ventilation duct rake from vertical is less than 45^o;
.3 number of duct bends does not exceed 2;
.4 length of the ventilation duct does not exceed 5 m;
.5 ventilation performance does not depend on the wind direction;
.6 cross-sectional area of the ventilation duct fulfils the requirements specified in paragraph 19.4.6.

19.4.8 Where the air flow rate determined in accordance with formula 19.4.5 is 85 m^3/h or more, the accumulator battery room shall be provided with mechanical exhaust ventilation.

19.4.9 Internal surfaces of the exhaust ducts, as well as the fans and their motors shall be effectively protected against the action of electrolyte vapours.

19.4.10 Fan motors shall not be positioned in the discharged air flux.

Fans shall be so constructed that the possibility of sparking be reduced to a minimum.

19.4.11 Air gap between the casing and rotor shall not be less than 0.1 of the rotor shaft bearing journal diameter and not less than 2 mm, however it is not required for the air gap to be greater than 13 mm.

19.4.12 Inlet openings of ventilation ducts shall be protected from entering of foreign matters into the fan casings by means of wire net, with square net mesh of the side length not exceeding 13 mm.

19.5 Ventilation of Fire-extinguishing Stations

19.5.1 Fire-extinguishing stations of foam and carbon dioxide systems shall be provided with effective ventilation.

19.5.2 Fire-extinguishing stations of carbon dioxide systems shall be fitted with natural exhaust ventilation from the lower parts of the compartment, and with supply ventilation in the upper parts of the compartment.

Where the fire-extinguishing station of carbon dioxide system is situated below the open deck, exhaust ventilation system shall be mechanical and shall ensure at least 6 air changes per hour and the fans shall be started automatically as the door of the station is opened. During operation of the fan, a light signal, visible after the door is opened, shall be provided.

19.5.3 Ventilation system of the high-expansion foam fire-extinguishing system station shall ensure free inflow of air sufficient for the foam generators to operate properly.

20 OIL FUEL SYSTEM

20.1 Pumps

20.1.1 At least two power-driven pumps shall be provided for fuel transfer. In vessels with a daily fuel consumption not exceeding 2 tonnes, one hand operated pump may be accepted.

20.1.2 In addition to local control, oil fuel transfer pumps and oil separator pumps shall be capable of being stopped from a position outside the space in which they are situated.

20.2 Piping, Valves and Fittings

20.2.1 Oil fuel piping shall be separated from any other systems.

20.2.2 Piping conveying oil fuel under pressure shall be located in an easily visible and accessible positions.

20.2.3 Oil fuel pipes shall not be led above internal combustion engines, heating appliances, exhaust gas pipes or other hot surfaces.

In exceptional cases, oil fuel pipes may be lead above this piping and equipment, provided that no detachable joints have been installed on pipes in those locations and the pipes have been provided with drip trays preventing the oil fuel from coming into contact with this machinery and equipment.

20.2.4 Oil fuel suctions from service tanks and other tanks of more than 50 litres in capacity shall be provided with a remote-controlled shut-off valve fitted directly on the tank. Control of the valve shall be effected from a safe position located outside the space where the tank is situated.

20.2.5 Valves and fittings of fuel systems shall always be readily accessible.

20.3 Water Draining Arrangements for Tanks

Settling and daily service tanks shall be provided with self-closing valves and drain pipes led to the drain tank. The drain pipes shall be fitted with sight glasses. Where a drip tray has been provided, an open funnel, instead of the sight glass, may be fitted.

20.4 Oil Fuel Leakage Collecting Arrangements

20.4.1 Drip trays shall be fitted where oil fuel leakage from tanks which do not form an integral part of the hull structure, pumps, filters and other equipment may be expected.

20.4.2 Drain pipes from the drip trays shall be led to drain tanks. The pipes shall not be led to bilges and overflow tanks.

20.4.3 The internal diameter of the drain pipes shall not be less than 25 mm.

20.4.4 Drain pipes shall be led to the drain tank bottom as close as practicable. Where the drain tank is situated in the double bottom, structural measures shall be taken to prevent penetration of water into the engine room through the open ends of the drain pipes in event of damage to the shell plating. It is recommended that non-return valves reacting to a small pressure difference be used.

Provision shall be made for signals warning of the maximum permissible level being achieved in a drain tank.

20.5 Bunkering

20.5.1 Bunkering of oil fuel shall be effected by means of a permanent pipeline provided with necessary valves and fittings enabling all storage tanks to be filled with oil fuel.

The filling pipes shall be led to the tank bottom as close as practicable.

20.5.2 Filling pipes of the tanks shall be led through the tank wall in its upper part. Where such an arrangement is impracticable, the filling pipes shall be fitted with non-return valves installed directly on the tanks.

Where the filling pipe is also used as a suction pipe, the non-return valve shall be replaced with a valve remotely closed from a readily accessible position outside the space in which the tank is located.

20.6 Oil Fuel Tanks

20.6.1 Construction of oil fuel tanks which do not form an integral part of the hull structure shall withstand the test pressure of the water head of 2.5 m above the upper edge of tank (0.025 MPa). If necessary, appropriate wash divisions shall be applied.

If the tank is filled by pump through a connection, the test pressure shall correspond to an increased by 0.025 MPa pressure which may be caused by the oil fuel overflowing through an air pipe or overflow pipe.

Oil fuel tanks made of carbon steel, stainless steel, aluminium alloys which do not form an integral part of the ship's structure as well as integral tanks made of glass-reinforced plastic shall have wall thickness not less than the values specified in Table 20.6.1, unless the strength criteria require greater values. The material used shall have the manufacturer's certificate.

Table 20.6.1
Minimum wall thickness for freestanding oil fuel tanks

Oil fuel tank capacity [*l*]	Minimum wall thickness [mm] for a type of material			
	Carbon steel	Stainless steel	Sea-water resistant aluminium alloy	Glass-reinforced plastic (GRP)
0 ÷ 49	1.5	0.8	4.0	4.0
50 ÷ 99	3.0	0.8	4.0	4.0
100 ÷ 199	3.0	1.0	4.0	4.0
200 ÷ 999	5.0	3.0	6.0	5.0
1000 and more	5.0	4.0	7.0	5.0

Welded joints permissible in oil fuel tanks:
– butt weld,
– double side fillet joint with full penetration,
– single side fillet joint with full penetration,
– fillet joint for plate thickness not exceeding 3 mm.

The possibility for application of other welds is subject to PRS consent in each particular case.

Oil fuel tanks made of aluminium alloys or GRP shall not be fitted in machinery spaces or paint lockers. If such tanks are adjacent to the spaces containing heat emitting appliances installations, the tanks shall be separated by cofferdam whose bulkhead shall be insulated with rockwool of at least 15 mm in thickness, or by an equivalent material, with non-combustible oiltight coating.

20.6.2 Oil fuel tanks shall be separated from drinkable water tanks and lubricating oil tanks by cofferdams in accordance with the requirements specified in *Part II – Hull*.

20.6.3 Daily service tanks shall not be integral wing tanks or bottom integral tanks. They may, however, be formed by tight divisions as a part of the hull side tanks provided that they are not bounded by the shell plating.

20.6.4 Arrangement of oil fuel tanks in machinery spaces shall fulfil the requirements specified in paragraph 2.2.2.

20.6.5 In ships with wooden or glass reinforced plastic hull, oil fuel tanks shall not be adjacent to accommodation spaces. The air gap between the oil fuel tank and accommodation spaces shall be effectively ventilated.

In general, oil fuel tanks shall not be located in machinery spaces, and if this is the case, they shall be constructed of steel or another material equivalent in respect of fire integrity.

20.6.6 Freestanding oil fuel tanks shall be painted with a bright paint and located in such a distance from the shell plating that any possible leakage can be readily visible.

20.6.7 Oil fuel tanks situated on open decks, superstructure decks and in other places open to the atmosphere shall be protected from exposure to the sun rays.

20.6.8 Oil fuel tanks shall not be situated in the forepeak.

20.7 Oil Fuel Supply to Internal Combustion Engines

20.7.1 The equipment of oil fuel system shall provide the engine with oil fuel prepared and purified to such a degree as is required for the particular engine (see also 3.2.10).

It is recommended that oil fuel dehydrators be used.

20.7.2 In systems, where the main engine is supplied with oil fuel by a main engine-driven booster pump, provision shall be made to ensure the oil fuel delivery to the engines in the event of the booster pump failure. For this purpose, a daily service tank located at an appropriate height and an emergency connection bypassing the booster pump. Before the piston-type booster pump there shall be provided a coarse filter (gauze) appropriate for that pump.

20.7.3 Oil fuel shall be supplied from the daily service tank to the generating set engine through a separate pipeline.

20.8 Oil Fuel Supply to Cooking and Heating Appliances

The requirements concerning oil fuel supply to cooking and heating appliances are specified in sub-chapter 12.1.

21 LUBRICATING OIL SYSTEM

21.1 Pumps

Each main and auxiliary engine as well as their gear boxes and hydraulic coupling filling systems shall be provided with individual independent lubricating systems.

21.2 Lubricating Oil Supply to Internal Combustion Engines and Gears

21.2.1 Drain pipes from the engine crankcase to the circulating tank – their lower ends – shall be so arranged that they are permanently submerged in oil during the engine operation. The drain pipes of two or more engines shall not be interconnected.

21.2.2 The pipes of lubricating oil system shall not be connected to the pipes of other systems, except connections to the purifiers, which may be used for oil fuel purification, provided that reliable structural arrangements preventing oil fuel from being mixed with lubricating oil have been employed.

Where lubricating oil purifiers are employed, means preventing the main engine oil from being mixed with the auxiliary engines' lubricating oil shall be provided.

21.2.3 The pipes of the lubricating oil circulation systems shall be fitted with:
.1 magnetic strainer – on the suction pipe of lubricating pumps serving gears;
.2 one coarse filter (gauze) – on the pumps' suction pipe;
.3 generally two parallel filters, one interchangeable duplex filter or one self-cleaning filter – on the discharge pipe of the lubricating pump serving the main engine (see also 3.2.12).

21.2.4 In the case of remote starting of an engine or machinery requiring initial lubrication, it shall be activated automatically before the machinery has been brought into operation, and efficient lubrication shall condition the start-up of such machinery.

21.2.5 The throughput of each lubricating oil filter shall exceed by 10% the capacity of the largest pump.

21.3 Lubricating Oil Tanks

21.3.1 Lubricating oil tanks shall be separated from oil fuel, boiler feed water and drinking water tanks by cofferdams.

21.3.2 Circulation tanks shall in each case be separated from the bottom shell plating by a cofferdam.

21.3.3 Pipelines draining oil from tanks located outside the double bottom shall be provided with shut-off valves installed directly on those tanks.

21.3.4 Where an electric heater of the lubricating oil is provided, the requirements specified in sub-chapter 12.4 of *Part VII – Electrical Installations and Control Systems* shall be fulfilled.

21.3.5 For lubricating oil tanks located in the machinery spaces, the requirements specified in paragraphs 2.2.2, 20.4.1 and (in respect of their location) 20.6.1.

21.3.6 Where a complete drainage of the oil from the oil sump – for its periodical complete change – is expected to be performed through a hose, the shut-off valve installed on the engine (before the hose) shall be made of a suitable material and reliably protected against spontaneous opening.

22 COOLING WATER SYSTEM

22.1 Pumps

22.1.1 Each main propulsion liquid-cooled engine shall, depending on the cooling system employed, be provided with a main engine-driven pump of an open or closed cycle. Restricted service ships having one main engine, to be assigned mark I or II in the symbol of class, shall be fitted with the main engine cooling system so arranged that in case of the main engine-driven service pump failure another suitable pump of sufficient capacity could be operated after an appropriate readjustment of valves. For this purpose, a general service pump (used for clean water only), fire pump or another suitable main engine-driven pump or independent pump may be used. Alternatively, subject to the engine manufacturer's approval, the sea water cycle may be periodically connected into the fresh water cycle which, however, is not recommended.

The engine operation with connected the above mentioned stand-by pumps shall be tested during the ship trials.

Other equivalent solutions are subject to PRS consent in each particular case.

22.1.2 On the pipelines, before each engine, a valve shall be installed to control the cooling water flow.

22.2 Arrangement of Pipes and Pipe Connections

22.2.1 Sea-water supply to the cooling system shall be provided by means of at least two inlet valves, one of which shall be located at the bottom, the other one on the side of the ship. These valves shall be interconnected and the cooling water shall be taken from the connecting sea-water main. Handwheels of bottom sea-valves shall be situated in a readily accessible position under the floor.

22.2.2 Where sea valves (chests) are not blown through with compressed air, main engine cooling water outlet shall be connected there by shut-off non-return valve.

22.2.3 Where flexible hose assemblies are used in the sea-water system at the controls of the shut-off valves for the sea-water supply to such assemblies, nameplates shall be affixed to inform of the necessity to close the valves after rendering the engine room out of operation.

22.3 Cooling Water Strainers

On the sea-water main supplying the cooling systems of main and auxiliary I.C. engines water strainers shall be fitted. Cleaning the strainers shall not cause stopping the cooling water supply to the engines. The strainers' bodies shall be made of neither cast iron nor aluminium alloys.

22.4 Cooling of I.C. Engines

22.4.1 In the fresh water cooling system provision shall be made for fresh water expansion tank, in which the level of water shall be higher than the highest level of water in the engine. The expansion tank, which may be serve the cooling system of several engines, shall be connected to the suction piping of pumps and shall be fitted with an alarm to give warning on the minimum water level.

Location of a discharge line in the fresh water cooling system of the engines shall ensure covering with water the uppermost cooled areas of engines, water coolers and oil coolers, and shall prevent the formation of stasis and air-locks. Recovery of the heat from engine cooling for heating of the accommodation spaces shall take place only by separate heat exchanger provided with a by-pass pipe.

22.4.2 If the air for engine cooling is taken from the machinery space, its temperature shall not exceed 45ºC. The engine cooling air shall be extracted outside the machinery space to the open air and shall not be used for direct heating of any spaces.

23 COMPRESSED AIR SYSTEM

23.1 Number of Receivers and Reserve of Starting Air

23.1.1 The reserve of compressed air necessary for starting the main engines and functioning of their control systems shall be stored in at least two receivers or groups of receivers so arranged that they can be used independently. Each of these receivers or groups thereof shall contain a reserve of starting air not less than half of the amount required in paragraph 23.1.2. Where an electric whistle is provided, the reserve of compressed air may be stored in one receiver.

23.1.2 The reserve of compressed air intended for starting the non-reversible main engines driving controllable pitch propellers or connected to other machinery which enable starting the engines without load, shall be sufficient to assure each engine, ready for operation, to make at least 6 starts of each engine from the cold condition. For reversible engines, the reserve of compressed air intended for their starting shall ensure at least 12 consecutive starts, ahead and astern alternately, of each engine from the cold condition.

23.1.3 For starting the auxiliary engine or the unit form which a compressor may be started, at least one receiver of a capacity sufficient for 6 starts shall be provided.

Where one compressed air receiver is fitted, provision shall be made for filling it from the receivers intended for starting of the main engines, however the reverse flow of air shall be precluded.

23.1.4 Where special air receiver is fitted for the whistle, its capacity shall be sufficient to enable the whistle to work continuously for 2 minutes and hourly capacity of the air compressor shall not be less than that required to provide 8 minutes operation of the whistle.

Where it is intended to take air from the receiver provided for the whistle, the capacity of the receiver shall be increased accordingly and automatic topping-up or an alarm device warning in the case when the amount of air in the receiver reaches the lower limit required for the whistle shall be arranged.

23.1.5 Compressed starting air receiver mentioned in paragraph 23.1.1 may provide air for starting the auxiliary engines, for supplying the whistle or for other purposes, provided that the capacity of the air receivers is increased respectively or automatic topping-up of the compressed air receivers is provided.

23.1.6 Compressed air system documentation shall contain the following information:

.1 compressed air consumption for 6 main engine starts from the cold condition at the rated initial pressure and ambient temperature 0ºC;

.2 as in .1 – for the auxiliary engine;

.3 for ambient conditions as in .1 – for 8 minutes of continues operation of the applied whistle.

23.2 Compressors

23.2.1 At least two compressors shall be provided, one of which may be a main engine-driven compressor. Instead of a compressor driven by the main engine, arrangements for charging the starting air receivers from the main engine cylinders may be applied.

23.2.2 In restricted service ships assigned mark III in the symbol of class fitted with the main engine of a rated power not exceeding 88 kW and with no auxiliary engines, one of the compressors may be hand-operated.

Otherwise, a hand-operated compressor may be considered as an additional one, e.g. for charging the starting air receiver for an auxiliary engine which has also another starting arrangement.

23.2.3 Total capacity of power-driven compressors shall be sufficient to charge, within 1 hour, the main engine starting air receivers from the atmospheric pressure to a pressure necessary for the required number of starts and reverses as specified in paragraph 23.1.2.

23.2.4 The capacity of a hand-operated compressor mentioned in paragraph 23.2.2 shall be sufficient to charge, within 1 hour, the main engine one starting air receiver from the atmospheric pressure to a pressure necessary for the required number of starts and reverses as specified in paragraph 23.1.2.

23.3 Starting Air Receivers

23.3.1 Construction of starting air receivers is subject to PRS approval in each particular case.

23.3.2 Each starting air receiver and each group of interconnected starting air receivers shall be provided with a non-disconnectable safety valve.

23.3.3 Safety valves shall be of a spring-loaded type.

23.3.4 Safety valves shall have such discharging capacity that in any conditions the working pressure cannot be exceeded by more than 10%.

23.3.5 Safety valves shall be so designed as to be capable of being sealed or fitted with an equivalent means to prevent their unauthorised adjustment. Materials used for springs and sealing surfaces of valves shall be resistant to corrosive effect of the medium.

23.3.6 Each starting air receiver shall be provided with shut-off devices, installed directly on its body, intended for its disconnecting from the connected pipes.

23.3.7 Each starting air receiver and each group of interconnected starting air receivers shall be provided with a pressure gauge. The same pressure gauge shall also be installed at steering post in the wheelhouse.

23.3.8 After being lifted, safety valves of the starting air receivers for main and auxiliary engines shall completely stop the air escape at the pressure inside the receiver not less than 0.85 of the working pressure.

23.3.9 Where air compressors, reducing valves or pipes from which air is supplied to the receivers are provided with safety valves so adjusted to prevent the receivers from being supplied with air of the pressure higher than the working pressure, safety valves need not be fitted on such receivers fusible plugs shall be fitted on the receivers instead of the safety valves.

23.3.10 Fusible plugs shall have a fusion temperature within 100-130ºC. The fusion temperature shall be permanently marked on the fusible plug. Air receivers having a capacity over 0.7 m^3 shall be fitted with plugs not less than 10 mm in diameter.

23.3.11 Air receivers shall be equipped with water-draining arrangements. In air receivers positioned horizontally, the water draining arrangements shall be installed at both ends of the receiver.

23.4 Arrangement of Pipes and Pipe Connections

23.4.1 Pipes intended for charging starting air receivers shall be led to these receivers directly from the compressors and shall be completely separated from the starting air pipes.

23.4.2 Non-return shut-off valve shall be fitted in the discharge pipe of each compressor.

23.4.3 Each starting air receiver mentioned in sub-chapter 23.1 shall be capable of being charged by each compressor mentioned in sub-chapter 23.2.

23.4.4 The temperature of compressed air entering the receiver from the charging valve on the engine shall not exceed 90°C. Where necessary provision shall be made for an air cooler.

23.4.5 The pipes used for charging the receivers shall not be led under the floor plates.

23.4.6 The connections between the main engine starting air receivers shall be so arranged as to enable the main engine start from both receivers, to enable the air flow between the receivers, to ensure that the air may be used for other purposes from only one of the receivers through a non-return shut-off valve.

23.4.7 Compressed air pipelines shall be so arranged as to preclude the formation of loop water seals.

23.4.8 Non-return valve shall be fitted in the air pipe supplying each engine.

23.4.9 Compressed air supply to the sea chests shall be arranged in accordance with the requirements specified in paragraph 15.9.11.

23.4.10 Overpressure protection of the pipes shall be in accordance with the requirements specified in sub-chapter 15.5.

24 TECHNICAL REQUIREMENTS REGARDING MARINE ENVIRONMENT PROTECTION

In the scope of marine environment protection, the relevant requirements specified in *Part IX* of the *Rules for Statutory Survey for Sea-Going Ships* shall be fulfilled.

25 ADDITIONAL REQUIREMENTS FOR SHIPS WITH ICE CLASS – mark: Lm1

25.1 Effective power of the main engines on the coupling connecting them with shaft shall not be less than 120 kW.

25.2 Propeller shaft dimensions determined in accordance with the requirements specified in sub-chapter 4.5 shall be increased by 5 %.

25.3 Propeller blade thickness determined in accordance with the requirements specified in paragraph 5.2.1 shall be increased by 7 %.

Propeller blade tip thickness shall not be less than $0.005D$.

25.4 For bottom and side sea chests, recirculation of cooling water shall be provided. The diameter of the recirculating water pipe shall not be less than 0.85 of the water outlet pipe.

26 ADDITIONAL REQUIREMENTS FOR TUGS – MARK: hol

26.1 The requirements concerning the allowable equivalent stress in the propulsion system due to torsional vibration are specified in sub-chapter 8.2.

26.2 Exhaust gas system shall fulfil the requirements specified in paragraph 18.1.3.

26.3 In restricted service tugs fitted with one main engine and assigned mark **I** or **II** in the symbol of class, the lubricating oil system shall be served by two pumps of the capacity sufficient to ensure the main engine operation with full power. One of those pumps may be a main engine-driven pump. This requirement also applies to main propulsion gears, hydrodynamic main couplings and hydraulic-compression friction main clutches.

27 ADDITIONAL REQUIREMENTS FOR RESCUE SHIPS – Mark: rat

27.1 If a ship is also intended to operate as a tug, the allowable equivalent stress in the propulsion system due to torsional vibration are specified in paragraph 8.2.1.1.2.

27.2 Exhaust gas system shall fulfil the requirements specified in paragraph 18.1.3.

27.3 In restricted service rescue ships fitted with one main engine and assigned mark **I** or **II** in the symbol of class, the lubricating oil system shall be served by two pumps of the capacity sufficient to ensure the main engine operation with full power. One of those pumps may be a main engine-driven pump. This requirement also applies to main propulsion gears, hydrodynamic main couplings and hydraulic-compression friction main clutches.

28 ADDITIONAL REQUIREMENTS FOR PASSENGER SHIPS – Marks: pas A, pas B, pas C, pas D

28.1 Passenger Ships – Mark: pas A

Passenger ships which are to be assigned the additional mark **pas A** in the symbol of class, irrespective of compliance with the requirements of this Part of the *Rules*, shall comply with the applicable requirements of sub-chapter 2.2 and 2.5 of *Part VII – Main and Auxiliary Machinery and Equipment*, of the *Rules for the Classification and Construction of Sea-going Ships* and as far as practicable and reasonable, the requirements of 3 Chapter II-1, parts A-1, C, E and G and 4 Chapter II-2, part G of *Publication 100/P.*

28.2 Passenger Ships – Marks: pas B, pas C, pas D

Passenger ships which are to be assigned the additional mark **pas B**, **pas C** or **pas D** in the symbol of class, irrespective of compliance with the requirements of this Part of the *Rules*, shall comply with the applicable requirements of sub-chapter 2.2 and 2.5 of *Part VII – Main and Auxiliary Machinery and Equipment*, of the *Rules for the Classification and Construction of Sea-going Ships* and as far as practicable and reasonable, the requirements of 3 Chapter II-1, parts A-1, C, E and G and 4 Chapter II-2, part G of *Publication 100/P.*

28.2.1 For each type of fuel – used in a passenger ship – necessary for its propulsion and power supplies for essential services, 2 oil fuel daily service tanks, each of the capacity sufficient for at least 8 hours (for ships with mark pas B or pas D) and 4 hours (for ships with mark pas C or pas D) of the main propulsion operation with the rated power at the normal load of the generating sets while at sea.

28.2.2 For existing passenger ships constructed or modified before 31 December 2005 and engaged only on short domestic day trips, it is permitted that the main engines and propellers' control stand situated in the engine room be provided with two independent means of two-way communication with the navigation bridge instead of the engine room telegraph required in 1.12.1.

29 ADDITIONAL REQUIREMENTS FOR FISHING VESSELS – Mark: sr

29.1 The requirements concerning the allowable equivalent stress in the propulsion system due to torsional vibration are specified in sub-chapter 8.2.

29.2 In fishing vessels, it is recommended that fixed arrangements be installed for cutting wires and fishing nets, which can twist round the propeller and propeller shaft. Construction of such arrangements is subject to PRS consideration in each particular case.

29.3 It is recommended that fishing vessels, in addition to the requirements specified in this *Part* of the *Rules*, fulfil the relevant optional requirements specified in the *Code of Safety for Fishermen and Fishing Vessels, 2005*.

30 SPARE PARTS

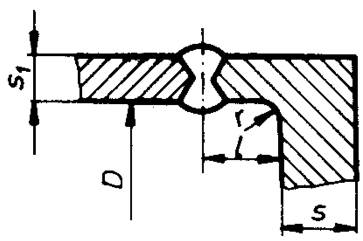
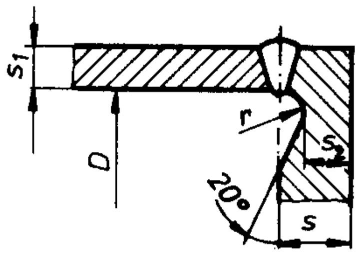
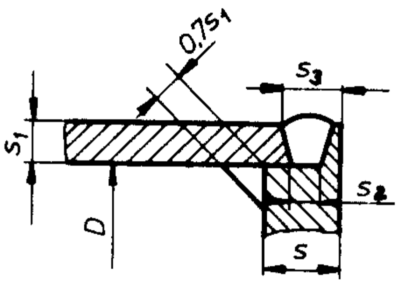
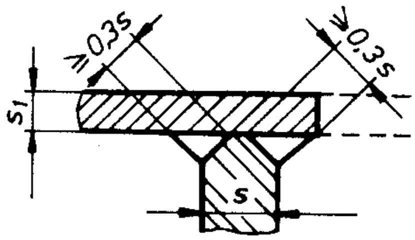
Spare parts for the main propulsion as well as essential machinery in the quantities specified by the Owner for a particular operating area taking account of the machinery manufacturer's recommendations shall be available on board the ship.

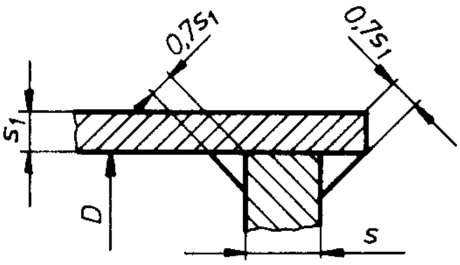
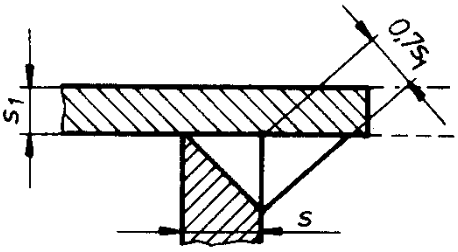
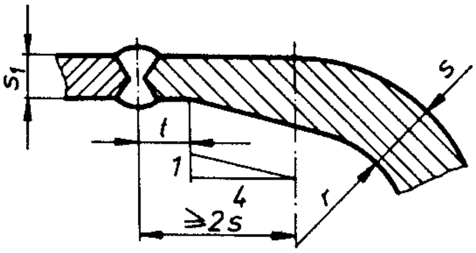
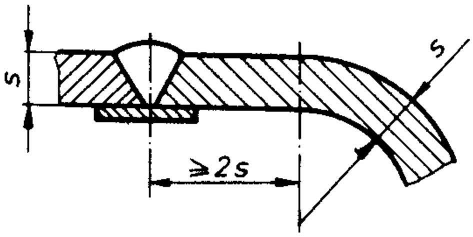
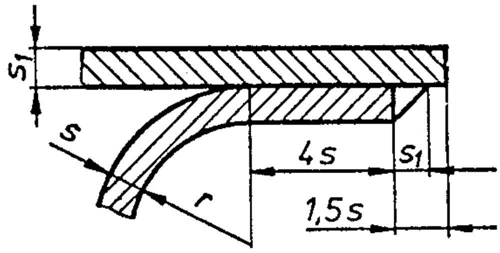
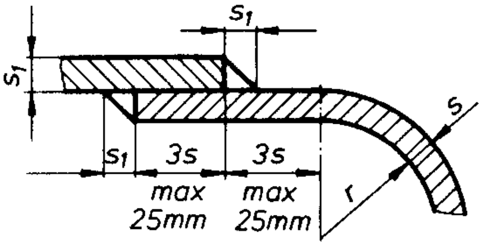
Annex

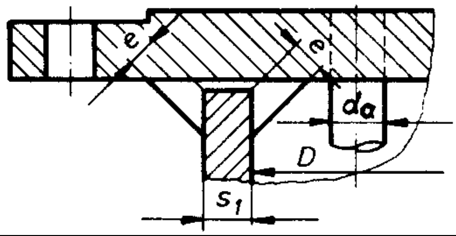
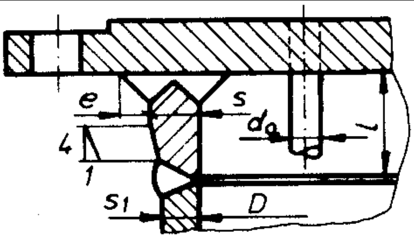
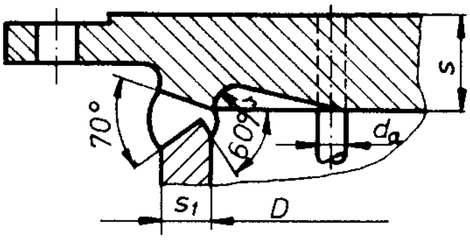
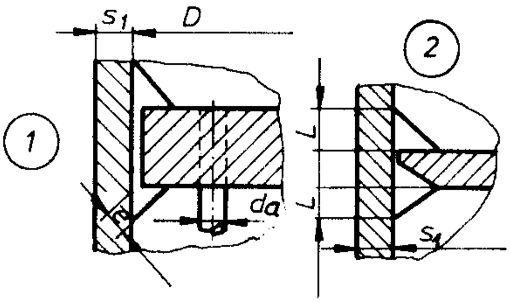
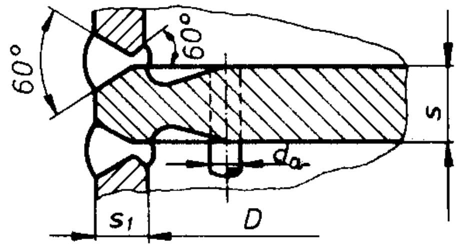
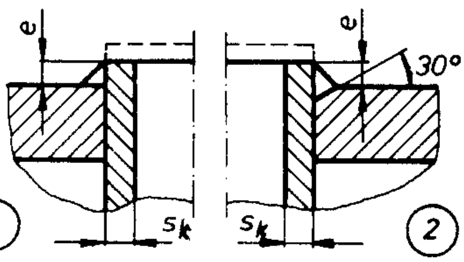
EXEMPLARY WELDED JOINTS USED IN BOILERS, PRESSURE VESSELS AND HEAT EXCHANGERS

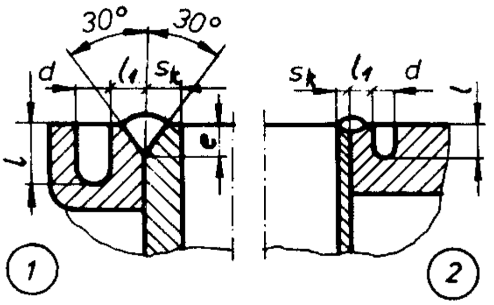
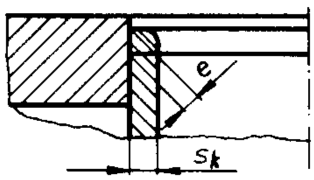
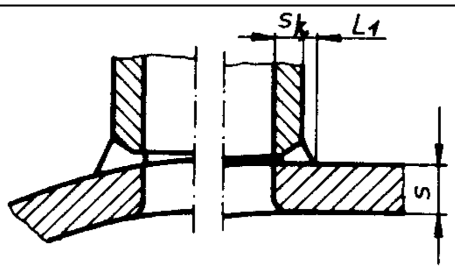
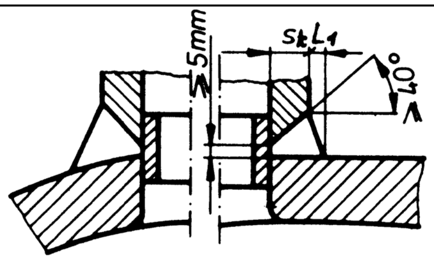
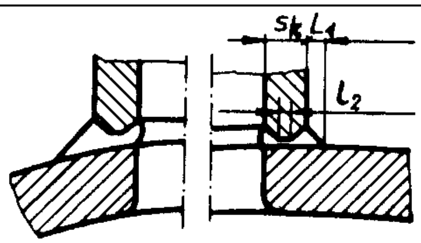
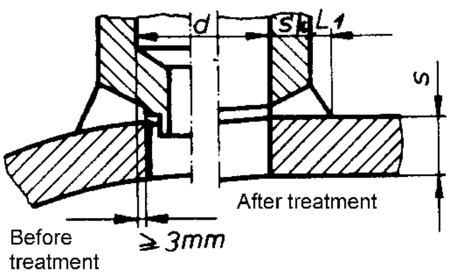
Dimensions of the components prepared for welding as well as the dimensions of welds shall be determined in accordance with the national standards depending on the welding method. The examples of the most frequently used joints are shown in the tables below.

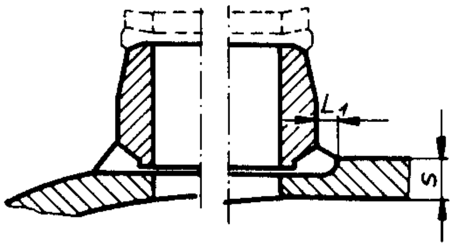
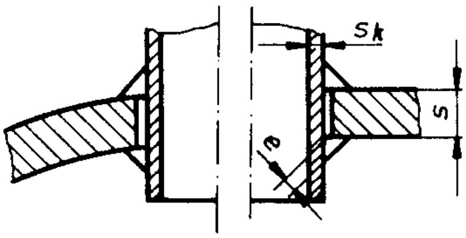
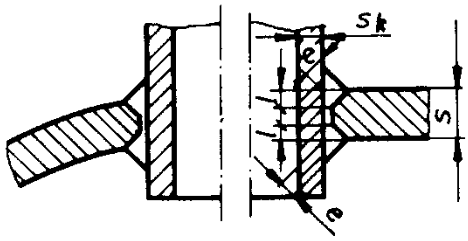
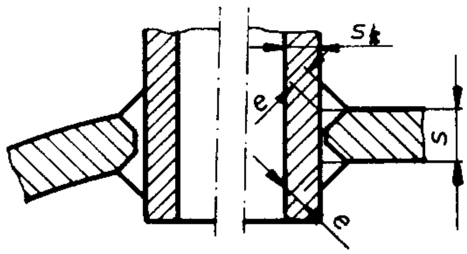
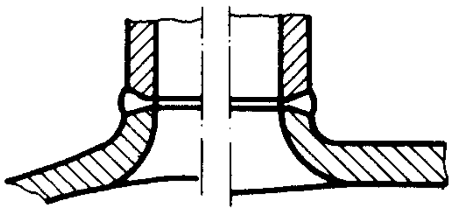
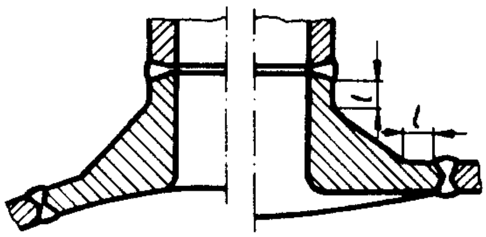
Other welded joints may also be performed having regard to the mechanical properties of parent materials and welding procedure. Such joints as well as the necessary modifications to the exemplary joints are subject to PRS consent in each particular case.

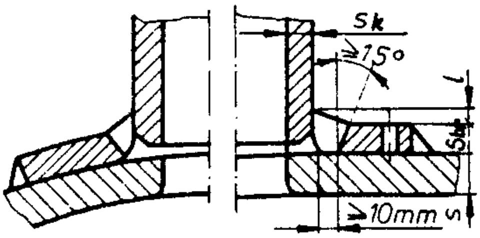
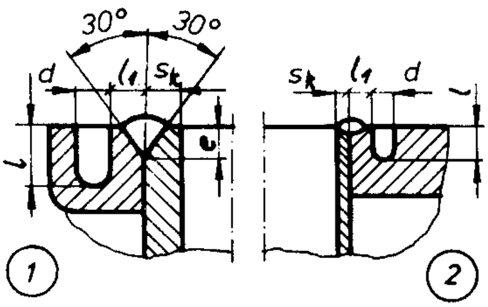
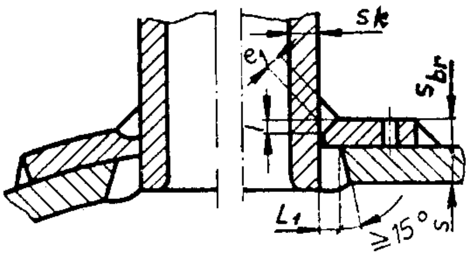
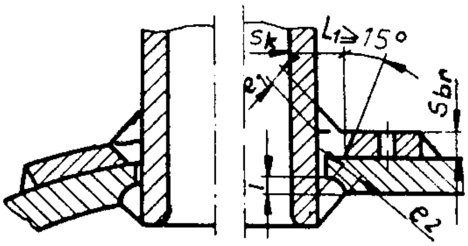
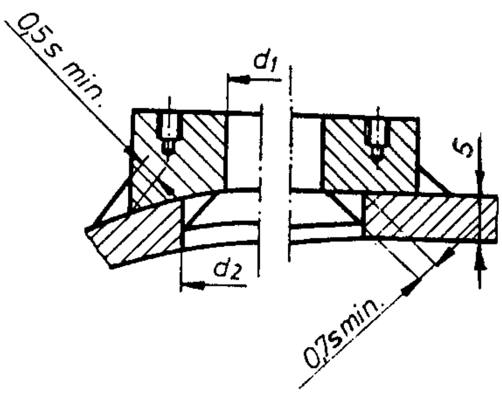
Item	Drawing (example)	Application
1	**Flat end plates and covers**	
1.1		$K = 0.38$ $r \geq \frac{s}{3}$ however not less than 8 mm $l \geq$ s
1.2		$K = 0.45$ $r \geq 0.2\,s$ however not less than 5 mm $s_2 \geq 5$ mm (see Note 1)
1.3		$K = 0.5$ $s_2 \leq s_1$ however not less than 6.5 mm $s_3 \geq 1.24\,s_1$ (see Note 1)
1.4		$K = 0.45$ (see Note 1)

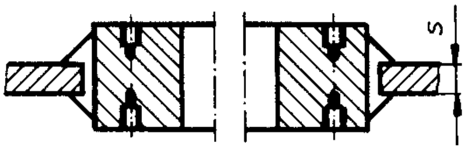
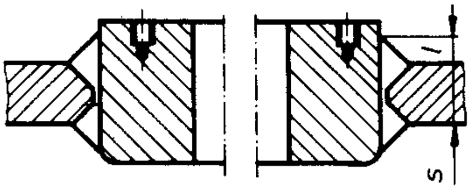
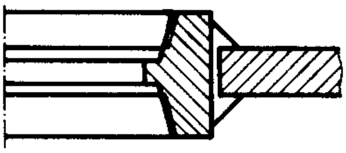
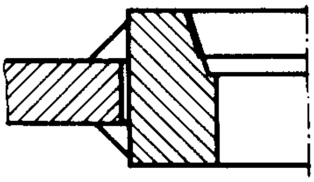
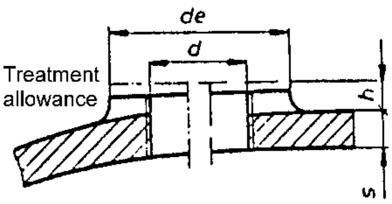
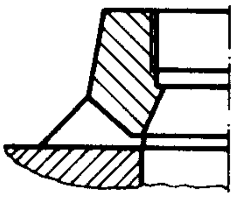
Item	Drawing (example)	Application
1.5		*K* = 0.55 (see Note 1)
1.6		*K* = 0.57
2	**Dished ends**	
2.1		The joint may be used in boilers and pressure vessels of Class I, II and III (see Notes 2 and 17)
2.2		The joint may be used in boilers and pressure vessels of Class II and III
2.3		Not recommendable joint – it may be used only for pressure vessels of Class II not exposed to corrosion $s_1 \leq 16$ mm $D \leq 600$ mm
2.4		The joint may only be used for pressure vessels of Class III $s_1 \leq 16$ mm $D \leq 600$ mm

Item	Drawing (example)	Application
3	**Tube plates**	
3.1		$K = 0.45$ $e = 0.7\ s_1$ $s_1 \leq 16$ mm (see Notes 3 and 4)
3.2		$K = 0.45$ $e = \frac{1}{3} s_1$ $e > 6$ mm $s_1 > 16$ mm (see Notes 5 and 6)
3.3		$K = 0.45$ $r \geq 0.2\ s$ however not less than 5 mm
3.4		$K = 0.45$ $e \geq 0.7\ s_1$ if $L > 13$ mm variant 2 , where $L = \frac{1}{3} s_1$ and $L \geq 6$ mm is recommendable (see Note 7)
3.5		$K = 0.45$ $r \geq 0.2\ s$ however not less than 5 mm
4	**Tubes**	
4.1		$e = s_k$ $e \geq 5$ mm $s_k \geq 2.5$ mm (see Notes 8, 9 and 10)

Item	Drawing (example)	Application
4.2		$d = s_r$; $l_1 = s_r$ $1.5\ s_r < l < 2\ s_r$ alternative 1: $s_r \geq 5$ mm; $l = s_r$ alternative 2: $s_r < 5$ mm (see Note 12)
4.3		$e = 0.7\ s_k$ $s_k \geq 3$ mm (see Note 12)
5	**Branch pieces and joints**	
5.1	**Non-through welded branch pieces**	
5.1.1		$s_k \leq 16$ mm $L_1 = \frac{1}{3} s_k$ however not less than 6 mm
5.1.2		$L_1 = \frac{1}{3} s_k$ however not less than 6 mm (see Note 13)
5.1.3		$L_2 = 1.5 \div 2.5$ mm $L_1 \geq \frac{1}{3} s_k$ however not less than 6 mm (see Note 14)
5.1.4		$L_1 \geq \frac{1}{3} s_k$ however not less than 6 mm (see Notes 15 and 16)

Item	Drawing (example)	Application
5.1.5		L_1 = 10 ÷ 13 mm (see Note 15)
5.2	**Welded penetrating branch pieces**	
5.2.1		Generally, used where $s_k < \frac{1}{2}s$ $e = s_k$
5.2.2		Generally, used where $s_k \cong \frac{1}{2}s$ e = 6 ÷13 mm $e + l = s_k$
5.2.3		Generally, used where $s_k > \frac{1}{2}s$ $e \geq \frac{1}{10}s$ but not less than 6 mm
5.3	**Offset branch pieces**	
5.3.1		
5.3.2		(see Note 17)

Item	Drawing (example)	Application
5.4	**Branch pieces with reinforcing rings**	
5.4.1		$l \geq \frac{1}{3} s_k$ however not less than 6 mm
5.4.2		$l \geq \frac{1}{3} s_k$ however not less than 6 mm $L_1 \geq 10$ mm
5.4.3		$e + l = s_k$ or s_{br} (whichever is lesser) $L_1 \geq 10$ mm
5.4.4		$e_2 + l \geq s_k$ $L_1 \geq 10$ mm $2\ s_k \leq (e_2 + l)$ plus $(s_{br} + e_1)$ or L_1 (whichever lesser of the latest)
5.5	**Pads and branch pieces with threaded holes**	
5.5.1		$d_2 \leq d_1 + 2\ s_{min}$ (see Note 18)

Item	Drawing (example)	Application
5.5.2		$s \leq 10$ mm (see Notes 19 and 20)
5.5.3		$L \geq 6$ mm $s \leq 20$ mm
5.5.4		$s \geq 20$ mm
5.6	**Pads and branch pipes for screw joints**	
5.6.1		
5.6.2		
5.6.3		$d \leq s$ $d_e = 2\ d$ $h \leq 10$ mm $h \leq 0.5\ s$ (see Note 21)
5.6.4		

Notes to the drawings:

1. The joint may be used in such pressure vessels, made of steel, for which $R_m \leq 470$ MPa or $R_e \leq 370$ MPa.
2. The reduction of the thickness of the shell or of the flanged portion of the end plate may be effected in the inside or on the outside.

3. The joint used when welding can be done at either side of the shell.
4. The shells of more than 16 mm in thickness shall have the edges for fillet welds beveled in accordance with drawing 3.2.
5. The joints used when welding is possible at the outside of the shell only.
6. In the shells of no more than 16 mm in thickness the joints may be single-side welded. The breadth of the ring shall not be less than 40 mm.
7. The distance between the internal shell diameter and the external tube plate-diameter shall be as small as practicable.
8. The end of the tube projecting beyond the weld shall be milled or ground.
9. The spacing of the tubes shall not be less than 2.5 s_k and not less than 8 mm.
10. In the case of manual electrical welding the dimension s_k shall not be less than 2.5 mm.
11. Recommendable when maximum reduction of the tube plate deformation occurring in the process of welding is necessary.
12. The tubes shall be welded manually by electric arc welding.
13. The backing ring shall fit tight and shall be removed after welding.
14. The joint used when welding can be done on the inside of the branch piece.
15. The joint used when the size of branch pieces is exceptionally small in respect to that of the vessel.
16. After welding the branch piece shall be machined to the final size.
17. The ring shaped portions *l* shall permit the examination of the joints by X-ray radiography when necessary.
18. The distance between the ring pad and the shell shall not be greater than 3 mm.
19. The distance between the diameter of the opening in the shell and the external diameter of the ring shall be as small as practicable and shall not be greater than 3 mm.
20. The upper and lower bolt holes in the pad shall be shifted in respect to each other.
21. Combined thickness of the ship's shell and deposited welded material shall be sufficient for necessary number of thread turns.

List of amendments effective as of 1 July 2023

Item	*Title/Subject*	*Source*
Page 2	Reference to Publication 100/P, added	PRS
28.1 28.2	Amendments related to the issuance of the new Publication 100/P	PRS
