Polski Rejestr Statków

RULES
FOR THE CLASSIFICATION AND CONSTRUCTION OF NAVAL SHIPS

PART VII

MACHINERY, BOILERS AND PRESSURE VESSELS

2008

GDAŃSK

RULES FOR THE CLASSIFICATION AND CONSTRUCTION OF NAVAL SHIPS prepared and edited by Polski Rejestr Statków S.A. further referred to as PRS consist of the following Parts:

Part I	–	Classification Regulations
Part II	–	Hull
Part III	–	Hull Equipment
Part IV	–	Stability and Subdivision
Part V	–	Fire Protection
Part VI	–	Machinery Installations and Refrigerating Plants
Part VII	–	Machinery, Boilers and Pressure Vessels
Part VIII	–	Electrical Installations and Control Systems
Part X	–	Statutory Equipment

With regard to materials and welding, the requirements of Part IX of the *Rules for the Classification and Construction of Sea-going Ships* apply.

Part VII – Machinery, Boilers and Pressure Vessels – 2008, was approved by the PRS S.A. Board on 24 June 2008 and enters into force on 1 August 2008.

From the entry into force the requirements *of Part VII – Machinery, Boilers and Pressure Vessels* apply to:

– new naval ships, the building contract for which will be signed on or after 1 August 2008 – within the full scope,

– existing naval ships, in accordance with principles specified in *Part I – Classification Regulations.*

The requirements of this Part of the Rules are extended by the below-listed PRS Publications:

Publication No. 4/P	–	Inspection of Mass Produced I.C. Engines
Publication No. 5/P	–	Inspection of Mass Produced Internal Combustion Exhaust Driven Turboblowers
Publication No. 8/P	–	Calculation of Crankshafts for Diesel Engines
Publication No. 28/P	–	Tests of I.C. Engines

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1 GENERAL

1.1 Application

1.1.1 The requirements of this Part of the Rules – *Machinery, Boilers and Pressure Vessels* are applicable to machinery referred to in 1.1.2 intended to be installed onboard naval ships classed by PRS.

1.1.2 The requirements for machinery are applicable to:
- **.1** Diesel engines and gas turbines (further referred to as the engines) for main propulsion;
- **.2** Reduction gears, disengaging and flexible clutches of main propulsion system;
- **.3** Complete power generating sets and their motors;
- **.4** Pumps included into the systems covered by provisions of *Part V – Fire Protection* and *Part VI – Machinery Installations and Refrigerating Plants*;
- **.5** Air and refrigerating compressors;
- **.6** Blowers and turbochargers;
- **.7** Fans included into the systems covered by the provisions of *Part VI – Machinery Installations and Refrigerating Plants*;
- **.8** Fuel and oil separators;
- **.9** Steering gears;
- **.10** Windlasses;
- **.11** Towing and mooring winches;
- **.12** Hydraulic drives;
- **.13** Thrusters;
- **.14** Steam boilers including exhaust gas boilers and steam superheaters of working pressure 0.07 MPa and more;
- **.15** Heating oil boilers;
- **.16** Water boilers with water temperature exceeding 115 °C;
- **.17** Boiler economisers of working pressure 0.07 MPa and more;
- **.18** Liquid fuel firing equipment of boilers;
- **.19** Condensers;
- **.20** Pressure vessels and heat exchangers containing in working condition entirely or in part gas or steam of working pressure 0.07 MPa or more, for which the product of pressure (MPa) and volume (dm^3) amounts to 30 and more;
- **.21** Coolers and fuel, oil and water filters for main and auxiliary engines;
- **.22** Air coolers and air heaters of working pressure in the air space 0.07 MPa and more.

1.2 Definitions and Explanations

Definitions and explanations relating to general terminology of the *Rules for the Classification and Construction of the Naval Ships* (hereinafter referred to as the *Rules*) are given in *Part I – Classification Regulations.* Where in *Part VII* definitions from other Rule parts have been used, reference to those parts is given. For the purpose of this Part of the Rules, the following definitions have been adopted:

Boiler design capacity – a maximum hourly amount of steam that can be generated by the boiler at design pressure and temperature on continuous-duty runs.

Design pressure – manometric pressure, not lower than the opening pressure of safety valves or other protecting devices, taken for the strength calculations.

Design wall temperature – a temperature in the midst of the wall thickness used in calculation of allowable stress according to the ambient temperature and the heating conditions.

Product – engine, boiler, or any appliance and mechanism subject to the requirements of this Part of the *Rules*.

Product certificate – acceptance certificate issued by PRS on the basis of the carried out survey of the product manufacture and testing.

Product type approval certificate – a certificate issued by PRS after completion of appropriate approval procedure, which authorises technical services of the manufacturer to perform inspections and issue acceptance documents for the products covered by the certificate. PRS may consider such manufacturer's acceptance documents as equivalent to its certificates.

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

1.3 Scope of Supervision

1.3.1 Machinery items intended for the naval ship propulsion and steering, systems and installations indispensable in the ship normal operation and the ship specific systems are subject to the PRS supervision exercised in the process of construction.

Machinery technical documentation approved by the PRS and additional requirements, if any, consequent upon NATO standardisation agreements, standards and international conventions, are the basis for survey.

1.3.2 Each machinery item referred to in 1.3.1 shall be subjected to testing carried out according to testing programme approved by PRS. Testing results are considered positive when it has been ascertained that they conform to design data and the acceptance criteria adopted for particular tests have been complied with. The machinery testing and essential checks shall be performed in the presence of the surveyor to PRS.

1.3.3 On the basis of performed manufacture survey and positive results of testing, PRS issues the product certificate. PRS reserves itself the right to issue the certificate after completion of sea trials.

1.3.4 Acceptance document, issued by the manufacturer for the machinery item holding valid *Product Type Approval Certificate*, may be equivalent to the PRS issued certificate (see also 1.2).

Such document shall include the following clause:

„The product approved by PRS
Product Type Approval Certificate for Naval Ships, No. ....
Valid until ....”

1.4 Technical Documentation

1.4.1 General Requirements

The technical documentation including items listed below shall be forwarded to PRS for approval prior to commencement of equipment construction. The documentation should be submitted in triplicate.

1.4.2 Documentation for Approval of Diesel Engines

1.4.2.1 The following documentation of diesel engines shall be forwarded to PRS for engine type approval:

.1 Technical description and basic technical specifications, including specification of the appliances mounted on the engine;
.2 Data for crankshaft calculation acc. to *PRS Publication No. 8/P – Calculation of Crankshafts for I.C. Engines*;
.3 Engine transverse sectional drawing containing assembly dimensions;
.4 Engine longitudinal section drawing containing assembly dimensions;
.5 Drawing of bedplate or crankcase, cast or welded, with welding details and procedure;
.6 Thrust bearing assembly drawing[3)];
.7 Drawing of thrust bearing bedplate, cast or welded, with welding details and procedure;
.8 Tie rods drawings;
.9 Assembly drawing of cylinder cover;
.10 Drawing of cylinder jacket or engine block[1), 2)];
.11 Drawing of cylinder liner;
.12 Drawing of crankshaft with details;
.13 Drawing of crankshaft assembly;
.14 Drawing of thrust or intermediate shaft (if integral with the engine);
.15 Drawing of connecting bolts of clutches;
.16 Drawing of counterweights with connecting bolts (if not integral with the crankshaft);
.17 Drawing of connecting rod;
.18 Drawing of connecting rod assembly[2)];
.19 Drawing of crosshead assembly[2)];
.20 Drawing of piston assembly;
.21 Drawing of camshaft drive assembly;
.22 Material specifications of essential parts with detailed information on non-destructive and pressure tests;
.23 Arrangement of foundation bolts;
.24 Schematic layout of starting air pipe system or other equivalent documents[5)];
.25 Schematic layout of fuel oil pipe system or other equivalent documents[5)];
.26 Schematic layout of lubricating oil pipe system or other equivalent documents[5)];

.27 Schematic layout of cooling water pipe system or other equivalent documents[5];
.28 Schematic layout of the engine control system and safety systems[5];
.29 Assembly drawing of shielding and insulation of exhaust pipes;
.30 Shielding of high pressure fuel pipes;
.31 Crankcase safety devices and their arrangement[4];
.32 Engine maintenance and service manual[6), 8];
.33 Engine type test programme;
.34 Engine test programme[7].

References:

[1] For one cylinder only.
[2] Required when the engine cross sections do not contain all the details.
[3] If integral with engine but not built in the bedplate.
[4] Only for engines with cylinder bore exceeding 200 mm.
[5] The system documentation shall include the components delivered by the engine manufacturer.
[6] Single copy only.
[7] Not requested for engine production in single pieces.
[8] Operation and service manuals must contain requirements for engine operation (repair and servicing), detailed information about special tools and control and measuring equipment (including their outfit and settings) as well as about tests which are required to be made after completion of repair and maintenance.

1.4.2.2 Documentation of turbo-blowers, air coolers, etc. – see 1.4.3 and 1.4.7.

1.4.2.3 Updated documentation of the engine type is the base for PRS' supervision during the diesel engine manufacturing.

1.4.2.4 If the diesel engine is being built under licence and the engine manufacturer does not posses *Type Approval Certificate* for the engine, he ought to provide the documentation in the scope given in 1.4.2.1 with a detailed listing of the introduced changes as compared with the approved type. PRS may dispense from repeating the type approval procedure, provided that the manufacturer has an approved quality management system and has gained confirmation of presented changes by the licence holder – who possess the *Type Approval Certificate*.

1.4.3 Documentation for Approval of Gas Turbines

1.4.3.1 The following documentation shall be supplied to PRS for type approval of a gas turbine:

.1 Technical description and basic technical specifications including diagram of power and rotations versus inlet air temperature, as well as the requirements for exhaust gas system and air intake system;
.2 Assembly drawings and sectional drawings with assembly dimensions;
.3 Drawings of casings, rotors, vanes and vane attachment, seals, bearings, fuel injectors and combustion chambers, heat exchangers integral with the turbine, together with specification of the materials;

.4 Specification of mechanical properties and chemical composition of the materials used. For parts and materials working in temperature over 400 ºC, the detailed temperature related mechanical, creep and corrosion specifications must be provided;
.5 Technical details of heat treatment for essential parts;
.6 Drawings of thermal insulation;
.7 Foundation and attachment drawings;
.8 Diagram of temperature field in the turbine at rated nominal power and at a maximum allowable short time power;
.9 Strength calculations of rotors, vanes and vane attachment;
.10 Torsional vibration analysis[1)] and if applicable vanes vibration calculations;
.11 Gas turbine strength analysis for the whole operational life period, of components that are highly loaded and work in the highest temperature, taking into account creep behaviour and strength and high temperature corrosion;
.12 Diagrams of rotation speed control and adjustment system, alarm and safety system;
.13 Detailed information regarding speed governors and safety controller;
.14 Diagrams of lubrication and fuel systems;
.15 Rotor balancing procedure;
.16 Failure analysis and the safety system effectiveness analysis;
.17 Gas Turbine Type Test Programme[2)];
.18 Gas Turbine Test Programme[2)];
.19 Operation Manual including Manual for Emergency Situation Measures;
.20 Instructions for Preventive Maintenance.

References:

1) See 1.4.3.3.

2) Test programme shall have acceptance criteria defined. In case of turbines produced as single units separate test and type test programmes are not required.

Note:

In case of gas turbines with a power below 100kW and for turbines dedicated for auxiliary purposes, the scope of the documentation required for approval may be reduced after agreeing it with PRS.

1.4.3.2 Documentation for heat exchangers, see 1.4.7.

1.4.3.3 Updated documentation of the gas turbine type and torsional vibration calculations for the particular drive system are the basis for PRS' supervision of the gas turbine manufacture.

1.4.3.4 If the gas turbine is being built under licence and turbine manufacturer does not posses *Type Approval Certificate* for the turbine, he shall provide the documentation in a scope given in 1.4.3.1 with detailed listing of all introduced changes as compared with the approved type and design. PRS may dispense with repeating type approval procedure, provided the manufacturer has an approved quality management system and has gained confirmation of presented changes by the licence holder – who is in possession of the *Type Approval Certificate*.

1.4.4 Documentation for Approval of Gears, Clutches, Auxiliary and Deck Mechanisms

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

.1 Technical description and basic technical specifications, including specifications of suspended machinery items;
.2 Assembly drawing with cross section and dimensional data;
.3 Drawings of foundations, crankcases, columns, casings, etc. with all details and welding technology;
.4 Drawings of cylinder covers and cylinder liners;
.5 Drawings of piston rods, connecting rods assemblies and pistons;
.6 Drawings of rotors of blowers and compressors;
.7 Drawings of crankshafts and other shafts transmitting torque moment;
.8 Drawings of pinions and toothed gear wheels (see also 4.2.1.2);
.9 Drawings of disengaging and flexible clutches (see also 4.3.1.2);
.10 Drawings of the main machinery thrust bearing if not integrated with the main machinery;
.11 Drawings of torsional vibrations dampers;
.12 Diagrams of control, alarm and safety systems within the machinery installation;
.13 Diagrams of the fuel oil, lubricating oil, cooling water and hydraulic system pipelines within the machinery – including the information on flexible hose assemblies applied;
.14 Thermal insulation drawings, including exhaust pipes;
.15 Drawings of foundations of the main mechanisms, gears, steering gears, windlasses, mooring, towing and sweep winches, as well as winches for the unmanned undersea vehicle and towed sonar;
.16 Material specifications for essential parts with all details on non-destructive testing, pressure testing and special technologies used during manufacturing;
.17 Test programme[1)].

References:

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

1.4.5 Documentation for Approval of Thrusters

1.4.5.1 For type approval of the thrusters the following documentation should be submitted to PRS:

.1 Technical description and basic technical specifications;
.2 Assembly drawing in cross section with dimensions;
.3 Drawings of bodies, shafts and gears;
.4 Drawings of the nozzle and the propeller or other applied propulsion device;
.5 Drawings of pitch control device or vanes of cycloidal type propellers;
.6 Drawings of bearings and dynamic seals of propeller shaft and rotating column;

.7 Hydraulic, electrical and pneumatic diagrams together with specification of the components;
.8 Diagrams of lubricating and cooling system, if applicable;
.9 Diagram presenting starting torque of the engine driving the rotation of the column;
.10 Material specification for all essential parts as mentioned in .3, .4 and .5 with all details on non-destructive testing, pressure testing and special technologies used during manufacturing;
.11 Torsional vibration calculations;
.12 Toothed gears and roller bearings durability calculations;
.13 Operation and Service Manual;
.14 Type testing programme[1)];
.15 Product test programme[1)].

References:

1) Test programme should include acceptance criteria. In the case of production of a single unit separate test programmes for type and product are not required.

1.4.5.2 Updated documentation of the type is a base for PRS' supervision during manufacturing of the thrusters.

1.4.5.3 If the device is being built under licence and device manufacturer does not posses *Type Approval Certificate* for the product type, he shall provide the documentation in a scope given in 1.4.5.1 with a detailed listing of all introduced design changes as compared with the approved type. PRS may dispense with repeating the type approval procedure, provided the manufacturer has an approved quality management system and has gained confirmation of presented changes by the licence holder – who is in possession of the *Type Approval Certificate*.

1.4.6 Documentation for Approval of Hydraulic Propulsion Systems

Documentation of the hydraulic propulsion systems shall include:
.1 Technical description and basic technical specifications;
.2 Diagrams of hydraulic systems and specification of elements, pipes and pipe connectors;
.3 Test programme.

1.4.7 Documentation for Approval of Boilers, 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 this part of *the Rules* and type and arrangement of all welds with dimensions;
.2 Drawings of other boiler, pressure vessels and heat exchangers parts to be surveyed (with the exception of charging air coolers), the dimensions of which are given in *the Rules*;

.3 Arrangement of valves and fittings including their specifications;
.4 Safety valves, their characteristics and data for calculation of their cross-sectional area;
.5 Material specification with all data concerning welding consumables;
.6 Welding and heat treatment procedures;
.7 Diagrams and drawings of boiler firing equipment including the systems of automatic control, safety and signalling;
.8 Test programme.

1.5 Pressure Tests

1.5.1 Parts of Diesel Engines

The components of diesel engines should be subjected to pressure tests according to Table 1.5.1.

Table 1.5.1

Item	Component name		Test pressure [MPa]
1	Cooling space of cylinder cover1)		0.7
2	Cylinder liner over the whole length of the cooled space		0.7
3	Cooling space of cylinder block		1.5p not less than 0.4
4	Exhaust valve cooling space		1.5p not less than 0.4
5	Cooling space of the piston end (where the cooling space is sealed by piston rod or by rod and skirt, test after assembling)1)		0.7
6	High pressure fuel injection system	Fuel injection pump body, pressure side	1.5 p or p+30 whichever less
		Fuel injector	1.5 p or p+30 whichever less
		Fuel injector pipes	1.5 p or p+30 whichever less
7	Hydraulic system (high pressure hydraulic tubes, pumps, servomotors, etc. for exhaust valve control)		1.5 p
8	Turbocharger, cooling space		1.5p not less than 0.4
9	Exhaust pipe, cooling space		1.5p not less than 0.4
10	Engine driven compressors (cylinders, covers, air coolers)	Air space	1.5 p
		Water space	1.5p not less than 0.4
11	Coolers, at both sides2)		1.5 p not less than 0.4
12	Engine driven pumps (lub. oil, water, fuel and bilge pumps)		1.5p not less than 0.4

Notes to Table 1.5.1:

1) For cylinder covers and piston ends made of forged steel, PRS may accept other methods of testing than hydraulic tests, for instance appropriate non-destructive tests and recording of dimension check results.

2) The charging air coolers may be upon PRS consent tested at the water side only.

p – maximum working pressure for the given part.

1.5.2 Machinery Parts and Fittings working under Pressure

1.5.2.1 The parts of machinery and fittings working under pressure should be, after final mechanical machining, but before application of protective coatings, tested with hydraulic pressure determined from the following formula:

$$p_{pr} = (1.5 + 0.1\ K)\ p, \quad [\text{MPa}] \qquad (1.5.2.1)$$

p – working pressure, [MPa];
K – coefficient acc. to Table 1.5.2.1.

In each case, however, the test pressure should not be less than:

- pressure with fully opened safety valve,
- 0.4 MPa for all cooling spaces and their seals, and
- 0.2 MPa in all other cases.

If either the working temperature or working pressure exceeds the values given in Table 1.5.2.1, then the test pressure should be in each case agreed with PRS.

Table 1.5.2.1

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
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	–	–	–	–	–	–	–

1.5.2.2 The pressure tests of machinery parts can be performed separately for each space, applying the test pressure relevant to working pressure and temperature in the given space.

1.5.2.3 Parts or assemblies of engines and machinery containing petrol products or their vapours (reduction gear casings, oil sumps, etc) under hydrostatic or atmospheric pressure should be tested for tightness applying procedure agreed with PRS. In welded structures, only welded joints should be tested for tightness.

1.5.3 Boilers, Pressure Vessels and Heat Exchangers

1.5.3.1 All parts of boilers, pressure vessels and heat exchangers, upon completion of their construction and assembling, should be pressure tested according to Table 1.5.3.1.

Table 1.5.3.1

		Test pressure, [MPa]	
Item	Specification	upon completion of construction or assembling of strength members, less mountings and fittings	upon completion of assembling including mountings and fittings
1	2	3	4
1	Boilers, steam superheaters, economisers and parts thereof operating at temperature below 350 °C	1.5 p_w, not less than $p_w + 0.1$	1.25 p_w, not less than $p_w + 0.1$
2	Steam superheaters and parts thereof operating at temperature exceeding 350 °C	$1.5 p_w \dfrac{R_e^{350}}{R_e^t}$	1.25 p_w
3	Pressure vessels, heat exchangers[1] and parts thereof, operating at temperature below 350 °C and the following pressure: – up to 15 MPa – 15.0 MPa and more[2]	 1.5 p_w, not less than $p_w + 0.1$ 1.35 p_w	 – –
4	Heat exchangers[2] and parts thereof, operating at temperature exceeding 350 °C and the following pressure: – up to 15 MPa – 15.0 MPa and more[2]	 $1.5 p_w \dfrac{R_e^{350}}{R_e^t}$ $1.35\, p_w \dfrac{R_e^{350}}{R_e^t}$	 – –
5	Boiler firing system parts subject to fuel oil pressure	–	1.5 p_w, not less than 1
6	Gas spaces of waste heat boilers	–	air test for pressure equal to 0.01 MPa
7	Boiler mountings and fittings	According to 1.5.2.1, not less than 2 p_w	test of closure tightness for pressure equal to 1.25 p_w

1	2	3	4
8	Boiler feed valves and stop valves of heating oil boilers	2.5 p_w	test of closure tightness for pressure equal to 1.25 p_w
9	Mountings and fittings of pressure vessels and heat exchangers	According to 1.5.2.1	test of closure tightness for pressure equal to 1.25 p_w
10	Thermal oil boilers	1.5 p_w , not less than $p_w + 0.1$	1.5 p_w , not less than $p_w + 0.1$

Notes to Table 1.5.3.1:

[1] Pressure testing must be done separately for each side of the heat exchanger. Testing of coolers of the diesel engines – see Table 1.5.1.

[2] For pressure p_w = 15 to 16.6 MPa, constant value of p_w = 16.6 MPa should be applied

p_w – working pressure, [MPa].

R_e^{350} – yield point of material at temperature 350 °C, [MPa].

R_e^t – yield point of material at working temperature, [MPa].

1.5.3.2 Pressure tests shall be carried out upon completion of all welding operations and prior to the application of insulation and protective coatings.

1.5.3.3 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 assembling.

1.5.3.4 Steam boilers, after being installed on board the naval ship, shall be steam tested under the working pressure.

1.5.3.5 Compressed air vessels, after being installed onboard the naval ship (with fittings and mountings), shall be tested with compressed air under the working pressure.

1.5.4 Hydraulic Propulsion Systems and Elements

1.5.4.1 The hydraulic systems pipings shall be subjected to pressure testing defined in 1.5.4 of *Part VI – Machinery Installations and Refrigerating Plants*.

1.5.4.2 Batteries and hydraulic servomotors shall be pressure tested in scope required for the pressure vessels of appropriate class (see Table 1.5.3.1).

1.5.4.3 Combustible hydraulic liquid receivers shall be pressure tested in scope required for fuel tanks (see subchapter A/6.3 of *Part II – Hull*).

1.6 Materials and Welding

1.6.1 Materials applied for the construction of parts of engines, and other machinery covered by the requirements of *Part VII* should comply with applicable provisions of *Part IX – Materials and Welding*. Examples of applied welded joints are presented in the Annex 1 to this Part of the *Rules*.

1.6.2 Where high strength alloy steels (including creep resisting and heat resisting steels), cast steel or alloy cast iron are used for the construction of machinery parts, the data concerning chemical composition, mechanical and other special properties of material, to prove its suitability for the production of the part in question, shall be submitted to PRS.

1.6.3 Materials used for the parts of gas turbines operating at high temperatures (400 °C and more) shall be subjected to tensile test at the design temperature.

When necessary, PRS may require submission of the data concerning the range of material creep strength at the design temperature.

1.6.4 Carbon and carbon-manganese steels may be used for parts of boilers, pressure vessels and heat exchangers with design operation temperatures not exceeding 400 °C. Low-alloy steel may be used for the components with design temperatures of up to 500 °C.

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

The components and fittings of boilers and heat exchangers operating at temperatures exceeding 500 °C shall be made of alloy steel.

1.6.5 Upon agreement with PRS, hull steels complying with *Chapter 3, Part IX – Materials and Welding* may be used in the construction of pressure vessels and heat exchangers operating at design temperatures below 250 °C.

1.6.6 The use of alloy steels for the construction of boilers, pressure vessels and heat exchangers will be specially considered by PRS. The data on 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 consideration.

1.6.7 Boiler fittings of diameter up to 200 mm for the working pressures up to 1.6 MPa and for temperatures of up to 300 °C, except for safety, feeding and blowdown valves, may be manufactured of ferritic nodular cast iron complying with the requirements of Chapter 15, *Part IX – Materials and Welding*.

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 of Chapter 15, *Part IX – Materials and Welding.*

In other cases the application of cast iron shall be subject of separate consideration by PRS.

1.6.9 Copper alloys may be used for parts and fittings of boilers, pressure vessels and heat exchangers operating at working pressures up to 1.6 MPa and design temperatures up to 250 °C. In other cases the use of copper alloys will be specially considered by PRS.

1.6.10 Seamless pipes should be generally used for parts being the subject of this Part of the *Rules*. However, unless any special reservations have been formulated, upon agreement with PRS, longitudinally or spirally welded pipes may be used, provided their equivalence with seamless pipes has been proved.

1.6.11 Introduction and use of materials with asbestos content is forbidden. This 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) and high pressures (over 7 MPa);

.3 flexible parts of thermal insulation for temperatures over 1000 ºC.

1.7 Heat Treatment

1.7.1 Components in which the material structure may undergo changes as a result of welding or plastic forming shall be subjected to appropriate heat treatment.

In case of heat treatment applied to the welded parts, procedures must comply with the requirements specified in Chapter 23, *Part IX – Materials and Welding*

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 details previously welded;

.3 hot formed parts, when this operation was completed at 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 boilers, heat exchangers and pressure vessels of Class I (see Table 8.1) made of steel, of wall thickness exceeding 20 mm;

.3 boilers, heat exchangers and pressure vessels of Class II (see Table 8.1) made of carbon or carbon-manganese steel of tensile strength greater than 400 MPa and of wall thickness exceeding 25 mm;

.4 heat exchangers and pressure vessels made of alloy steel in case the heat treatment is required by the appropriate standards;

.5 tube plates welded of parts, the annealing being recommended to be carried out prior to drilling the holes.

1.8 Non-Destructive Testing

1.8.1 The non-destructive tests during the manufacturing process should be applied to the following parts of engines and machinery in piece production:

- **.1** cast steel parts including their welded joints (for instance main bearing supports in engine bedplates);
- **.2** crankshafts forged as a single piece;
- **.3** cast steel or forged parts of built-up crankshafts;
- **.4** cast steel or forged parts of semi-built-up crankshafts;
- **.5** connecting rods;
- **.6** piston rods;
- **.7** steel piston ends;
- **.8** tie bolts;
- **.9** bolts subjected directly to variable loads (bolts of main bearings, crosshead bearings and cylinder cover bolts);
- **.10** steel cylinder covers;
- **.11** steel gear wheels of camshaft drive;
- **.12** shafts, rotors and rotor disks of turbines as well as the bolts connecting the casings of high pressure turbines;
- **.13** shafts of main reduction gears and tillers of weight exceeding 100 kg;
- **.14** gear wheels and toothed rims of weight exceeding 250 kg.

1.8.2 The following parts should be subjected to ultrasonic tests confirmed by report signed by the manufacturer:

- parts of diesel engines of cylinder bore up to 400 mm specified in .1, .2, .3, .4, .7 and .10 under 1.8.1,
- parts of diesel engines of cylinder bore exceeding 400 mm specified in .1 through .7 and .10 under 1.8.1,
- rotor vanes of main and auxiliary turbines and main turbines fixed blades.

1.8.3 Surface defect detecting tests with use of magnetic particle crack detection method or with use of liquid dye penetrants at places agreed with the PRS surveyor should be applied to:

- parts of diesel engines of cylinder bore up to 400 mm specified in .1 through .5 under 1.8.1,
- parts of diesel engines of cylinder bore exceeding 400 mm specified in .1 through .11 under 1.8.1,
- rotor vanes of main and auxiliary turbines as well as fixed blades of main turbines.

Threaded parts of tie bolts should be tested at the length equal to double length of the threaded section.

1.8.4 Welded seams inspection with use of the PRS approved methods may be required for essential engine parts important for load bearing.

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

1.8.6 The non-destructive tests should be made in compliance with the provisions of *Part IX – Materials and Welding*.

1.9 General Technical Requirements

1.9.1 The design and make of machinery being the subject of this Part of *the Rules* shall ensure reliable operation thereof under environmental conditions defined in 1.6, *Part VI – Machinery Installations and Refrigerating Plants*.

1.9.2 Oil fuel used in driving engines and firing boilers shall comply with the requirements of 1.18, *Part VI – Machinery Installations and Refrigerating Plants.*

1.9.3 Hot surfaces of machinery, engines, boilers and heat exchangers, shall be insulated in accordance with 1.9.6*, Part VI – Machinery Installations and Refrigerating Plants*.

1.9.4 Fasteners used in moving parts of engines and machinery, as well as those fasteners that are inaccessible, shall be provided with special structural arrangements preventing their loosening.

1.9.5 The piping systems within engines, machinery and boilers shall comply with the relevant requirements of *Part VI – Machinery Installations and Refrigerating Plants.*

1.9.6 The electrical equipment of machinery, engines and boilers shall comply with the requirements of *Part VIII – Electrical Installations and Control Systems.*

1.9.7 The parts of engines and machinery that are in contact with corrosive media shall be made of anticorrosive materials or should have anti-corrosion coatings.

Protecting anodes shall be fitted in cooling spaces of machinery and coolers with sea water circulation.

1.9.8 Engines and machinery shall be provided with such measuring instruments and gauges as are necessary to check their proper operation. The number of measuring instruments and gauges is determined by the manufacturer and they should comply with the requirements of 1.15, *Part VI – Machinery Installations and Refrigerating Plants.*

Monitoring and control instruments for engines to be installed in unattended machinery spaces shall comply with the requirements *of Chapter 21, Part VIII – Electrical Installations and Control Systems*.

1.9.9 Remote and automatic control systems as well as safety and alarm systems of engines and machinery shall comply with the requirements of *Part VIII – Electrical Installations and Control Systems*.

2 DIESEL ENGINES

2.1 General Requirements

2.1.1 The requirements of this Chapter are applicable to all diesel engines of 55 kW and more.

The application of these requirements to engines below 55 kW will be specially considered by PRS in each particular case.

2.1.2 A type of engine shall be defined by:

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

2.1.3 Engines are considered to be of the same type when all the parameters and data specified under 2.1.2 are the same and when there are no essential differences in design, components and materials.

2.1.4 The rated power[*)] should be ensured at the environmental conditions specified in Table 2.1.4.

Table 2.1.4

Ambient conditions	For ships of unrestricted service	For ships of restricted service (outside the tropics)
Atmospheric pressure	100 kPa	100 kPa
Air temperature	+ 45 °C	+ 40 °C
Relative air humidity	60%	50%
Sea water temperature	+ 32 °C	+ 25 °C

2.1.5 Engines and their systems shall be capable of operating for a period not less than 8 hours after the compartment where they are installed has been flooded by sea water up to the lowest end of their main bearings at the less favourable ship’s list.

[*)] As the rated power is assumed the power defined by the manufacturer, developed for unlimited time at the ambient conditions according to Table 2.1.4 with mechanical and thermal load not exceeding the values defined by the manufacturer and confirmed by engine operation test.

2.1.6 The engines for main propulsion shall comply with the requirements of 1.8, *Part VI – Machinery Installations and Refrigerating Plants.*

2.1.7 The engines of emergency power generating sets shall be provided with self-contained fuel, cooling water and lubricating systems.

2.2 Engine Casing

2.2.1 The crankcase and detachable or opened covers of its openings shall be of suitable strength and the fastenings of covers shall be strong enough to prevent displacement of the covers in case of explosion.

2.2.2 The engine casing and adjacent parts shall be provided with draining arrangements (drain grooves, pipes, etc.) or other means preventing penetration of fuel and water into lubricating oil as well as penetration of oil into cooling water.

The cooling spaces shall be fitted with drain arrangements providing for complete drying.

2.2.3 Pipelines connections shall be so designed as to prevent transferring excessive loads on the engine casing. The design of pipelines and servicing platforms shall ensure compensation of elongations due to thermal expansion.

2.2.4 The crankcases are, generally, not to 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 fitted (e.g. to detect the smoke inside crankcase), the vacuum should not exceed 0.25 kPa.

Interconnection of air pipes or lubricating oil drain pipes of two or several engines is not permitted.

The diameter of crankcase venting pipes shall be as small as possible. 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 air vent pipes shall be led to the weather deck to the places excluding the fire-hazardous suction of vapours.

2.2.5 The crankcases of engines of cylinder bore exceeding 200 mm and crankcases exceeding 0.6 m^3 in volume shall be provided with safety devices (explosion relief valves) in the following way:

.1 for engines of cylinder bore not exceeding 250 mm, at least two safety devices shall be fitted, one per each end of the crankcase; for engines having 8 cylinders or more one additional safety device shall be fitted in the middle of crankcase;

.2 for engines of cylinder bore exceeding 250 mm but not exceeding 300 mm, one safety device shall be fitted for every two piston sets (not less than 2 devices for each engine);

.3 for engines of cylinder bore exceeding 300 mm, the number of safety devices shall be equal to the number of piston sets.

2.2.6 The cross-sectional area of explosion relief valve should not be less than 45 cm^2, while the total cross-sectional area of explosion relief valves for a single engine should amount to at least 115 cm^2 per each 1 m^3 of the total crankcase volume. The volume of engine parts permanently fitted inside the crankcase may be deducted from the crankcase design volume.

2.2.7 The safety devices (explosion relief valves) should be in compliance with the following requirements:

.1 they shall be of type approved by PRS;

.2 they shall provide for immediate opening of the valve at the excessive pressure in the crankcase not exceeding 0.02 MPa and its quick closing to prevent the inrush of air into the crankcase;

.3 the valve outlets shall be suitably screened to provide protection for persons being near the engine in the way of flame emission.

2.2.8 On both sides of the engine, on its crankcase covers, plates or notices shall be fitted warning against opening the doors, covers or sight glasses in a period of time necessary for cooling down the engine parts after its stopping. Such warning may be placed in the control position.

2.2.9 When the cylinder bore is 230 mm or more, each cylinder shall be fitted with control device signalling reaching the maximum rated overpressure in cylinders.

2.2.10 Separate spaces of the crankcase such as gears or spaces for chain driving timing gear or similar drives, the total volume of which exceeds 0.6 m^3, shall be equipped with additional explosion relief valves; fulfilling the requirements of 2.2.5 and 2.2.6.

2.2.11 It is recommended that the engine should be provided with:

– high temperature alarm for thrust bearing if the bearing is fitted inside the engine and is connected with crankcase,
– bearings high temperature alarm or an alarm of high oil mist concentration in the crankcase for all engines with cylinder bore exceeding 300 mm or total output over 2250 kW.

2.3 Crankshaft

2.3.1 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 comply with the requirements of the PRS *Publication No. 8/P – Calculation of Crankshafts for Diesel Engines*.

2.3.2 The designs of crankshafts not covered by the PRS *Publication No. 8/P* or crankshafts made of nodular cast iron with $500 \le R_m \le 700$ MPa are subject to special consideration by PRS, provided that complete strength calculations or the experimental data are submitted.

2.3.3 The fillet radius at the shaft junction into flange is not to be less than 0.08 of the shaft diameter.

2.3.4 The surface hardening of crank pins and journals is not to be applied to the fillets, with exception of the cases when the whole shaft has been subjected to surface hardening.

2.3.5 Reference marks shall be provided at the outer side of the crank webs junction with the journals of semi-built crankshafts.

2.3.6 Where the thrust bearing is built into the engine bedplate, the diameter of the thrust shaft is not to be less than that specified in sub-chapters 2.4 and 23.2.2.2 (if applicable) of *Part VI – Machinery Installations and Refrigerating Plants.*

2.4 Scavenging and Supercharging

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

2.4.2 Main engines for which the turbochargers do not provide sufficient charging pressure during engine start and operation at low rotational speed, shall be fitted with additional air charging system enabling to obtain such engine speed at which the required charging will be ensured by the turbochargers.

2.5 Fuel System

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

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

If flexible hoses are used as shielding pipes, these shall be of an approved type.

When in return piping the pulsation of pressure with peak to peak values exceeds 2 MPa, effective shielding of this piping is also required.

2.5.3 All surfaces where temperature exceeds 220 ºC and where there is a risk of fuel stream blow-out from damaged fuel piping shall be properly insulated.

2.5.4 Fuel piping shall be properly (as far as it is technically feasible) covered or protected in other way against fuel leak or spray onto hot surfaces, air inlets for machinery devices or other sources of possible fire. Number of joints in such installation shall be limited to a minimum.

2.5.5 The design of fuel filters fixed on main and auxiliary engines shall be such as to enable their cleaning without engine stopping. In the case a switch-over dual filter is applied, this requirements is considered fulfilled.

2.6 Lubrication

2.6.1 The main and auxiliary engines of output power more than 220 kW shall be equipped with alarm devices giving audible and visible signals in the case of lubricating oil pressure drop at the engine inlet.

2.6.2 Every branch piece supplying lubricating oil to the engine cylinders, as well as the branch pieces installed in the upper part of cylinder liner shall be provided with non-return valves.

2.6.3 The design of oil filters fixed on main engines shall enable their cleaning without stopping the engine.

By-pass filters, i.e. those which transfer only part of the oil delivered by the pump, shall not be applied.

Application in high-speed engines of emergency automatic device directing the lubricating oil to a by-pass filter pipe, is possible only upon PRS consent.

2.7 Cooling

Where telescopic devices are used for cooling the pistons or supplying lubricating oil to the moving parts, protection against water hammer shall be provided.

2.8 Starting Arrangements

2.8.1 Besides the non-return valves required in 17.3.2, *Part VI – Machinery Installations and Refrigerating Plants*, the starting air pipelines of diesel engines shall be provided with bursting disks or flame arresters as follows:

– for reversible engines with main starting manifold – at each branch piece supplying the compressed air to starting valves,
– for non-reversible engines – at the inlet to starting manifold.

This does not apply to engines with cylinder bore below 230 mm.

2.8.2 It is recommended that the engines provided with electrical start be equipped with engine driven generators for automatic charging the starting batteries.

2.8.3 Automatic starting systems of emergency power generating sets engines shall comply with the requirements of 9.5, *Part VIII – Electrical Installations and Control Systems*.

2.9 Exhaust Gas System

In engines fitted with the exhaust gas turbo-blowers operating on the pulse principle, provision shall be made to prevent broken piston rings and valves pieces from entering the turbo-blower.

2.10 Controls and Governors

2.10.1 The main engines shall be fitted with torque limiters (fuel dose) preventing the engine load exceeding the rated torque, resulting from the power output defined for conditions specified in Table 2.1.4.

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

- **.1** the torque limiter should 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 maximum overload torque being indicated on the engine control position;
- **.2** the torque limiter should be set into maximum overload torque and a visual or audible signalling device shall be provided to give a continuous signal when the rated torque is exceeded.

2.10.2 Engines of power generating sets should withstand a short duration overload with torque equal to 1.1 of the rated torque, at rated rotational speed.

The engines of power generating sets shall be fitted with torque limiters (fuel dose) preventing the engine load by a torque exceeding 1.1 of the rated torque, resulting from the output power defined for the conditions specified in Table 2.1.4.

2.10.3 The coefficient of speed fluctuation of power generating sets is not to exceed the values given in 4 of Annex 2 to *Part VIII – Electrical Installations and Control Systems.*

2.10.4 The starting and reversing arrangements shall be made in the way eliminating the possibility of:

- **.1** running the engine in direction opposite to the desired one;
- **.2** reversing the engine when the fuel supply is on;
- **.3** starting the engine before reversing is completed;
- **.4** starting the engine with the turning gear engaged.

2.10.5 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 disengaging clutch or which drives a controllable pitch propeller, shall be provided with separate overspeed governor to prevent the rated speed from being exceeded by more than 20%.

Alternative solution may be approved after PRS considers it eqivalent.

The device preventing from excessive increase of rotational speed, together with dedicated drive system, shall be independent of the required rotational speed governor.

2.10.6 Each engine driving the main or emergency power generator shall be provided with a rotational speed governor which shall meet the following requirements:

- **.1** The momentary variation of the voltage frequency in electric circuit is not to exceed 10% of the nominal frequency. After such variation, the voltage frequency shall reach the rated value after no longer than 5 s period, with the maximum power consumer being switched on or off.

In the case of switching off the consumer of the maximum rated power equal to generator power, the momentary variation of the rotational speed can exceed 10% of the rated speed, provided this does not actuate the device protecting against excessive increase of rotational speed (see 2.10.5);

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

.3 Application of electrical load should be possible with two load steps (see also .4) – so that the generator running at no load could be suddenly 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 should be achieved in not 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 newly applied load;

.4 In special cases, PRS may permit the application of electrical load in more than two load steps in accordance with Fig. 2.10.6.4, provided that this has been already allowed for at the design stage and confirmed by the tests of the ship's electric power plant. In this case, the power of electrical equipment switched on automatically and sequentially after the voltage recovery in bus-bars shall be taken into account, and – for generators operating in parallel – the case of taking over the load by one generator when the other one is switched off shall be also considered;

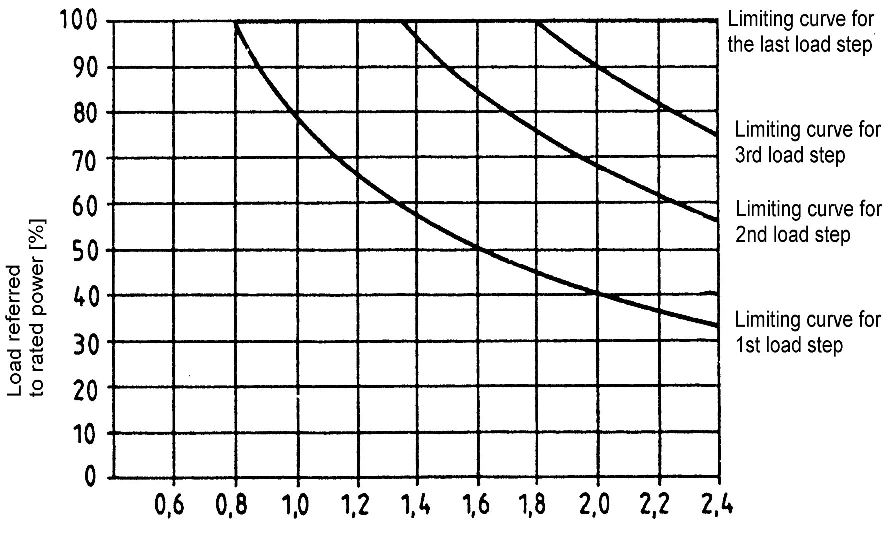


Fig. 2.10.6.4
Limiting curves for loading 4-stroke engines step by step from no load to rated power as the function of the mean effective pressure p_e, [MPa]

.5 Emergency generators shall satisfy the requirements specified in items .1 and .2 at sudden application of the rated load.

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

2.10.7 Power generating sets intended for the parallel operation shall comply additionally with the requirements of 3.2.2, *Part VIII – Electrical Installations and Control Systems*.

2.10.8 Electronic governors of rotational speed should also comply with the relevant requirements of *Part VIII – Electrical Installations and Control Systems*.

2.10.9 Each engine control position shall be equipped with a device for quick manual cutting fuel supply in emergency.

The control devices of the main diesel engine and the engine control position shall comply with the requirements of sub-chapters 1.12 and 1.13 of *Part VI – Machinery Installations and Refrigerating Plants.*

2.11 Torsional Vibration Dampers

2.11.1 The damper design shall be such as to enable taking the work liquid samples.

2.11.2 In general, the engine lubricating oil circulation system shall be used for lubrication of spring-type torsion dampers.

2.11.3 The construction of a damper installed at the free end of crankshaft shall be such as to enable fitting torsional vibration measuring device.

2.12 Supervision, Testing and Certificates

2.12.1 The following essential engine parts are subject to supervision in the process of construction for compliance with the approved documentation:

- crankshaft[M)],
- crankshaft coupling flange (non-integral)[M)],
- crankshaft coupling bolts [M2)],
- piston crowns[M2)],
- connecting rods with bearing caps[M)],
- cylinder blocks and liners[M1)],
- cylinder covers[M1)],
- welded bedplates: bearing stiffening made of forged or cast steel and plates[M)],
- engine frame and crankshaft casing[M)],
- engine load bearing structure[M)],
- tie rods[M)],
- turbocharger shaft and rotor including vanes (it applies also to blowers driven mechanically from the engine shaft like Root's blowers, does not apply to auxiliary blowers)[M1)],

– bolts and studs for: cylinder covers, cylinder blocks, main and connecting rod bearings,
– gear wheels for camshaft drives.

Supervision for compliance with the approved documentation for mass produced engines is subject of a separate consideration by PRS.

Notes and index explanations:

M) – material shall be accepted by PRS.
M1) – material for parts of engines of cylinder bore 300 mm and more should be accepted by PRS.
M2) – material for parts of engines of cylinder bore 400 mm and more should be accepted by PRS.

The above list does not cover compressed air system piping and equipment as well as other pressure systems integral with engines, for which the testing of applied materials may be demanded by PRS.

2.12.2 The supervision of mass produced diesel engines and turboblowers is carried out according to provisions specified in PRS *Publications*: *No. 4/P – Inspection of Mass Produced Internal Combustion Engines* and *No. 5/P – Inspection of Mass Produced Exhaust Driven Turboblowers.*

2.12.3 Upon completion of assembly, adjustment and running in, each engine shall be subjected to running tests at manufacturer's works, according to the test programme agreed with PRS. The tests of diesel engines should be carried out taking into consideration the requirements specified in *PRS Publication No. 28/P – Tests of I.C. Engines*.

2.12.4 Any changes and deviations from the drawings approved in the course of type approval, that the manufacturer intends to incorporate in the engine, shall be presented, and appropriately substantiated, to PRS. The engine testing may be started no sooner than those changes and deviations are approved.

2.12.5 The engine type tests shall be carried out according to test programme, which ensures checking the engine technical condition and its ability for long-time performance.

The engine type tests shall be carried out in accordance with the requirements of PRS *Publication No. 28/P – Tests of I.C. Engines.*

After the change of cylinder number, the engine type re-test may be dispensed with if according to 2.1.3 it has been considered that the engine type is unchanged.

2.12.6 PRS may consider dispensing with the type re-tests of the engine with the rated power increased up to 10%, provided that the engine complies with the requirements of 2.4.

3 GAS TURBINES

3.1 Application

The requirements of this Chapter are applicable to the gas turbines for main propulsion and to the engines driving generators and auxiliaries.

3.2 Definitions and Explanations

3.2.1 For the purpose of this Chapter the following definitions have been adopted:

G a s t u r b i n e – the engine consisting of:
– compressor,
– combustion chamber(s),
– exhaust gas generator turbine and heat exchanger (if applicable),
– power turbine,
together with foundation, control devices and all integrated systems.

P o w e r t u r b i n e – each turbine connected directly or through disengaging clutch with an external power consumer (gear, generator).

3.2.2 In this Chapter, a turbine means an exhaust gas generator turbine as well as power turbine.

3.3 Reference Conditions

The model reference atmosphere shall be taken according to ISO 2314 Standard:
– an ambient temperature of +15 °C,
– a relative humidity of 60%,
– an atmospheric pressure of 101.3 kPa.

3.4 Arrangement

3.4.1 The air-inlet system shall be so located and designed as to preclude, as far as possible, entering harmful foreign matter, including seawater and exhaust gas, into compressor. Where considered necessary, the air intake system shall incorporate filters, de-icing arrangements and washing arrangements for removal of crystallised salt. Bolted connections are not allowed and riveted connections are not recommended in the structure of air-intake system.

3.4.2 The exhaust outlets shall be so located and arranged as to preclude, as far as possible, reverse suction of combustion gases to the compressor. Where considered necessary, the outlets shall be provided with exhaust gases cooling arrangements for reducing the emitted thermal field.

3.4.3 Multi-engine installations shall have inlets and outlets separated, and shall be so designed as to prevent induced circulation through a stopped engine.

3.4.4 Pipe connections shall be made in such a way as to prevent the transmission of excessive loads to the engine casing. Pipes connected to the casings as well as servicing platform gratings shall be so arranged that their thermal expansion is not restricted.

3.4.5 If the temperature on the external surface of the engine casing exceeds 220 °C and the casing cannot be insulated in a way that precludes possibility of leakage of flammable liquid onto that surface, then the turbine shall be fitted within an enclosure. The enclosure shall be provided with appropriate mechanical ventilation, a fire detection system and an automatic fire-extinguishing system.

3.5 General Design Requirements

3.5.1 The engines and associated systems shall be capable of operation for a period not longer than 8 hours when their compartment has been flooded by sea water to the height of the lowest end of their casings.

3.5.2 The design of the gas turbine shall ensure that, after possible failure and separation of any rotor blade, no damage is done to the structure outside turbine and compressor casings. In particular, the possibility of subsequent fire, fuel or any other combustible liquid leak and injury to the personnel should be avoided.

3.5.3 The turbine service life between subsequent major overhauls, as set by the manufacturer and confirmed by tests, in general, is not to be less than 5000 hours, for typical operational conditions of the naval ship.

3.5.4 The insulation of the gas turbines shall comply with the requirements of 1.9.6 of *Part VI – Machinery Installations and Refrigerating Plants.*

3.6 Starting Arrangements

3.6.1 Starting program, if applicable, shall ensure that the starting is aborted if during the starting sequence the appropriate parameters, such as rotating speed of rotors, air pressure after compressor or lubricating oil pressure, are not met.

3.6.2 With the use of automatic or interlocked means, clearing all parts of the gas turbine of the accumulation of fuel shall be ensured before ignition commences on. Prior to each ignition, the purge phase shall be of sufficient duration to displace the turbine volume minimum 3 times.

3.6.3 If the ignition does not occur within a preset time, the control system shall automatically abort the starting operation, shut off the main fuel valve and commence a purge phase.

3.7 Controls and Governors

3.7.1 All turbines shall be provided with overspeed protective device independent of the speed governor, to prevent the r.p.m. exceeding the rated speed by more than 15%.

3.7.2 Each power turbine coupled to reverse gear, electric transmission, controllable-pitch propeller, or other free-coupling arrangement shall be fitted with a separate independent speed governor system. This governor system shall be capable of controlling the speed of the unloaded turbine without bringing the overspeed protective device into action.

3.7.3 Each gas turbine driving the main or emergency electric generator shall comply with the provisions of 2.10.6.1, 2.10.6.2 and 2.10.6.3. However, if the sum of all emergency loads that can be automatically connected is more than 50% of the rated load of the emergency generator, the gas turbine shall be able to accept that sum of the emergency loads.

3.7.4 Gas turbines shall be fitted with automatic control systems to maintain within acceptable limit the temperatures in the following systems, throughout the turbines' normal operating ranges:
– lubricating oil,
– fuel oil (or, in lieu of temperature, viscosity),
– exhaust gas.

3.7.5 Each turbine control post shall be provided with an arrangement for quick manual shutting-off fuel supply in emergency.

Control devices of main propulsion engines and control positions shall comply with the requirements of subchapters 1.12 and 1.13 of *Part VI – Machinery Installations and Refrigerating Plants*.

3.8 Monitoring Systems

3.8.1 Monitoring systems shall comply with the requirements for automation and remote control systems specified in *Part VIII – Electrical Installations and Control Systems*, sub-chapters 20.1, 20.2, 20.3, 20.4. Additionally, the monitoring systems shall comply, to the scope agreed with PRS, with applicable requirements of *Part VIII*, sub-chapters 20.5 and 20.6.

3.8.2 The gas turbines for main propulsion shall be fitted with alarm systems and safety systems in accordance with Table 3.8.2-1. Other engines shall be fitted with alarm systems and safety systems in accordance with Table 3.8.2-2. For engines having a rated power of less then 100 kW, those requirements may be relieved upon an agreement with PRS.

Table 3.8.2-1
Monitoring systems for main propulsion gas turbines

No.	Monitored parameter	Alarm system: monitored value of parameter	Safety system*	Comments
1	Rotational speed **	maximum	shutdown	applies to every exhaust gas generator turbine and power turbine shaft
2.	Lubricating oil pressure	low	–	–
		minimum	shutdown	
3.	Lubricating oil pressure in reduction gear	low	–	–
		minimum	shutdown ***	
4.	Differential pressure across lube oil filter	maximum	–	–
5.	Lubricating oil temperature	maximum	–	–
6.	Fuel supply pressure	maximum		–
7.	Main bearings temperature	maximum	–	–
8.	Fuel temperature	maximum	–	–
9.	Cooling liquid temperature	maximum	–	
10.	Flame and ignition	flame decay and ignition failure	shutdown ***	see also 3.6.2
11.	Starting procedure	starting failure	shutdown	see also 3.6.2
12.	Vibration	high	–	–
		maximum	shutdown ***	
13.	Axial displacement of rotor	maximum	shutdown ***	not applicable to turbines with roller bearings
14.	Exhaust gas temperature	high	–	applicable to combustion chambers and turbines
		maximum	shutdown***	
15.	Vacuum at compressor inlet	high	–	–
		maximum	shutdown	
16.	Control system power supply	minimum	–	applies also to the hydraulic liquid pressure of the speed governor and the safety system servo-motors
17.	Safety system	automatic shutdown	–	applies also to the hand trip

* The shutdown by the safety system means: acting according to the requirements of 3.8.3. After the shutdown, the turbine may be motored.

** It is recommended to set the alarm level at 5÷8% above rated speed. The shutdown level shall be set at 15% above the rated speed.

*** Instead of the automatic shutdown an immediate power reduction to idle may be used, provided that it is proved during the failure mode and effect analysis that it will cause no damage to the turbine or the ship.

Table 3.8.2-2
Monitoring systems for auxiliary gas turbines

No.	Monitored parameter	Alarm system: monitored value of parameter	Safety system	Comments
1.	Rotational speed *	maximum	shutdown	–
2.	Lubricating oil pressure	low	–	–
		minimum	shutdown	
3.	Lubricating oil temperature	maximum	–	–
4.	Exhaust gas temperature	maximum	–	at the turbine inlet
5.	Flame and ignition	flame decay and ignition failure	shutdown	–
6.	Vibration	high	–	–
7.	Control system power supply	power loss	–	–
8.	Safety system	automatic shutdown	–	–

* It is recommended to set the alarm level at 5÷8% above rated speed. The shutdown level shall be set at 15% above the rated speed.

3.8.3 Shutdown by the safety system shall be executed by a quick shutting-off the fuel supply to the turbines, near the injectors.

3.8.4 For main propulsion turbines, it is recommended to provide, besides of alarms listed in Table 3.8.2-1, a low-level alarm for lubricating oil system tank.

3.9 Supervision, Testing and Certificates

3.9.1 Gas turbines intended for naval ships shall be of type approved by PRS.

3.9.2 PRS may agree, after consideration of technical documentation, to the application of the gas turbine that is type approved by another Class Society or by a specialised national agency.

3.9.3 Each gas turbine as described in 3.9.1 and 3.9.2 shall be submitted to the PRS' supervision during manufacturing and tests, according to 3.9.4 to 3.9.19.

3.9.4 PRS' supervision during manufacture and shop tests of the gas turbine comprises:

.1 checking the applied materials and technologies for conformity with approved technical documentation;

.2 checking the configuration for conformity with the approved technical documentation;

.3 product testing, including:
- pressure test of casings, piping and fittings,
- testing the components,
- turbine shop trial.

Product tests shall be carried out in accordance with an approved test programme.

Tests of the components and turbine shop trial shall be attended by the PRS surveyor. Other tests and checks may be carried out by the manufacturer's personnel, if it is allowed by the type approval documentation approved by the PRS and the manufacturer has introduced the quality management system.

3.9.5 The following essential parts of the gas turbines are subject to supervision in the process of construction for compliance with the approved documentation:
- casings of turbines and compressors[M)],
- combustion chambers[M)],
- rotor vanes of turbines and compressors[M)],
- rotor assemblies: shafts, disks, clutches[M)],
- turbine expansion apparatus,
- bolts connecting components of the rotor, casing, clutches[M)],
- dynamic seals,
- pipings and fittings[M)].

Notes and explanations:

[M)] material shall be accepted by PRS.

The above list does not cover compressed air system piping and equipment as well as other pressure systems integral with turbines, whose applied materials may be required to be tested by PRS.

3.9.6 Applied materials supervised in the process of construction in accordance with 3.9.5, as well as welding procedures, heat treatment and other technologies agreed at the approval of technical documentation are subject to checking.

3.9.7 Any changes and deviations of the turbine components from the drawings approved in course of type approval, that the manufacturer suggests to incorporate to the product, shall be presented, appropriately substantiated, to PRS for approval. The certification testing may be started no sooner than those changes and deviations are approved.

3.9.8 All casings shall be tested to a hydraulic pressure according to 1.5.2.1, where the *p* value, used in calculations, is the highest pressure in the casing during normal operation, or the pressure during starting, whichever is higher. For test purposes if necessary, the casings may be subdivided with temporary diaphragms for distribution of test pressure. Heat exchangers shall be pressure tested according to Table 1.5.3.1.

3.9.9 In the course of testing the components, the dynamic balancing of all the compressor and turbine rotors shall be checked. All the rotors shall be tested for strength, for five minutes at 5 percent above the nominal setting of the overspeed protective device, or 15 percent above the maximum design speed, whichever is greater.

3.9.10 The shop trial of the turbine shall be carried out using its intended powered machine. If this is not practical, the test shall be performed with coupling system representing a reaction moment similar to that of the intended driven system. PRS may consider carrying-out, partly or fully, the shop trial aboard the ship.

Shop trial includes:

.1 starting and stopping tests;

.2 checking the engine smooth running at no load;

.3 the test to demonstrate the turbine performance during load alterations that can occur in the real operating conditions, including a 100% instantaneous load shed if it is likely to occur and is acceptable to the propelling system;

.4 the monitoring systems test. During the test, the turbine shall be brought up to its overspeed limit to enable the operation of the overspeed protective device to be checked;

.5 the test to demonstrate power delivery at points along the propeller curve, in the case that the turbine is intended for main propulsion;

.6 the performance test of the turbine, carried out according to the international or national standard accepted by PRS. The test shall be carried out under ambient conditions being as close as possible to the standard reference conditions as set in 3.3. The methods for calculation of rated power for standard reference conditions shall be accepted by PRS;

.7 recording the vibration levels, from zero speed to 110% of rated r.p.m, including starting, running under load and free slowing down.

3.9.11 For turbines driving electric generators it is recommended to check, within shop trials, the capability of delivering 110% of the rated power for a period of 5 minutes and to check the compliance with 3.7.3.

3.9.12 After the trial, a lubricating oil sample shall be tested for traces of metallic and non-metallic particles.

3.9.13 After the shop trial, the visual examination of the gas turbine unit shall be carried out as well as a boroscope inspection of combustion chambers, turbines and compressors.

3.9.14 Certification tests are positively accepted when the test results comply with the design data and for every test the acceptance criteria from the PRS approved test schedule are met.

3.9.15 PRS product certificate for the gas turbine is issued after the acceptance of the complete certification test report. PRS reserves the right to issue the certificate after completion of sea trials.

3.9.16 The sea trials for gas turbine shall be carried out according to an approved test programme. The compliance of the turbine and its installation with the approved documentation shall be demonstrated, as well as its capability to provide the main propulsion or other power delivery in all real variants of running at sea and of manoeuvres. Testing the main and auxiliary propulsion driving turbines shall include, e.g:

– vibration levels measurement and analysis,
– starting test, together with simulated start failures,
– operation test of overspeed safety system,
– test of the fuel treatment system performance,
– test of the reversing system, if applicable,
– checking the proper setting of safety system and alarm system thresholds.

3.9.17 During the test of monitoring systems, the compliance with provisions of 3.8.1 shall be demonstrated.

3.9.18 PRS may require, after sea trials, to open the engine for inner inspection or to carry out the inspection with boroscope.

3.9.19 Upon completion of the sea trials of main propulsion turbines, a copy of the test report shall be submitted to PRS for consideration. PRS may also demand submitting sea trial test reports for gas turbines intended for other purposes.

4 GEARS, CLUTCHES AND COUPLINGS

4.1 General Requirements

4.1.1 The design of the gears shall provide for their normal operation in the conditions defined in p. 1.6, *Part VI – Machinery Installations and Refrigerating Plants*. Flexible and disengaging clutches fitted in propulsion lines at naval ships provided with ice strengthenings shall comply with the requirements of 23.2.4.1 of *Part VI*.

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

.1 Static balancing shall be applied to parts rotating with tangential velocity:
$v \geq 40$ m/s, if subjected to entire 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 tangential velocity:
$v \geq 50$ m/s.

4.2 Reduction Gears

4.2.1 General Requirements

4.2.1.1 The requirements of this sub-chapter are applicable to the propulsion and auxiliary gearing with cylindrical wheels of external and internal meshing having spur or helical teeth of involute profile.

Other types of transmission gear are subject to special consideration by PRS.

4.2.1.2 The technical documentation of reduction gears (see 1.4.4) shall contain all data necessary for design calculation, carried out according to principles specified under 4.2.3. The calculation applies to gear wheels and shafts transmitting the power from the engine output to gear output.

4.2.2 Data for Stress Calculation in Teeth of Gear Wheels

4.2.2.1 Symbols and definitions used in this sub-chapter are based mainly on ISO 6336 Standard, PN-92/M-88509/00 and PN-93/14-88509/01 Standards concerning the calculation of gear load transmission capacity, taking into account the contact stress (according to the method specified under 4.2.4) and bending stress in the tooth root (according to the method specified under 4.2.5).

4.2.2.2 In order to make the requirement provisions more simple, the following nomenclature has been assumed for the pair of gear wheels:

pinion – this gear wheel of the pair that has less number of teeth (all symbols concerning this wheel are indexed with 1),

wheel – this gear wheel of the pair that has greater number of teeth (all symbols concerning this wheel are indexed with 2).

The following symbols are used in the formulae concerning the check calculation of ship gears (gear wheels):

a – wheel gears centre distance, [mm];
b – meshing width (common for the pair of gear wheel), [mm];
b_1 – width of toothed rim – pinion, [mm];
b_2 – width of toothed rim – 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 – maximal power transmitted by the gear (in case of main gears intended for ice strengthened naval ships, the requirements of 23.2.4.1, *Part VI – Machinery Installations and Refrigerating Plants* shall be taken into consideration), [kW];
T_1 – torque transmitted by pinion, [Nm];
T_2 – torque transmitted by wheel, [Nm];
u – gear ratio;
v – tangential velocity at working 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 – substitute 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 – face overlap ratio [–];
ε_β – pitch overlap ratio [–];

ε_γ – total overlap 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 meshing.
2. In the formula defining the teeth contact stress, b is the meshing 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 should be greater than b by more than one module (m_n) at each side.
4. The meshing width b may be used in the formula defining the bending stress in teeth roots if barrel shape or relief of teeth tips has been applied.

4.2.2.3 Selected formulae for meshing

The meshing ratio u is defined as follows:

$$u = \frac{z_2}{z_1} = \frac{d_{w2}}{d_{w1}} = \frac{d_2}{d_1} \qquad (4.2.2.3)$$

where u has the following values:

– positive for external meshing,
– negative for internal meshing.

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

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

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

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

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

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

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

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

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

$$\varepsilon_\alpha = \frac{0.5\sqrt{d_{a1}^2 - d_{b1}^2} \pm 0.5 \cdot \sqrt{d_{a2}^2 - d_{b2}^2} - a \cdot \sin\alpha_{tw}}{\pi \cdot m_n \cdot \dfrac{\cos\alpha_t}{\cos\beta}}$$

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

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

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

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

$$v = \frac{\pi \cdot d_1 \cdot n_1}{60000} = \frac{\pi \cdot d_2 \cdot n_2}{60000}$$

$$d_{w1} = 2a \cdot \frac{z_1}{z_1 + z_2} \;;\quad d_{w2} = 2a \cdot \frac{z_2}{z_1 + z_2}, \quad \text{[mm]}$$

4.2.2.4 Rated static force F_t

The 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 into account the requirements given in 23.2.4.1 of *Part VI – Machinery Installations and Refrigerating Plants*, with the use of the following formulae:

$$T_1 = 9549\frac{P}{n_1} \;;\quad T_2 = 9549\frac{P}{n_2} \qquad (4.2.2.4\text{-}1)$$

$$F_t = 2000\frac{T_1}{d_1} = 2000\frac{T_2}{d_2}, \quad \text{[N]} \qquad (4.2.2.4\text{-}2)$$

4.2.3 The Coefficients Common for the Checked Strength Conditions (contact and bending stresses)

This section defines the coefficients applied in the formulae checking gear wheel teeth strength for contact stress (according to 4.2.4) and for tooth root bending stress (according to 4.2.5). Other coefficients specific for the strength formulae are presented in 4.2.4 and 4.2.5.

All the coefficients shall be calculated from the respective formulae or according to given instructions.

4.2.3.1 Application factor K_A

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

For gears designed for unlimited life time, the K_A must be defined as a ratio of maximal torque arising in the gear (assuming the periodically variable load) in respect to the rated torque.

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

The requirements of 23.2.4.1, *Part VI – Machinery Installations and Refrigerating Plants* shall be also taken into account, if applicable.

K_A factor depends mainly on:

– characteristics of driving and driven equipment,
– masses ratio,
– type of clutches,
– operating conditions (excessive speed – so called overspeed, variation of propeller load, etc.).

The operation conditions shall be carefully analysed for the operation in the vicinity of critical rotational speed.

K_A factor shall be found experimentally or with use of analytical method approved by PRS. If the factor value can not be defined in this way, its value may be taken from Table 4.2.3.1.

Table 4.2.3.1
K_A values depending on the gear applied

Drive operating with gear	K_A values	
	Main drive	Auxiliary drive
Diesel piston engine with hydraulic or electromagnetic sliding clutch	1	1
Diesel piston engine with highly flexible clutch	1.3	1.2
Diesel piston engine with other type of clutch	1.5	1.4
Electric motor	–	1

4.2.3.2 Load distribution factor K_γ

The load distribution factor K_γ takes into account non-uniform distribution in multi-stage or multi-way gears (double tandem, planetary, double helical, etc. gears).

K_γ factor shall be defined as the ratio of maximum load in true meshing to the uniformly distributed load. This factor depends mainly on accuracy and flexibility of gear stages and ways of load distribution.

K_γ factor shall be found experimentally or analytically. If these methods are unavailable, K_γ shall be calculated using the below given formulae:

– for planetary gears:

$$K_\gamma = 1 + 0.25\sqrt{n_{pl} - 3} \qquad (4.2.3.2\text{-}1)$$

where:

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

– for double tandem gears:

$$K_\gamma = 1 + \frac{0.2}{\phi} \qquad (4.2.3.2\text{-}2)$$

where:

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

– for double-helical gears:

$$K_\gamma = 1 + \frac{F_{ext}}{F_t \cdot \mathrm{tg}\,\beta} \qquad (4.2.3.2\text{-}3)$$

where:

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

4.2.3.3 Dynamical factor K_v

The dynamical factor K_v takes into account the dynamical loads arising inside the gear as the result of vibrations of pinion and wheel in respect to each other.

The K_v factor must be defined as the ratio of maximum dynamic load at side surface of tooth in respect to maximum external load defined as ($F_t \cdot K_A \cdot K_\gamma$).

This factor depends mainly on:

– meshing errors (depending on pitch and profile errors),
– masses of pinion and wheel,
– changes in meshing rigidity during the wheel loading cycle,
– tangential velocity at working cylinder,
– dynamical unbalancing of wheels and shaft,
– rigidity of shaft and bearings,
– gear attenuation characteristics.

Where all the following conditions are met:

a) steel gear wheels or wheels with heavy rims are applied,

b) $\frac{F_t}{b} > 150$ [N/mm],

c) $z_1 < 50$,

d) parameter $\frac{v \cdot z_1}{100}$ is within 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 $\frac{v \cdot z_1}{100} < 3$,

dynamical factor K_v may be calculated in the following way:

.1 for spur gears:
K_v – according to drawing 4.2.3.3-2,

.2 for helical gears:
– if $\varepsilon_\beta > 1$
K_v – according to Fig. 4.2.3.3-1,
– if $\varepsilon_\beta < 1$

K_v – is calculated from linear interpolation according to the following formula
$K_v = K_{v2} - \varepsilon_\beta \cdot (K_{v2} - K_{v1})$,
where:
K_{v1} is K_v value for helical gears, see Fig. 4.2.3.3-1,
K_{v2} is K_v value for spur gears, see Fig. 4.2.3.3-2.

.3 The factor K_v for all gear types can be calculated from the following formula:

$$K_v = 1 + K_1 \cdot \frac{v \cdot z_1}{100} \qquad (4.2.3.3.3)$$

where:
K_1 – according to Table 4.2.3.3.

Table 4.2.3.3
K_1 values for calculation of K_v factor

	K_1 values					
	Class of accuracy acc. to ISO 1328					
	3	4	5	6	7	8
Spur gears	0.022	0.030	0.043	0.062	0.092	0.125
Helical gears	0.0125	0.0165	0.0230	0.0330	0.0480	0.0700

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

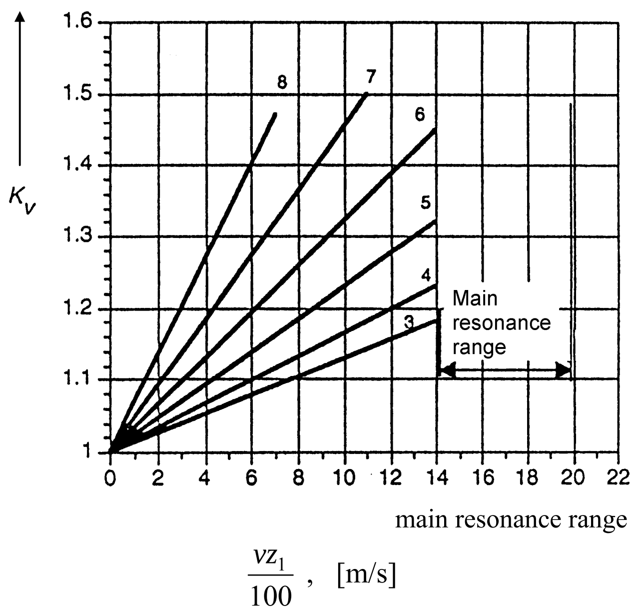
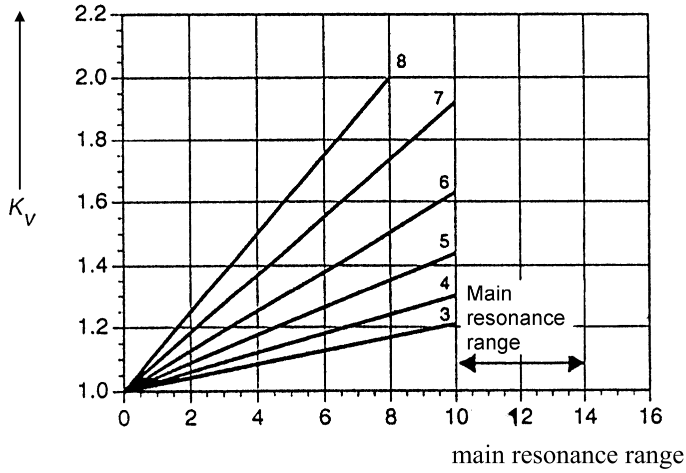


Fig. 4.2.3.3-1 Dynamical factor for helical gears. Accuracy classes 3 – 8 acc. to ISO 1328



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

Fig. 4.2.3.3-2 Dynamical factor for spur gears. Accuracy classes 3 – 8 acc. to ISO 1328

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

4.2.3.4 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 bending stress in tooth root, consider the effects of non-uniform load distribution over the tooth face length.

$K_{H\beta}$ shall be defined as follows:

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

$K_{F\beta}$ shall be defined as follows:

$$K_{F\beta} = \frac{\text{max. bending stress in tooth root}}{\text{mean bending stress in tooth root}}$$

The mean bending stress in tooth root is referred to the considered gear face width b_1 or b_2.

$K_{F\beta}$ and $K_{H\beta}$ factors depend mainly on:

– accuracy of teeth machining;
– assembly errors resulting from hole boring errors;
– bearing clearances;
– misalignment of pinion and wheel axes;
– deformations resulting from insufficient rigidity of gear parts, shafts, bearings, casing and foundation;

– thermal elongations and deformations at working temperature;
– compensating construction of parts (barrel shape, relief of tooth tips etc.).

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

.1 For greater pressure at tooth tips, $K_{F\beta}$ factor must be calculated by the following equation:

$$K_{F\beta} = \left(K_{H\beta}\right)^{N} \tag{4.2.3.4.1}$$

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

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

.2 Where the teeth tips are subjected to small pressure or are relieved (barrel shape, tip relieving):

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

$K_{H\beta}$ contact load longitudinal distribution factor and $K_{F\beta}$ bending load longitudinal distribution factor for tooth root may be found according to the requirements of ISO 6336/1 Standard – method C2.

4.2.3.5 Transverse load distribution factors $K_{H\alpha}$ and $K_{F\alpha}$

Transverse load distribution factors:
$K_{H\alpha}$ – for contact stresses,
$K_{F\alpha}$ – for tooth root bending stresses
consider the effects of pitch and profile errors on the transversal load distribution between two or more pairs of meshing.

$K_{F\alpha}$ and $K_{H\alpha}$ factors depend mainly on:
– general rigidity of meshing;
– total tangential force ($F_t \cdot K_A \cdot K_\gamma \cdot K_v \cdot K_{H\beta}$);
– pitch error on pitch cylinder;
– blunting of tooth tip;
– allowed non-uniformity of tangential velocity.

$K_{H\alpha}$ contact load transverse distribution factor and $K_{F\alpha}$ bending load transverse distribution factor for tooth root, shall be found according to requirements of ISO 6336 Standard – method B.

4.2.3.6 Other methods of selection of the factors than those specified under 4.2.3 may be used for the calculation if approved by PRS.

4.2.4 Contact Stress of Gear Wheels

4.2.4.1 Strength criterion for contact stress is formulated with use of Hertzian formulae for calculation of superficial pressure at active point of meshing (or at internal point of meshing) of single pair of teeth. The contact stress σ_H shall be less or equal to permissible contact stress σ_{HP}.

4.2.4.2 The basic formula of contact stress σ_H is as follows

$$\sigma_H = \sigma_{H0}\sqrt{K_A \cdot K_\gamma \cdot K_v \cdot K_{H\alpha} \cdot K_{H\beta}} \le \sigma_{HP}\,, \quad [\mathrm{N/mm^2}] \qquad (4.2.4.2)$$

where:

σ_{H0} – basic value of contact stresses for pinion and wheel found from the following formula:

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

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

where:

F_t, b, d, u (see 4.2.2);
Z_B – factor of single pair of meshing, for pinion (see 4.2.4.4);
Z_D – factor of single pair of meshing, for wheel (see 4.2.4.4);
Z_H – zone factor (see 4.2.4.5);
Z_E – flexibility factor (see 4.2.4.6);
Z_ε – meshing index factor (see 4.2.4.7);
Z_β – tooth helix angle factor (see 4.2.4.8);
K_A – application factor (see 4.2.3.1);
K_γ – load distribution factor (see 4.2.3.2);
K_v – dynamical factor (see 4.2.3.3);
$K_{H\alpha}$ – transverse load distribution factor (see 4.2.3.5);
$K_{H\beta}$ – longitudinal load distribution factor (see 4.2.3.4).

4.2.4.3 Calculation of permissible contact stress σ_{HP}

Permissible contact stresses σ_{HP} shall be calculated separately for each gear pair (pinion and wheel) from the following formula:

$$\sigma_{HP} = \frac{\sigma_{H\lim}}{S_H} \cdot Z_N \cdot Z_L \cdot Z_v \cdot Z_R \cdot Z_W \cdot Z_X\,, \quad [\mathrm{N/mm^2}] \qquad (4.2.4.3)$$

where:

σ_{Hlim} – fatigue strength of tooth material for contact stress, [N/mm^2] (see 4.2.4.9);
S_H – safety factor for contact stresses (see 4.2.4.14);
Z_N – factor of durability (see 4.2.4.10);
Z_L – factor of lubrication (see 4.2.4.11);
Z_v – factor of velocity (see 4.2.4.11);

Z_R – factor of roughness (see 4.2.4.11);
Z_W – factor of hardness ratio (see 4.2.4.12);
Z_X – factor of size (see 4.2.4.13).

4.2.4.4 Factors of single pair of meshing Z_B and Z_D

Factors of single pair of meshing, Z_B for pinion and Z_D for wheel, consider the effect of tooth side curvature on the contact stress at the point (line) of contact of single pair of teeth in respect to Z_H.

The factors transform meshing pole contact stresses for contact stresses, taking into account tooth side curvature at the central point of contact of single pair of teeth.

Z_B factor for pinion and Z_D factor for wheel shall be defined in the following way:

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

$$Z_B = \max\,(M_1;1) \tag{4.2.4.4-1}$$

$$Z_D = \max\,(M_2;1) \tag{4.2.4.4-2}$$

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 gear wheels,
if $\varepsilon_\beta \geq 1$

$$Z_B = Z_D = 1$$

if $\varepsilon_\beta < 1$, the Z_B and Z_D values shall be found by linear interpolation between Z_B and Z_D values for spur gears and Z_B and Z_D values for helical gears, for which $\varepsilon_\beta \geq 1$.
Thus:

$$Z_B = \max\left\{\left[M_1 - \varepsilon_\beta \cdot (M_1 - 1)\right]\,;\,1\right\} \tag{4.2.4.4-3}$$

$$Z_D = \max\left\{\left[M_2 - \varepsilon_\beta \cdot (M_2 - 1)\right]\,;\,1\right\} \tag{4.2.4.4-4}$$

4.2.4.5 Zone factor Z_H

Zone factor Z_H considers the effect of side curvature of tooth at the meshing pole on the superficial pressure defined by Hertzian formulae as well as the ratio of tangential force at pitch cylinder in respect to normal forces at working cylinder.

Zone factor Z_H shall be calculated in the following way:

$$Z_H = \sqrt{\frac{2\cos\beta_b \cdot \cos\alpha_{tw}}{\cos^2\alpha_t \cdot \sin\alpha_{tw}}} \qquad (4.2.4.5)$$

4.2.4.6 Factor of flexibility Z_E

Factor of flexibility Z_E considers the effect of elasticity properties of material defined by Young's modulus of elasticity and Poisson's number on superficial pressure calculated by Hertzian formulae.

Factor of flexibility Z_E shall be calculated by the following formula:

$$Z_E = \sqrt{\frac{E_1 \cdot E_2}{\pi\left[\left(1-\nu_1^2\right)\cdot E_1 + \left(1-\nu_2^2\right)\cdot E_2\right]}} \quad , \quad [\mathrm{N}^{1/2}/\mathrm{mm}] \qquad (4.2.4.6)$$

where:

E_1, E_2 – Young's modulus of elasticity of tooth material, [N/mm^2];

ν_1, ν_2 – Poisson's number of tooth material, [–].

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

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

ISO 6336 Standard may be used for finding the value of Z_E factor.

4.2.4.7 Factor of meshing index Z_ε

Factor of meshing index Z_ε considers the effect of face meshing index ε_α and pitch meshing index ε_β on the specific contact load of teeth.

Factor Z_ε shall be calculated in the following way:

– for spur wheels using formula:

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}} \qquad (4.2.4.7\text{-}1)$$

– for helical wheels using formula:

where $\varepsilon_\beta < 1$

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}\cdot\left(1-\varepsilon_\beta\right)+\frac{\varepsilon_\beta}{\varepsilon_\alpha}} \qquad (4.2.4.7\text{-}2)$$

where $\varepsilon_\beta \geq 1$

$$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}} \qquad (4.2.4.7\text{-}3)$$

4.2.4.8 Factor of tooth helix angle Z_β

Factor of tooth helix angle Z_β considers the effect of teeth helix angle on the surface durability, considering such variables as load distribution along the contact line. The Z_β factor depends on teeth helix angle only.

The Z_β factor shall be calculated from the formula:

$$Z_\beta = \sqrt{\cos\beta} \tag{4.2.4.8}$$

4.2.4.9 Fatigue strength for contact stress $\sigma_{H\lim}$

For a given material, $\sigma_{H\lim}$ is the value of permissible, repeatable contact stresses that can be transmitted in a continuous way. This value may be considered as a level of contact stresses which can be applied to the material during at least $5 \cdot 10^7$ cycles of load without causing the pitting.

For this purpose the pitting may be defined:

– for not hardened surfaces of teeth, if the pitting area exceeds 2% of total active surface,

– for hardened surfaces of teeth, if the pitting area is greater than 0.5% of total active surface or exceeds 4% of total surface of a single tooth.

The value of $\sigma_{H\lim}$ is equivalent to 1% (or less) probability of failure.

The fatigue strength for contact stress depends mainly on:

– material composition, its uniformity and defects;
– mechanical properties;
– residual stresses;
– hardening process, depth of hardened layer, hardening gradient;
– material inner structure (forged piece, rolled material, cast piece).

Admissible value of contact stress $\sigma_{H\lim}$ shall be found according to the test results of material applied to the construction. If such results are not available, then the contact stresses must be defined according to the requirements of ISO 6336/5 Standard – Quality Class MQ.

4.2.4.10 Factor of durability Z_N

Factor of durability Z_N considers higher allowable contact stresses when limited durability (number of load cycles) are 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.

The Z_N factor shall be defined according to the requirements of ISO 6336/2 Standard – method B.

4.2.4.11 Factors of lubrication, velocity and roughness Z_L, Z_v and Z_R

Factor of lubrication Z_L considers the lubricant type and viscosity, factor of velocity Z_v considers the effect of tangential velocity (v) at working cylinder while the factor of roughness Z_R considers the effect of surface roughness on its durability.

These factors shall be calculated for the more softer material if the co-acting teeth have different hardness.

These factors depend mainly on:
- viscosity of lubrication oil in teeth contact area;
- sum of momentary velocities at the surface of teeth;
- load;
- relative radius of curvature at meshing pole;
- roughness of tooth surface;
- hardness of pinion and wheel.

These factors shall be calculated as follows:

.1 Factor of lubrication Z_L shall be calculated from the formula:

$$Z_L = C_{ZL} + \frac{4(1 - C_{ZL})}{\left(1.2 + \frac{134}{\nu_{40}}\right)^2} \qquad (4.2.4.11.1)$$

where:

ν_{40} – rated kinematic viscosity of oil applied to the gear at temperature of 40 °C.

$$C_{ZL} = \left(\frac{\sigma_{H\lim} - 850}{350}\right)0.08 + 0.83 \quad \text{for } 850 \le \sigma_{H\lim} \le 1200 \quad [\text{N/mm}^2]$$

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

.2 Factor of velocity Z_v shall be calculated from the formula:

$$Z_v = C_{ZV} + \frac{2(1 - C_{ZV})}{\sqrt{0.8 + \frac{32}{v}}} \qquad (4.2.4.11.2)$$

where:

$$C_{ZV} = \left(\frac{\sigma_{H\lim} - 850}{350}0.08\right) + 0.85 \quad \text{for } 850 \le \sigma_{H\lim} \le 1200, \quad [\text{N/mm}^2]$$

Note:
If $\sigma_{H\lim} < 850$ MPa, coefficient $C_{ZV} = 0.85$. If $\sigma_{H\lim} > 1200$ MPa, coefficient $C_{ZV} = 0.93$.

.3 Factor of roughness Z_R shall be calculated from the formula:

$$Z_R = \left(\frac{3}{R_{Z10}}\right)^{C_{ZR}} \qquad (4.2.4.11.3)$$

where:

$$C_{ZR} = 0.32 - 0.0002\sigma_{H\lim} \quad \text{for } 850 \le \sigma_{H\lim} \le 1200, \quad [\text{N/mm}^2]$$

Note:

If $\sigma_{H\lim} < 850$ N / mm^2, coefficient $C_{ZR} = 0.150$. If $\sigma_{H\lim} > 1200$ N / mm^2, coefficient $C_{ZR} = 0.080$.

R_{Z10} – mean amplitude of roughness of co-acting pair of wheels referred to relative radius of teeth curvature, [μm]

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

where:

R_{red} – mean amplitude of roughness height of co-acting pair of wheels (to be calculated according to ISO 6336 Standard), [μm]

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

if the roughness is given as mean value – R_a

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

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

where:

R_{Z1} – roughness height of pinion, [μm];

R_{Z2} – roughness height of wheel, [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:

The roughness shall be measured at sides of several teeth.

ρ_{red} – relative radius of teeth curvature of co-acting gear wheels

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

where:

$$\rho_{1,2} = 0.5d_{b1,2}\,\text{tg}\,\alpha_{tw}$$

Note:

d_{b2} is negative for inner gears.

4.2.4.12 The hardness ratio factor Z_W

The hardness ratio factor Z_W considers the effect of durability of teeth made of soft steel, co-acting with much harder teeth with smooth surface.

Z_W factor is applied only to softer teeth and depends mainly on:

– hardness of softer teeth;
– alloy components of softer teeth;
– roughness of harder teeth sides.

Z_W factor shall be calculated from the following formula:

$$Z_W = 1.2 - \frac{HB - 130}{1700} \qquad (4.2.4.12)$$

where:
HB – Brinell's hardness of softer material,
– for $HB < 130$, $Z_W = 1.2$;
– for $HB > 470$, $Z_W = 1$.

4.2.4.13 The size factor Z_X

The size factor Z_X considers the effect of tooth size on permissible contact stresses, as well as non-uniform properties of materials.

This factor depends mainly on:
- material and heat treatment;
- size of teeth and gear;
- hardening depth ratio in respect to tooth dimensions;
- hardening depth ratio in respect to substitute radius of curvature.

For through hardened teeth and surface hardened teeth with hardening depth appropriate to teeth size and the relative radius of curvature, $Z_X = 1$. If hardening depth is relatively small, the lower values of Z_X shall be chosen.

4.2.4.14 The contact stress safety factor S_H

Numerical value of contact stress safety factor S_H depends on application of the gear, as well as whether it is used as a single unit or as a set of two or more gears.

The safety factor must be selected from Table 4.2.4.14.

Table 4.2.4.14

Gear type	S_H	
	Two and more sets	Single set
Main propulsion	1.2	1.4
Auxiliary	1.15	1.2

For gearing of independent duplicated main propulsion or auxiliary machinery installed onboard the ship in a quantity beyond that is required by *the Rules*, a reduced value of S_H can be assumed at the discretion of PRS.

4.2.5 Bending Stress in Geared Wheel Tooth Roots

4.2.5.1 A criterion for bending strength in tooth root defines permissible level of local tensile stress in the tooth root. The root bending stress σ_F and permissible root bending stress σ_{FP} shall be calculated separately for the pinion and wheel. The σ_F value is not to exceed σ_{FP} value. The following definitions apply to gear wheels with toothed rims thickness greater than 3.5 m_n and 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 according to the requirements of ISO6336 – Method A

4.2.5.2 Basic formula for the bending stress calculation:

$$\sigma_F = \frac{F_t}{b \cdot m_n} \cdot Y_F \cdot Y_S \cdot Y_\beta \cdot K_A \cdot K_\gamma \cdot K_v \cdot K_{F\alpha} \cdot K_{F\beta} \le \sigma_{FP}\,, \quad [\mathrm{N/mm^2}] \qquad (4.2.5.2)$$

where:

F_t, b, m_n (see 4.2.2.2);
Y_F – tooth profile factor (see 4.2.5.4);
Y_S – stress correction factor (see 4.2.5.5);
Y_β – helix angle factor (see 4.2.5.6);
K_A – application factor (see 4.2.3.1);
K_γ – load distribution factor (see 4.2.3.2);
K_v – dynamic factor (4.2.3.3);
$K_{F\alpha}$ – transverse load distribution factor (see 4.2.3.5);
$K_{F\beta}$ – longitudinal load distribution factor (see 4.2.3.4).

4.2.5.3 Basic formula for calculation of permissible bending stress σ_{FP}:

$$\sigma_{FP} = \frac{\sigma_{FE}}{S_F} \cdot Y_d \cdot Y_N \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X\,, \quad [\mathrm{N/mm^2}] \qquad (4.2.5.3)$$

where:

σ_{FE} – fatigue bending strength, [N/mm^2] (see 4.2.5.7);
S_F – root bending safety factor (see 4.2.5.13);
Y_d – design factor (see 4.2.5.8);
Y_N – durability factor (see 4.2.5.9);
$Y_{\delta relT}$ – factor of relative sensitivity to notch effect (see 4.2.5.10);
Y_{RrelT} – relative surface factor (see 4.2.5.11);
Y_X – size factor (see 4.2.5.12).

4.2.5.4 Tooth profile factor Y_F

The tooth profile factor, Y_F , defines the effect of the tooth profile on the nominal bending stress with load applied at the outer contact point of single teeth pair. The Y_F factor shall be determined separately for the pinion and the wheel. In the case of helical gears, the profile factor shall be determined for normal section, i.e. for the virtual spur gear with virtual number of teeth (z_n).

The Y_F factor shall be found from the following formula:

$$Y_F = \frac{6 \cdot \dfrac{h_F}{m_n} \cdot \cos\alpha_{Fen}}{\left(\dfrac{s_{Fn}}{m_n}\right)^2 \cdot \cos\alpha_n} \quad \text{for } \alpha \le 25° \text{ and } \beta \le 30°, \qquad (4.2.5.4)$$

where:

h_F – arm of bending moment for tooth root bending stress calculation, resulting from force applied to outer contact point of single pair of teeth, [mm];
s_{Fn} – chord of tooth root at the critical cross-section, [mm];

α_{Fen} – profile angle at outer contact point of single pair of teeth of normal cross section, [°].

Note: For values used for calculation of Y_F , see Fig. 4.2.5.5.

For calculation of h_F , s_{Fn}, α_{Fen}, guidelines given in ISO6336 standard can be used.

4.2.5.5 The stress correction factor Y_S

The stress correction factor Y_S is applied to change the rated bending stress into local tooth root stress assuming that not only bending stress occurs in the root.

The Y_S factor relates to the force applied to outer contact point of single pair of teeth and shall be found separately for the pinion and the wheel.

The Y_S factor shall be found from the following formula:

$$Y_S = (1.2 + 0.13 \cdot L) \cdot q_S^{\left(\frac{1}{1.12 + \frac{2.3}{L}}\right)} \quad \text{for } 1 \le q_S < 8, \tag{4.2.5.5}$$

where:

q_S – notch parameter determined by the formula:

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

where:

ρ_F – tooth root fillet radius, [mm];

L – tooth bending factor determined by the formula:

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

h_F , s_{Fn} – see 4.2.5.4.

ISO 6336 Standard guidelines can be used for finding ρ_F values.

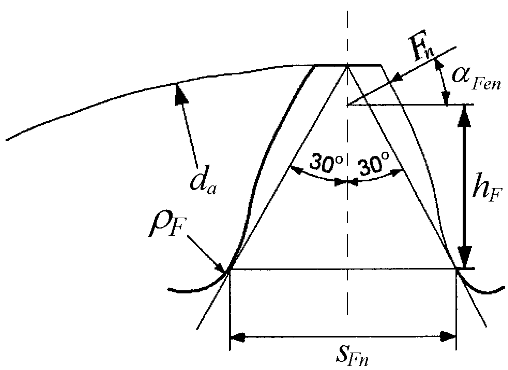


Fig. 4.2.5.5

4.2.5.6 The helix angle factor Y_β

The helix angle factor Y_β considers difference between helix gear and substitute spur gear at normal cross section used in the first step of calculation. In this way better conditions for tooth root stress, resulting from contact lines being deflected along the tooth side, are considered.

The Y_β factor, which depends on ε_β and β, shall be found from the following formula:

$$Y_\beta = 1 - \varepsilon_\beta \frac{\beta}{120} \qquad (4.2.5.6)$$

It is assumed that:
$\varepsilon_\beta = 1$ if $\varepsilon_\beta > 1$, and
$\beta = 30°$ if $\beta > 30°$.

4.2.5.7 The fatigue bending strength σ_{FE}

For a given material the fatigue bending strength σ_{FE} is a value of local stress at the tooth root which can be applied permanently.

According to ISO 6336 Standard, $3 \cdot 10^6$ is the lowest number of load cycles defining the fatigue strength.

The value of σ_{FE} is defined as non-directional pulse stress of minimum value equal to zero (neglecting the residual stress resulting from heat treatment). Other conditions, such as alternating stress, overstress, etc. are taken into account by Y_d design factor.

The value of σ_{FE} corresponds to 1% or lesser probability of failure.

The fatigue strength depends mainly on:
- material composition, its purity and flaws;
- mechanical properties;
- residual stresses;
- hardening process, hardened zone depth, hardness gradient;
- material structure (forged, rolled or cast material).

The fatigue bending strength σ_{FE} shall be defined from the test results of materials applied to the construction. If such results are not available, then the σ_{FE} value shall be defined according the requirements of ISO6336/5 Quality Class MQ Standard.

4.2.5.8 The design factor Y_d

The design factor Y_d considers the effect of load when running astern and overloads from shrink fitting on the strength of tooth root loaded non-directionally, as defined for σ_{FE} .

The Y_d factor for astern run shall be found from Table 4.2.5.8:

Table 4.2.5.8

	Y_d
Generally	1
For gear wheels loaded occasionally with partial power of astern run, such as main wheels of reversing gear	0.9
For idle run gear wheels	0.7

4.2.5.9 The durability factor Y_N

The durability factor Y_N considers the possibility of greater permissible bending stress in the case when limited durability (number of load cycles) of the gear is allowed.

This factor depends mainly on:

– material and hardening;

– number of load cycles;

– factors $Y_{\delta relT}$, Y_{RrelT} , Y_X .

The durability factor must be determined according to requirements of ISO 6336/5 Standard – method B.

4.2.5.10 The factor of relative sensitivity to notch effect $Y_{\delta relT}$

The factor of relative sensitivity to notch effect $Y_{\delta relT}$ presents the range within which the theoretical stress concentration exceeds the fatigue strength.

The $Y_{\delta relT}$ factor depends mainly on the material and relative gradient of stress.

This factor shall be determined in the following way:

– for notch parameters (see 4.2.5.5) within the range $1.5 \le q_S < 4$, $Y_{\delta relT} = 1$;

– for notch parameters outside this range, $Y_{\delta relT}$ shall be determined in accordance with ISO 6336 Standard.

4.2.5.11 The relative surface factor Y_{RrelT}

The relative surface factor Y_{RrelT} considers the dependence of tooth root strength on the condition of transition arc surface, mainly on the roughness amplitude.

The Y_{RrelT} factor shall be found from Table 4.2.5.11:

Table 4.2.5.11

	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

Notes:

1. R_Z – roughness height of transient arc surface between tooth root and rim.

2. If the roughness is defined as mean arithmetic deviation from mean line layout (R_a), then the following relation occurs:

$$R_Z = 6R_a$$

This method can be used only when the scratches and similar surface defects do not exceed $2R_Z$

4.2.5.12 The size factor Y_X

The size factor Y_X considers reduction of strength resulting from increased size of tooth.

The factor depends mainly on:
- material and heat treatment;
- dimensions of tooth and toothed wheels;
- relation of hardening depth in respect to tooth dimensions.

The Y_X factor shall be found from Table 4.2.5.12.

Table 4.2.5.12
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 and through hardened steels
$Y_X = 0.85$	for $m_n \ge 30$	
$Y_X = 1.05 - 0.010\,m_n$	for $5 < m_n < 25$	surface hardened steels
$Y_X = 0.80$	for $m_n \ge 25$	

4.2.5.13 The root bending safety factor S_F

The numerical value of safety factor S_F for bending stresses in teeth roots depends on application of the gear, as well as whether it is used as a single unit or a set of two or more gears.

The S_F factor shall be found from Table 4.2.5.13.

Table 4.2.5.13

Gear type	S_F	
	Two and more sets	Single set
Main propulsion	1.55	2
Auxiliary	1.4	1.45

For independent, duplicated main propulsion gears and gears of auxiliary machinery installed onboard the ship in a number higher than is required for class, a reduced value of S_H can be assumed at the discretion of PRS.

4.2.6 Shafts

Shafts not subjected to considerable variable bending loads have to comply with the applicable requirements of *Part VI – Machinery Installations and Refrigerating Plants.*

Main propulsion gears provided for ice-strengthened naval ships shall comply additionally with the requirements of 23.2.2.2 and 23.2.4.1, *Part VI – Machinery Installations and Refrigerating Plants.*

4.2.7 Gear Wheels Manufacturing – General Remarks

4.2.7.1 Welded gear wheels shall be stress relieved.

4.2.7.2 The tooth rims shrink fitted shall be designed for transmission of torque of at least twice the value of the maximum dynamic torque.

Friction coefficients for shrink fitting calculations shall be taken from Table 4.2.7.2.

Table 4.2.7.2

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

Instead of analytical calculation of shrink fitted tooth rim, the checking of fitting confirmed by test under load (within its full range) may be accepted; the test method and selection of load shall be agreed with PRS.

4.2.8 Bearings of Toothed Gear Shafts

4.2.8.1 The thrust bearing and its foundation shall be of sufficient stiffness to prevent harmful deflection and longitudinal shaft vibration.

4.2.8.2 Roller bearings of the main propulsion gear shall be, as a rule, calculated to life time L_{10} equal to:

– 40 000 hours for propeller thrust bearings;
– 30 000 hours for other bearings.

Shorter life time of bearings may be considered if monitoring equipment of bearing condition is provided for or the bearing inspections are required with proper frequency by the gear service manual.

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

4.2.9 Gear Casings

4.2.9.1 The gear casings and their foundations shall be designed in a way preventing any displacement and distortion in any direction at all service conditions.

Inspection openings are recommended to be done in the casing for making inspection of pinion teeth and toothed wheels possible.

4.2.9.2 Gear casings, both of welded and cast design, shall be, as a rule, subjected to stress relief annealing.

4.2.10 Lubrication

4.2.10.1 Lubrication system shall ensure proper lubrication of all bearings, meshing and other parts requiring lubrication. It shall comply with the requirements of 13.1.3 of *Part VI – Machinery Installations and Refrigerating Plants.*

4.2.10.2 For gears with moderate loads and speeds and with roller bearings, a splash lubrication system may be accepted.

4.2.10.3 Efficient filtering devices shall be provided in pressure oil systems.

The filters of lubricating systems of single main gears shall be so designed that their cleaning be possible without stopping the propulsion system.

4.2.10.4 The pressure lubrication systems shall be provided with pressure and temperature measuring equipment at the inlet and outlet, as well as the alarm signal for low pressure of oil.

The splash lubricating systems shall be provided with equipment measuring the oil level in the gear casing.

For the gears with total output exceeding 20 000 kW or single shaft output exceeding 12 000 kW, the high temperature alarm system shall be provided for all transverse and thrust bearings.

4.3 Clutches and Flexible Couplings

4.3.1 General Requirements

4.3.1.1 The requirements of the present sub-chapter are applicable to clutches and flexible couplings.

4.3.1.2 Documentation of flexible couplings (see 1.4.4.9) shall contain the following particulars:

T_{KN} – nominal rotational torque for continuous operation;
$T_{K\max}$ – maximum rotational torque for temporary running;
T_{KW} – permissible variable rotational torque for all torque loads between no load and T_{KN} ;

$C_{T\ DYN}$ – dynamical stiffness within the whole range of the variation of T_{KN} and T_{KW} torques;
– permissible rotational speed;
– permissible torque transmitted by the limiter of torsion angle (if applied).

Additionally, for information, the following values shall be specified:
- damping coefficient within the whole range of the variation of T_{KN} and T_{KW} torques;
- permissible power loss in the coupling P_{KV} ;
- permissible axial and radial displacement, as well as axis refraction;
- permissible running time of flexible parts until their obligatory renewal.

4.3.1.3 Stiff, torque transmitting elements of couplings (except bolts) shall be made of material of tensile strength 400 MPa < R_m ≤ 800 MPa .

4.3.1.4 Flange connections and connecting bolts shall comply with the requirements of 2.6, *Part VI – Machinery Installations and Refrigerating Plants* and keyless connections – with the requirements of 2.8 of *Part VI*.

4.3.2 Flexible Couplings

4.3.2.1 The flexible couplings shall be so designed as to ensure compensation of deflections in all directions liable to occur in service, without transmitting the impermissible loads to coupled elements.

The couplings shall transfer any thrust acting on the coupled elements during coupling operation.

4.3.2.2 The couplings of main drive diesel engines shall be capable of operation in a specified time without ignition in one of the engine cylinders. Strength calculations shall consider additional loads due to torsional vibrations defined in sub-chapter 4.1 of *Part VI – Machinery Installations and Refrigerating Plants.*

4.3.2.3 The flexible couplings intended for shaft lines of naval ships with one main engine shall be fitted with devices enabling to maintain the ship's minimum speed ensuring its steerability in the case of flexible parts failure.

4.3.2.4 If the requirement of 4.3.2.3 has not been complied with, then the static torque destructive for the flexible parts made of rubber or synthetic materials is not to be less than eight times the rated torque of the considered coupling.

4.3.2.5 The static torque destructive for the flexible parts of couplings intended for power generating sets is not to be less than the torque at short circuit current. If such data are not available, the destructive torque is not to be less than 4.5 times the rated torque of the considered coupling.

4.3.2.6 Flexible couplings shall ensure their normal work at long-term continuous load with the rated torque, within the ambient temperature range from 5 °C to 60 °C.

4.3.2.7 Flexible elements of couplings shall be made of materials resistant to fuel, lubrication oil and hydraulic oil.

4.3.3 Clutches

4.3.3.1 The clutches shall be so designed as to be capable of continuous operation in disengaged condition within rotational speed range between no speed to full rated speed.

4.3.3.2 In case of failure to control system of couplings (hydraulic, pneumatic or electrical), automatic switch to control by emergency power source shall be ensured.

4.3.3.3 Clutches of main engines shall be controlled from the main engine control position, and shall be fitted also with the arrangements for local control. The control devices shall ensure a smooth engagement in such a way that the instantaneous dynamic load does not exceed the maximum permissible clutch torque as defined by the manufacturer, or two times the rated torque of the engine.

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

4.3.3.5 Engaging and disengaging of a clutch cannot lead to overheating its elements.

4.3.3.6 Hydraulic clutches

Continuous operation of driving machinery shall be ensured when there is no hydraulic oil in the connecting clutch within the speed range between no speed and full rated speed.

The lubricating oil applied in diesel engines and gears installed onboard a naval ship is recommended to be used as a hydraulic liquid.

4.3.3.7 Clutches of propulsion systems consisting of various types engines (diesel engines and gas turbines) which are required to be started without stopping or reduction of rotational speed of engines under load

The safety factors of clutch parts loaded with rated torque shall be assumed not less than:

2.5 – for propulsion units provided with speed synchronisation of engines started and engines operating under load;

3.5 – for propulsion units without speed synchronisation.

Where friction couplings are used, their non-slip operation shall be ensured within the load range from 0 to 150% of rated torque.

4.3.4 Emergency clutches

Where the propeller is driven through a hydrokinetic or electromagnetic clutch, provision shall be made for emergency clutches connecting rigidly shaft line items to maintain the ship speed necessary for its steerability in the case of failure of the above-mentioned clutches.

4.4 Supervision, Testing and Certificates

4.4.1 Gears

4.4.1.1 PRS supervision during manufacture and shop test of gears comprises:
- checking the applied materials and technologies for conformity with approved technical documentation,
- checking the configuration for conformity with the approved technical documentation,
- product testing, including pressure test of casings (where applicable), pipings and fittings as well as gear shop trial.

Testing shall be carried out according to the approved tests programme.

The shop trial shall be attended by the surveyor to PRS.

4.4.1.2 After completion of shop trial, the visual external examination of the whole unit shall be carried out and, when required by PRS, internal inspection thereof.

Additionally, a lubricating oil sample shall be tested for traces of metallic and non-metallic particles.

4.4.1.3 PRS product certificate for the gear is issued after acceptance of the complete certification test report. PRS reserves the right to issue the certificate after completion of the sea trials.

4.4.1.4 The sea trials for each gear shall be carried out according to an approved test programme.

4.4.1.5 PRS may require, after sea trials, to open the gear for inner inspection.

Additionally, a lubricating oil sample shall be checked for traces of metallic and non-metallic particles.

4.4.2 Clutches and Flexible Couplings

4.4.2.1 The clutches and flexible couplings shall be of a type approved by PRS.

4.4.2.2 PRS may, after consideration of technical documentation, agree to application of coupling or clutch that has approval certificate issued by a Classification Society or specialised national authority.

In the case of delivery of single coupling/clutch, PRS may agree, after consideration of technical documentation, to application of coupling/clutch that has no type approval certificate. In this case PRS will specify application conditions.

4.4.2.3 PRS' survey during manufacturing and testing of couplings/clutches covers:
- checking the conformity of the applied materials and technologies with approved documentation,
- checking configuration conformity with the approved documentation,
- testing the device.

Testing shall be conducted according to an approved testing programme, in the presence of the surveyor to PRS.

4.4.3 The following essential parts of gears as well as clutches and flexible couplings are subject to supervision in the process of construction for compliance with the approved documentation:
- casings,
- shafts[M)],
- pinions, toothed wheels, toothed rims[M)],
- torque transmitting parts of the clutches/couplings:
  rigid parts[M)],
  flexible parts,
- connecting bolts.

Notes and explanations:

[M)] – material shall be accepted by PRS.

5 AUXILIARY MACHINERY

5.1 Power-driven Air Compressors

5.1.1 General Requirements

5.1.1.1 The compressors shall be so designed that the air temperature at the air cooler outlet does not exceed 90 °C.

5.1.1.2 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 safety valve design shall rule out any possibility of its adjustment or disconnection after being fitted on the compressor.

5.1.1.3 The compressor crankcases of more than 0.5 m^3 in volume shall be fitted with safety valves meeting the requirements of 2.2.5.

5.1.1.4 Delivery stub pipe or the immediate outlet of compressor shall be fitted with a fuse or signalling device activated at a temperature not exceeding 120 °C.

5.1.1.5 Bodies of coolers shall be fitted with safety devices ensuring a free outlet of air in the case of pipes breaking.

5.1.2 Crankshaft

5.1.2.1 The check calculation method specified in 5.1.2.3 and 5.1.2.4 applies to the steel crankshafts of naval ship air compressors and refrigerant compressors with in-line, and V-shaped arrangement of cylinders and with single and multi-stage compression.

5.1.2.2 The crankshafts shall be made of steel having tensile strength R_m from 410 to 780 MPa.

The use of steel having a tensile strength over 780 MPa shall be specially considered by PRS in each particular case.

The crankshafts may be made of nodular cast iron with the tensile strength 500 ≤ R_m ≤ 700 MPa, in accordance with Chapter 15, *Part IX – Materials and Welding*. The crankshafts of other dimensions than those determined by the formulae given below may be admitted – on agreement with PRS – provided that all strength calculations are submitted.

5.1.2.3 Crank pin diameter (d_k) of the compressor is not to be less than that determined by the formula:

$$d_k = 0.25K\sqrt[3]{D^2 p\sqrt{0.3L^2 f + (S\varphi)^2}}\ , \quad [\mathrm{mm}] \qquad (5.1.2.3\text{-}1)$$

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 – diameter of high pressure cylinder),

– for two-stage compression by a tandem piston

$D = 1.4\ D_W$,

– for two-stage compression by a differential piston

$D = \sqrt{D_n^2 - D_W^2}$ (D_n – diameter of low pressure cylinder);

p – for air compressors – compression pressure in high pressure cylinder, [MPa]; for refrigerant compressors, the value of p shall be taken as equal to design pressure at high-pressure side in accordance with 22.2.2 and 22.2.3, *Part VI – Machinery Installations and Refrigerating Plants*;

L – design distance between main bearings, [mm], equal to:

– $L = L'$, when one crank is arranged between two main bearings (L' – true distance between centres of main bearings);

– $L = 1.1L'$, when two cranks, turned by 180°, are arranged between two main bearings;

S – piston stroke, [mm];

K, f, φ – coefficients in accordance with Tables 5.1.2.3-1, 5.1.2.3-2 and 5.1.2.3-3.

Table 5.1.2.3-1
Values of *K* coefficient

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

Table 5.1.2.3-2
Values of *f* coefficient

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

Table 5.1.2.3-3
Values of φ coefficient

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

If shaft journals have co-axial holes with diameters exceeding 0.4 d_k,, then the diameters of the journal shall be determined by the formula:

$$d_{k0} \geq d_k \sqrt[3]{\frac{1}{1-\left(\frac{d_0}{d_a}\right)^4}}\ , \quad \text{[mm]} \qquad (5.1.2.3\text{-}2)$$

d_k – see formula 5.1.2.3-1;
d_0 – diameter of co-axial hole, [mm];
d_a – true diameter of shaft, [mm].

The 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.

5.1.2.4 The thickness of the crank web h_k shall be not less than that determined by the formula:

$$h_k = 0.105 K_1 D \sqrt{\frac{(\psi_1 \psi_2 + 0.4) P C_1 f_1}{b}}, \quad [\text{mm}] \qquad (5.1.2.4\text{-}1)$$

where:
K_1 – coefficient taking into account the influence of shaft material and determined by the formula:

$$K_1 = a \sqrt[3]{\frac{R_m}{2R_m - 430}} \qquad (5.1.2.4\text{-}2)$$

where:
$a = 0.9$ for shafts with entire surface nitrided or submitted to other kind of heat treatment agreed upon with PRS,
$a = 0.95$ for die forged shafts with the fibre continuity being maintained,
$a = 1$ for shafts without heat treatment;
ψ_1 and ψ_2 – coefficients determined in accordance with Tables 5.1.2.4-1 and 5.1.2.4-2;
P – compression pressure taken in accordance with appropriate provisions of 5.1.2.3;
C_1 – distance from the centre of the main bearing to the mid-plane of the crank web, [mm]; when two cranks are arranged between two main bearings the distance to the mid-plane of the web more remote of the support shall be taken;
b – breadth of the crank web, [mm];
f_1 – coefficient taken from Table 5.1.2.4-3;
R_m – tensile strength, [MPa].

Table 5.1.2.4-1
Values of ψ_1 coefficient

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

Explanations:
r – radius of the fillet of the crank web into the crank pin, [mm];
ε – value of overlap, [mm].

For crankshafts without the crank pin overlap, ψ_1 coefficient shall be taken as for $\varepsilon/h_k = 0$.

Table 5.1.2.4-2
Values of ψ_2 coefficient

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

For d_k – see formula 5.1.2.3-1.

The intermediate values of coefficients listed in Tables 5.1.2.4-1 and 5.1.2.4-2 shall be determined by interpolation.

Table 5.1.2.4-3
Values of f_1 coefficient

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

5.1.2.5 The fillet radius at the junction of the crank web with the journal is not to be less than 0.05 of the journal diameter.

The fillet radius at the junction of the coupling flange with the journal or pin is not to be less than 0.08 of the journal diameter.

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

5.2 Pumps

5.2.1 General Requirements

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

5.2.1.2 The pump glands arranged at the suction side are recommended to be fitted with hydraulic locks.

5.2.1.3 If the pump design does not preclude the possibility of pressure rise above the rated value, a safety valve shall be fitted on the pump casing or on the piping before the first stop valve.

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

5.2.1.5 Provision shall be made to prevent water hammer. The use of by-pass valves for this purpose is not recommended.

5.2.1.6 Strength calculation

The critical speed of pump impeller is not to be less than 1.3 of the rated r.p.m.

5.2.1.7 Self-priming pumps

The self-priming pumps shall ensure operation under “dry-suction” conditions; it is also recommended that they are fitted with arrangements preventing the self-priming device being damaged as the result of contaminated water pumping.

5.2.2 Additional Requirements for Flammable Liquid Pumps

5.2.2.1 The pump seals shall be of such design and materials, that in the case of leakage no vapour/air explosive mixtures are generated.

5.2.2.2 Movable seals shall be of design preventing overheating and ignition of seals due to friction of the moving parts.

5.2.2.3 The design of pumps which uses low electrical conductivity materials (plastics, rubber, etc.), shall prevent the accumulation of electric charges (electrostatic energy), or special means for electric charge neutralisation shall be provided.

5.3 Fans, Air Blowers and Turboblowers

5.3.1 General Requirements

5.3.1.1 The requirements of the present sub-chapter are applicable to fans intended for systems covered by requirements of *Part VI – Machinery Installations and Refrigerating Plants,* as well as to internal combustion engine turboblowers and boiler air-blowers.

5.3.1.2 The impellers of fans and air blowers together with couplings as well as the assembled rotors of turboblowers shall be dynamically balanced in compliance with 4.1.2.

5.3.1.3 The suction pipes of fans and air blowers shall be protected against entry of foreign particles.

5.3.1.4 Lubrication system of the turboblower bearings shall exclude the possibility of oil penetration into supercharging air.

5.3.1.5 The turboblowers shall comply with the applicable requirements of Chapter 3.

5.3.1.6 Strength calculation

The impeller parts shall be so dimensioned that at rotational speed equal to 1.3 the rated speed, the resultant stresses at any section are not in excess of 0.95 of the yield point of the material.

Subject to agreement with PRS, other safety factors may be admitted in the case of turboblowers if calculation methods defining maximum local stresses or photo-elasticity methods are applied.

5.3.2 Additional Requirements for Pump Room Fans

5.3.2.1 The air gap between the casing and rotor is not to 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 exceed 13 mm.

5.3.2.2 The terminals of ventilation ducts shall be protected against entry of foreign matters into the fan casings by means of wire net, with square net mesh of the side length not exceeding 13 mm.

5.3.2.3 Pump room ventilation fans shall be of non-sparking design. The fan is considered not sparking if in no conditions there is any risk of sparks generation. Casing and rotating parts of fan shall be made of materials, which do not cause electrostatic charge accumulation, and the fans installed shall be properly earthed to the hull of ship in accordance with the requirements of *Part VIII – Electrical Installations and Control Systems*.

5.3.2.4 Except the cases defined in 5.3.2.5, the impellers and parts of fan casing shielding the impeller shall be made of materials that do not generate sparks, as confirmed by appropriate tests.

5.3.2.5 The tests mentioned in 5.3.2.4 may be dispensed with for the fans made of the following combinations of materials:

.1 impeller and/or casing made of non-metallic materials with anti-electrostatic properties,

.2 impeller and casing made of non-ferrous metal alloys,

.3 impeller made of aluminium or magnesium alloy and steel casing (stainless austenitic steel included), where inside casing, adjoining to the impeller, a ring of adequate thickness made of non-ferrous material is used,

.4 any combination of steel impellers and casings (whereas impeller or casing may be of stainless austenitic steel) when the radial clearance between them is not less than 13 mm.

5.3.2.6 The following materials used for design of fan bodies and rotors are considered as sparking and their application is not allowed:

.1 Rotor made of aluminium or magnesium alloy and steel housing, irrespective of the radial air gap value,

.2 Housing made of aluminium or magnesium alloy and steel rotor, irrespective of the radial air gap,

.3 Any combination of rotor and housing made of steel with design radial gap lower than 13 mm.

5.4 Oil and Fuel Separators

5.4.1 General Requirements

5.4.1.1 The separator drums shall be dynamically balanced. The position of removable parts shall be reciprocally fixed and the separator shall be so designed as to exclude their wrong assembling.

5.4.1.2 The arrangement of the case and drum shall be such that the resonant speed of both the empty drum and the drum filled with liquid exceeds the nominal speed.

The resonant speed lower than the nominal one may be accepted, provided that in such case a long-time failure-free operation of the separator has been confirmed.

5.4.1.3 The construction of clutches shall preclude sparking and heating thereof in all running conditions and to enable removal of heat from the working surfaces.

5.4.2 Strength Calculations and Equipment of Separators

5.4.2.1 The strength of the separator rotating parts shall be checked by calculation for the rotational speed exceeding by at least 30% the nominal speed. The reduced stresses occurring in such conditions are not to exceed 0.95 of yield point of these parts.

5.4.2.2 The assembled prototype of separator shall be tested with oil by the manufacturer at rotational speed exceeding by 30% the nominal speed

5.4.2.3 Control devices of separating process and of the drum rotational speed shall be provided.

5.5 Supervision, Testing and Certificates

5.5.1 The following essential parts of piston type compressors and pumps are subject to supervision in the process of construction for compliance with the approved documentation:

– crankshafts[M)],
– connecting rods,
– pistons,
– piston bolts,
– cylinder blocks and cylinder covers,
– cylinder liners.

5.5.2 The following essential parts of centrifugal pumps, fans, air blowers and turboblowers are subject to supervision in the process of construction for compliance with the approved documentation:

– shafts,
– rotors,
– casings.

5.5.3 The following essential parts of fuel and oil separators are subject to supervision in the process of construction for compliance with the approved documentation:

– shafts,
– drum casing and disks,
– gear wheels.

Notes and explanations:

M) – material shall be PRS accepted.

6 DECK MACHINERY

6.1 General Requirements

6.1.1 Deck machinery shall be designed for service in conditions defined in 1.6 of *Part VI – Machinery Installations and Refrigerating Plants*.

6.1.2 The brake straps and their fastenings shall be resistant to sea water and petroleum products and to be heat resistant at temperatures up to 250 °C.

Heat resistance of the brake strap joint with the brake structure shall be appropriate for temperatures higher than the temperature which may occur in the joint in all possible working conditions of the mechanism.

6.1.3 The machinery having both manual and power drives shall be provided with interlocking arrangements preventing simultaneous operation of these drives.

6.1.4 It is recommended to make the deck machinery controls so that heaving-in is performed by turning the hand wheel clockwise or by moving the lever backwards, whereas veering-out is performed by turning the hand wheel anti-clockwise or by moving the lever forwards. Braking shall be performed by turning the hand wheel clockwise, whereas brake releasing – by rotating anti-clockwise.

6.1.5 Control and measuring instruments and gauges shall be so located as to be capable of being watched from the control station.

6.1.6 The machinery with hydraulic drive or control is also to comply with the requirements of Chapter 7.

6.1.7 Winch drums on which ropes are put in several layers and loaded shall have flanges protruding outside the external layer of winding by not less than 2.5 times the rope diameter.

6.2 Steering Gears and Their Installing on the Ship

6.2.1 General Requirements

6.2.1.1 The ship shall be provided with two steering gears: main and auxiliary complying with the requirements of 6.2.1.2, 6.2.1.3 and 6.2.1.5, respectively, unless specified otherwise (see 6.2.1.6).

The gears shall have independent effect on rudder stock, however they can have some common parts, such as tiller, quadrant, guide or cylinder block.

6.2.1.2 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.

[*)] For definition of main and auxiliary steering gear – see 1.2.3 of *Part III – Hull Equipment*.

The main steering gear may be hand driven if the rudder stock diameter does not exceed 120 mm (wihout ice strengthenings). In every other case the steering gear shall be driven by the power unit*).

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

6.2.1.3 The auxiliary steering gear*) shall enable putting the rudder by not less than 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 may be hand driven if the rudder stock diameter does not exceed 230 mm (without ice strengthenings). In every other case the steering gear shall be driven by the power unit.

6.2.1.4 The rated torque M_{ZN} of steering gear is the rudder stock torque at the following rudder angle:
35° – for main steering gear,
15° – for auxiliary steering gear,
at rated parameters of steering gear power units

6.2.1.5 The main steering gear and auxiliary steering gear shall be so designed that the following requirements are met:

.1 a failure in one of them will not render the other one inoperative;

.2 the auxiliary steering gear shall be capable of being brought into action quickly in case of failure of the main steering engine;

.3 the power driven main steering gear may not affect the steering wheel of the hand driven auxiliary steering gear.

6.2.1.6 Where the main steering gear is provided with at least two identical power units, the auxiliary steering gear need not be installed, provided that the main steering gear is so arranged that in case of a single failure of the pipelines or one of the power units, this failure could be separated so as to maintain or quickly restore the steering ability.

If the two power units are adapted for simultaneous work, then with the simultaneous operation of these units the steering gear may not be interlocked, this being proven by the structure analysis and tests.

6.2.1.7 Hydraulic steering gear with mechanical drive should be provided with:

.1 auxiliary manual pump permanently connected to hydraulic system directly after the oil circulation tank and fitted in an easy accessible place in the steering gear compartment. The steering gear with auxiliary manual drive shall enable putting the rudder to neutral position and by 10° to each side at reduced astern speed by not more than two persons;

.2 device for keeping the hydraulic oil clean adequate to type and design of the hydraulic system;

.3 low level alarm in each circulation oil tank. The visual and audible alarm signals shall be received in the main control position and Electromechanical Department control position;

.4 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 position in steering gear compartment;

6.2.1.8 Oil Seals

The 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.

6.2.1.9 Overload Protection

Each part of the power hydraulic system which may be separated from the system and subjected to load from drive source or from external forces (due to thrust on the rudder blade) shall be equipped with overflow valves set for the value not exceeding design pressure, however not less than 1.25 of the rated pressure in the system. The valve(s) passage capacity shall be not less than 1.1 times the total capacity of the pumps connected to the valve. In no case the pressure rise shall exceed 1.1 times the valve setting, taking into account change of oil viscosity in extremal conditions of the environment. The overflow valves shall be capable of being sealed.

The following overflow valves tests are recommended:

- capacity tests,
- water hammer strength tests.

6.2.1.10 Rudder position indicators

A rudder gear part rigidly coupled with the rudder stock (tiller, quadrant, etc.) shall be fitted with a dial to indicate the real position of the rudder in respect to ship's centre line.

6.2.1.11 Limit switches

Each steering gear shall be provided with device for breaking its action before the rudder came to limit switches, permanently fixed to the ship hull, having maintained the ability for immediate move of the rudder into the opposite direction.

6.2.1.12 Brakes

The steering gear shall be fitted with a brake or any other device ensuring 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).

In case of hydraulic steering gears, which can be kept steady by closing valves on oil lines, the special braking device may be omitted

6.2.1.13 Operating instructions

Operating instructions, including the block diagram and switching-over procedures for control systems, power units and hydraulic cylinders of steering gear, shall be posted permanently and at well visible places in the main control position and steering gear compartment

6.2.1.14 The requirements concerning the electric drives and signalling are defined in 5.5 of *Part VIII – Electrical Installations and Control Systems*, while the requirements for selection of steering gear for a given type of ship – in 2.6 of *Part III – Hull Equipment*.

6.2.2 Materials and Manufacturing of Hydraulic Systems

6.2.2.1 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 is not to exceed 650 MPa..

6.2.2.2 Hydraulic pipelines shall comply with effective requirements applicable to I class pipelines and flexible joints specified in 1.16.2 of *Part VI – Machinery Installations and Refrigerating Plants*.

6.2.2.3 Hydraulic pipelines shall be made so as to enable easy switching on and switching off individual hydraulic cylinders and assemblies, meeting at the same time the requirements of Chapter 7

A possibility of bleeding air from the pipelines shall be provided, where necessary.

6.2.2.4 The pumps of hydraulic steering gear shall be provided with protective devices preventing rotation of inoperative pump back or with an automatic arrangement shutting off the flow of liquid through the inoperative pump.

6.2.3 Construction and Strength Calculations

6.2.3.1 The steering devices should be so designed as to reduce as far as possible the local concentration of stress.

Welded parts and welding procedure are subject to PRS approval. All welded joints within hydraulic cylinders or interconnected parts situated in lines of the force flux, shall be, as a rule, made with full penetration of weld.

6.2.3.2 The parts of main and auxiliary steering gear situated in lines of the force flux shall be checked by calculations for strength when affected by loads corresponding to the design torque M_s (see 2.2.3 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 is not to 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.

6.2.3.3 The casings of steering gear actuators and hydraulic batteries shall comply with requirements for pressure vessels of class I specified in Chapter 8.

6.2.3.4 The stresses in considered part are not to 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 defined in Table 6.2.3.4.

Table 6.2.3.4

Factor	Steel	Cast steel	Nodular iron
A	3.5	4	5
B	1.7	2	3

PRS may demand fatigue strength calculations accounting for fatigue of materials caused by pressure pulsation in hydraulic system.

6.2.3.5 The parts of steering gear situated in lines of the force flux, not protected against overload by means of limiters fastened to the ship hull (see 2.6.2.2 of *Part III – Hull Equipment*) shall have strength not lower than that of the rudder stock.

6.2.4 Connection with Rudder Stock

6.2.4.1 The connection of the steering gear with parts rigidly coupled with the rudderstock shall be such as to eliminate the possibility of steering gear damage when the rudderstock is moved in axial direction.

6.2.4.2 The connection of the tiller, quadrant or yoke with the rudderstock shall be calculated for transmission of torque not less than $2M_s$ (see 2.2.3 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 $2M_s$.

6.2.5 Hand Operated Steering Gear

6.2.5.1 The main steering gear shall be of self-locking type. The auxiliary steering gear shall be also 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.

6.2.5.2 The main hand-operated steering gear shall meet the requirements of 6.2.1.2 – 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]).

6.2.5.3 The stand-by hand-operated steering gear shall meet the requirements of 6.2.1.3 when handled by not more than four men with a force not exceeding 150 N per helmsman, applied to steering gear handles.

6.2.5.4 For hand-operated main steering gear it is sufficient to provide the gear with buffer springs instead of protection against overload required by 6.2.1.9.

For the stand-by hand-operated steering gear the compliance with 6.2.1.9 is not compulsory.

6.3 Windlasses

6.3.1 Drive

6.3.1.1 The power of the windlass driving engine shall provide for a continuous heaving-in within 30 minutes of one chain cable with the anchor of normal holding force, with a mean speed at least 9 m/min (0.15 m/s) with a pull P in a chain on the sprocket not less than that determined by the formula:

$$P = 9.81ad^2, \quad [\text{N}] \tag{6.3.1.1}$$

a – factor equal to:
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 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 *a* factor may be reduced upon agreement with PRS;

Mean speed of the chain cable heaving-in shall be measured over 2 chain lengths, beginning when 3 chain lengths are freely hanging down.

6.3.1.2 The windlass drive shall provide the speed of hauling in the anchor to the hawse pipe not exceeding 0.15 m/s. It is recommended that this speed be not greater than 0.12 m/s.

6.3.1.3 To break the anchor from ground, the windlass driving gear at rated working cycle shall produce a continuous pull on one sprocket of at least 1.5 *P* during a time not less than 2 minutes; the requirement of 6.3.1.1 concerning the heaving-in speed does not need to be complied with.

6.3.2 Brakes and Clutches

6.3.2.1 The windlass shall be fitted with clutches arranged between the sprocket and its drive shaft.

The windlass with non-self-locking transmission gear shall be fitted with automatic brake fixing the shaft steady in case of power loss or driving system failure.

The automatic brake shall be capable of maintaining in the chain on sprocket wheel the pull not less than 1.3*P*.

6.3.2.2 Each chain sprocket shall be fitted with a brake ensuring effective and safe holding the running out chain. This brake shall ensure holding the chain cable without slip on the brake with the sprocket disengaged from the drive 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 is not to exceed 740 N.

6.3.3 Chain Sprockets

6.3.3.1 The chain sprockets shall have not less than five cams. For horizontal shaft sprockets, the rapping angle is not to be less than 115°, while for vertical shaft sprockets it is not to be less than 150°.

6.3.3.2 The chain sprockets shall be so designed that the detachable links (Kenter links) can pass through both in horizontal and vertical position.

6.3.4 Overload Protection

If the maximum torque of the windlass engine can generate the (reduced) stresses in the windlass components exceeding 0.95 times the yield point of the material used, or a rise to the force on the sprocket exceeding 0.5 times the test load, provision shall be made for a reliable protection clutch installed between the engine and the windlass to prevent such overloads.

6.3.5 Strength Calculation

Stresses of the windlass parts being in flux of the strain lines are not to exceed:

0.4 R_e – with the nominal power of driving motor,

0.95 R_e – at the maximum torque of driving motor,

0.95 R_e – with the maximum possible load of anchor chain held by brake – in accordance with 6.3.2.2; this applies to parts of the windlass being subject to the above load;

(R_e – the yield strength of material of the parts in question).

When designing the windlasses, attention should be paid to:

- notch stress concentration,
- dynamic loads caused by abrupt start or stop of driving motor,
- calculation methods and approximations applied for finding the stress value and cycle,
- reliable fastening the windlass to the foundation.

6.3.6 Additional Requirements for Windlasses with Remote Control

6.3.6.1 The 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.

6.3.6.2 The sprocket brake shall ensure smooth stopping of chain cable in time not exceeding 5 s and not less than 2 s from the moment of control station command.

6.3.6.3 The remote control station shall be fitted with indicator of released chain length and indicator of releasing speed – maximum permissible speed 3 m/s shall be marked on the indicator.

6.3.6.4 Remotely controlled windlasses shall be fitted with local manual control posts. In each case of remote control failure, the possibility of local control shall be maintained.

6.4 Mooring Winches

6.4.1 Drive

6.4.1.1 The motor of the mooring winch shall provide for an 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 heaving-in speed of the mooring line by mooring head at the rated load is not to exceed 0.3 m/s.

Provisions regarding the choice of rated pull are given in *Part III – Hull Equipment*.

6.4.1.2 The power transmission system of the mooring winch at the rated working cycle shall exert in the reeled first layer line a continuous pull not less than 1.5 of the rated pull within not less than 2 minutes.

The pull in the line designed for the work with mooring winch, induced by the maximum torque of the winch drive, is not to exceed 0.8 of the line breaking force.

6.4.1.3 An overload protection shall be provided if the maximum torque of the motor may bring about a load in the mooring winch components exceeding that specified in 6.4.3.

6.4.2 Brakes

6.4.2.1 The 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 at the power loss or drive failure.

6.4.2.2 The mooring winch drum shall be provided with a brake, the braking torque of which would prevent unreeling of the mooring line under the tension equal to 0.8 times the breaking load of the reeled first layer of rope.

The force applied to the brake handle to exert such torque is not to exceed 740 N.

Where the winch drum is fitted with a pawl and ratchet or other locking device, then the braking device shall be such that the winch drum can be released in a controlled manner while the mooring line is under tension.

6.4.3 Strength Calculation

6.4.3.1 The stresses exerted on parts attaching the mooring winch to the foundation and on load-bearing parts of the winch under mooring line breaking load exerted on the drum and on the mooring head in its mid-length, are not to exceed 0.95 times the yield strength of their material.

The stresses exerted on the winch parts shall be determined with all possible types and geometrical directions of loads likely to occur in the service conditions.

6.4.3.2 The strength characteristics of the line designed for the work with the mooring equipment shall be marked on the mechanism.

6.4.4 Additional Requirements for Mooring Winches with Automatic Control of the Pull Force

The mooring winches with automatic control of the pull force shall be provided with:

– indicator of the actual pull in the mooring line during the winch operation with automatic control of pull,

– device for automatic releasing the mooring line rendering tension which the winch can exert on the mooring line (with first layer reeled on), and which is not to exceed 1.5 times nor be less than 1.05 times the pre-set hauling tension.

6.4.5 Additional Requirements for Mooring Winches with Remote Control

6.4.5.1 The mooring winches with remote control shall be provided with alarm indicating in the remote control position that the permissible pull force has been exceeded. The alarm shall activate irrespective of the released rope length.

6.4.5.2 The mooring winches with remote control shall be capable of being controlled manually from the local posts. In each case of remote control system failure the possibility of local control shall be maintained.

6.5 Towing Winches

6.5.1 Where automatic devices are used for governing the tension of the towline, provision shall be made for checking the value of tension at every moment. The tension indicators shall be fitted at the towing winch and in the main control position.

6.5.2 Alarm system giving the warning signal when the maximum permissible length of the towline is veered out shall be provided.

6.5.3 The drums of towing winches shall comply with the requirements of 6.1.7 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 releasing the towline.

6.5.4 The design of the towing winch shall provide for quick release of the rope drum brake for a free veer of the towline.

6.5.5 The brakes of towing winches shall comply with the following requirements:

.1 The towing winch shall be provided with automatic braking device stopping 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 The rope drum shall be fitted with the brake capable to stop the drum, disengaged from drive, without slip, with the tension not less than the breaking load of the towline. Power operated drum brakes shall be also provided with a manual control system. The brake design shall provide for a quick release of the brake to ensure free heaving-in the towline.

6.5.6 The towline shall be fastened to the winch drum in such a way that in the case of full release of the towline, it is disconnected from the drum under the load equal to or slightly greater than the rated pull of the towing winch.

6.5.7 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 reduced stresses occurring in components, which may be subjected to acting forces caused by the above-mentioned loads, are not to exceed 0.95 of the component material yield point.

6.5.8 The strength characteristics of the towline designed for the work with the towing gear shall be marked on it.

6.6 Sweeping Winches of Sweep Equipment and Towing Winches of Sweeps

6.6.1 Where devices for governing the tension of the rope, are used provision shall be made for checking the value of tension at every moment. The tension indicators shall be fitted at the winch and in the main control position.

6.6.2 The winch shall be provided with the indicator of the veered rope length. Alarm system giving the warning signal when the maximum permissible length of the rope is veered out is also to be provided.

6.6.3 The winch shall be provided with drive remote control switch installed in the after part of the ship, which enables, after stopping rope drum, release or heaving-in sweeping unit.

6.6.4 The winch brakes shall comply with the following requirements:

.1 The winch shall be provided with automatic breaking devices holding the rope under tension of not less than 1.25 of the rated pull at the power decay or disengaging;

.2 The winch drum shall be provided with a brake, capable to stop the drum without slip and with the drum disengaged from the drive, when loaded with a force not less than the rope breaking load. Power operated drum brakes shall be also provided with a manual control system. The brake design shall provide for a quick stop and release of drum to ensure assembly or disassembly of the sweeping unit.

6.6.5 The rope shall be fastened to the winch drum in such a way that in the case of full release of the rope, it is disconnected from the drum under the load greater than 1.25 of the rated pull of the winch.

6.6.6 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 rope breaking load. The reduced stresses occurring in components, which may be subjected to acting forces caused by the above-mentioned loads, are not to exceed 0.95 of the component material yield point.

6.6.7 The strength characteristics of the rope designed for the work with the sweeping or towing gear shall be marked on it.

6.7 Towed Sonars Winches

6.7.1 Where devices are used for governing the tension of the cablerope, provision shall be made for checking the value of tension at every moment. The tension indicators shall be fitted at the winch local and remote control position as well as at the main control position.

6.7.2 The winch shall be provided with the indicator of the released cablerope length. Alarm system giving the warning signal when the maximum permissible length of the rope is veered out is also to be provided.

6.7.3 The drum shall be of such size which takes into account the cablerope reserve provided for the case of sonar being hooked around an underwater obstruction.

The winch shall be provided with alarm device giving the signal, in the local, remote and main control station, when the permissible pull has been exceeded.

6.7.4 The winch shall be adopted to the gantry - pulley block unit, designed for the applied cablerope.

6.7.5 The winch brakes shall comply with the following requirements:

1. The winch shall be provided with automatic breaking devices holding the cablerope at the power decay or disengaging;
2. The winch drum shall be provided with a brake, capable to stop the drum without slip and with the drum disengaged from the drive, when loaded with a force not less than 1.25 of the rated load. The brake design shall provide for a quick stop and release of cablerope drum.

6.7.6 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 cablerope breaking load. The reduced stresses occurring in components, which may be subjected to acting forces caused by the above-mentioned loads, are not to exceed 0.95 of the component material yield point.

6.7.7 The strength characteristics of the cablerope designed for the work with the towing gear shall be marked on it.

6.8 Supervision, Testing and Certificates

6.8.1 The following essential parts of the steering gears are subject to supervision in the process of construction for compliance with the approved documentation:

– tillers of main and auxiliary gear[M)],
– rudder quadrant[M)],
– rudderstock yoke[M)],
– pistons with pistons rods[M)],
– ram cylinders[M],
– drive shafts[M)],
– gear wheels, tooth rims[M)].

6.8.2 The following essential parts of the windlasses, mooring and towing winches are subject to supervision in the process of construction for compliance with the approved documentation:
– drive, intermediate and main drive shafts [M)],
– gear wheels, tooth rims,
– sprockets,
– claw clutches,
– brake bands.

Notes and explanations:
[M)] – material shall be PRS accepted.

6.8.3 The power units of hydraulic steering gears shall be type tested. The test time is not to be less than 100 hours. The test stand shall be arranged for idle running, as well as for operation with maximum capacity and under maximum working pressure. During the test the idle running periods should alternate the periods of operation under full load. Change from one operating condition to another one shall be effected as fast as during operation onboard ship. During the whole test no excessive heating, vibration or other irregularities of pump operation can occur. After the test, the pump shall be disassembled and its parts subjected to inspection.

The type test may be dispensed with for power units the reliability of which has been confirmed during ship's service.

6.8.4 The steering gears, after being installed onboard ship, shall be subjected to tightness and motion testing.

6.8.5 The scope of sea trials of steering gears shall include:

.1 checking compliance with the requirements given in 6.2.1.2 and 6.2.1.3 regarding the rudder deflection, by main and auxiliary steering gear. In case when the ship propeller is of variable pitch design, the propeller pitch shall be set as maximal for nominal engine rotation speed full ahead.

If trials are not possible with maximal ship draught, PRS may agree on other trial conditions;

.2 testing of steening gear power units 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 on system check;

.5 back up power supply as required in 5.5 *Part VIII – Electrical Installations and Control Systems*;

.6 control system operation, including control command transfer and local steering;

.7 checking communication means between main control position, Electromechanical Department control position and steering gear compartment;

.8 alarm system and indicators operation as required by Rules – subchapters 5.5 and 8.4 of *Part VIII – Electrical Installations and Control Systems*;

.9 checking where applicable if there is no hydraulic interlocking (hydraulic lock) in the gear.

7 HYDRAULIC DRIVES

7.1 Application

7.1.1 The requirements of this Chapter apply to all hydraulic devices and systems aboard the naval ship with the exception of those mentioned in p. 7.1.2.

7.1.2 The devices complying with recognised standards, independent and placed within own housing, which are not associated with ship's drive, control and manoeuvring do not need to comply with the requirements of this Chapter.

7.2 General Requirements

7.2.1 The hydraulic oil should not be a source of corrosion in the hydraulic system. Its ignition temperature shall be not less than 150 ºC. Hydraulic oil should be suitable for the whole working temperature range of the hydraulic device or system. It applies particularly to the changes of viscosity.

7.2.2 Hydraulic devices shall be protected with overflow valves. If there are no separate requirements in other parts of *the Rules*, the opening pressure of the overflow valve should not exceed 1.1 of the maximum working pressure.

The selected nominal flow rate of overflow valves should be calculated for the maximum pump output generating hydraulic oil pressure not exceeding 1.1 of the set opening pressure of the valve.

7.2.3 In case of the hydraulic systems and devices set for continuous operation, such as main hydraulic drives, steering gears, hydraulic clutches, provision shall be made for the arrangement ensuring cleaning oil filters without disengaging the system.

7.2.4 Damage to the hydraulic system is not to induce any failure in the related machinery.

7.2.5 Hydraulic systems of steering gears, as well as hydraulic systems actuating controllable pitch propellers can not be connected in any way with other hydraulic systems.

7.2.6 Where the hydraulic tubing of windlasses is connected to other hydraulic system tubing, the oil supply shall be provided by two separate pump units, each of them ensuring the windlass operation having the requirements of 6.3.1 fulfilled.

7.2.7 The hydraulic cylinders shall be so installed and connected to the frame as to preclude effect of external bending moments on the cylinder rod.

7.3 Flammable Hydraulic Oil Tanks

Flammable hydraulic oil tanks shall comply with the same requirements as fuel tanks, with the following exceptions:

.1 in case of tanks not adjacent to ship's hull plating and situated outside machinery compartments of category A, the application of cylindrical level indicator glasses is allowed in compartments situated above summer load waterline, where there are no ignition sources, like diesel engines or boilers;

.2 in case of tanks having capacity below 100 dm^3, situated in machinery compartments of category A, PRS may consider approval of cylindrical level indicator glasses.

7.4 Pipe Joints

Pipe joints shall comply with the requirements of 1.16, *Part VI – Machinery Installations and Refrigerating Plants* and additionally:

.1 pipes mounted onboard a naval ship shall have the internal surface sufficiently clean as it is required for hydraulic components;

.2 in case of pipes of diameter less than 50 mm, the use of threaded pipe joints approved by PRS is accepted, however pipe joints with rubber ring seals can be used only for connection of hydraulic components and not for connecting separate tube segments;

.3 non-approved pipe joints may be used, with PRS acceptance only, in case when they comply with related national standard and are granted appropriate acceptance certificate;

.4 no soldered joints are allowed in pipelines;

.5 flexible hoses with connection fittings shall comply with the requirements of 1.16.2.9 of the *Part VI – Machinery Installations and Refrigerating Plants* and shall be PRS type approved. Non approved fireproof hoses can be used with PRS acceptance except in installations of steering gears and drive systems of watertight doors, as well as ports and ramps fitted in ship's hull, provided they comply with related national standard and are granted appropriate approval certificate.

7.5 Hydraulic Components

7.5.1 Hydraulic accumulators shall comply with the strength requirements for pressure vessels of appropriate class. Each accumulator, which could be isolated of the hydraulic system shall be equipped with own overflow valve. There shall be a safety valve or other safety means on the gas side of the accumulator to prevent the excessive pressure rise.

7.5.2 The hydraulic cylinders shall comply with the strength requirements defined for pressure vessels of appropriate class.

7.5.3 The hydraulic cylinders shall be of type approved by PRS.

The non-approved cylinders can be used with PRS acceptance in case when they comply with related national standard and are granted appropriate approval certificate.

7.5.4 Valves, pumps, hydraulic motors and high pressure filters shall be PRS type approved.

7.5.5 Hydraulic cylinders which do not comply with the requirements of 7.5.3 and hydraulic components which do not comply with the requirements of 7.5.4 can be used only in the case they have been manufactured under PRS supervision on the basis of an approved documentation and have been accepted by surveyor to PRS at manufacturer's premises according to the approved test programme.

7.6 Drives of Lifting Appliances

7.6.1 Lifting appliances shall be provided with:

.1 central hydraulic system with two independent pumping units, each being capable of driving simultaneously all lifting appliances; or

.2 an independent hydraulic system for each lifting appliance, provided with a pumping unit capable of driving the appliance.

If the system specified in .1 is separate of all other ship's hydraulic systems, the second pumping unit need not be applied.

7.6.2 The local control positions of lifting appliances shall be provided with:

.1 controls;

.2 control and measuring devices of driving machinery;

.3 signal alarm activated during cargo handling operations. In regions where the noise exceeds the normal level, PRS may additionally require installation of visual alarm;

.4 instruments indicating the cargo movement direction.

The requirements of .3 and .4 do not apply to control positions from which the cargo movement can be observed.

7.6.3 The possibility of lifting appliance being started by means other than shifting the control instrument from zero point shall be precluded, eg. by engaging the pumping units.

7.6.4 Devices, which enable lowering cargo with constant speed, even when the cargo reaches its final position, shall be provided. Operation of these devices shall not cause disengaging the pumping units.

7.6.5 The drives shall be provided with devices which enable safe lowering and positioning suspended cargo in case of abrupt pressure drop of hydraulic oil in the hydraulic system

Hydraulic cylinders shall be fitted with safety devices.

7.7 Drives and Interlocking Devices of Ramps, Ports and Raised Decks

7.7.1 It is recommended to use hydraulic drives provided with two independent pumping units.

7.7.2 Reliable mechanical devices for interlocking the ramps, ports and raised decks in an open position shall be provided.

7.7.3 The drives of ramps, ports and raised decks shall also comply with the requirements specified in 7.6.2 to 7.6.5.

7.8 Supervision, Testing and Certificates

7.8.1 The following essential parts of hydraulic elements (see 7.5) are subject to supervision in the process of construction for compliance with the approved documentation:
– shafts and screw rotors,
– rods,
– pistons,
– casings, cylinders, screw pump bodies,
– gear wheels.

7.8.2 Tests of hydraulic driving units shall be conducted according to a test programme approved by PRS.

Test programme shall define the type and the scope of the tests, acceptance criteria, tests place and, when necessary, testing methods.

7.8.3 The tests shall include:

.1 pressure testing of the pipelines according to the requirements of 1.5.4 of the *Part VI – Machinery Installations and Refrigerating Plants*;

.2 checking pipelines cleanness after rinsing;

.3 running tests;

.4 hydraulic oil purity check before and after running tests.

8 BOILERS, PRESSURE VESSELS AND HEAT EXCHANGERS

8.1 General Requirements

Depending on the design and parameters, boilers, pressure vessels and heat exchangers are subdivided into classes as indicated in Table 8.1.

Table 8.1

Equipment	Class I	Class II	Class III
Steam boilers, including utilization boilers, water boilers for water temperature over 115°C, steam superheaters and steam receivers, thermal liquid heaters	$p > 0.35$	$p \le 0.35$	–
Steam-heated steam generators	$p > 1.6$	$p \le 1.6$	–
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, [MPa];
t – wall design temperature, [°C];
s – wall thickness, [mm].

8.2 Strength Calculations

8.2.1 General Requirements

8.2.1.1 The wall thicknesses obtained by calculation are the lowest permissible values under normal operating conditions. The formulae and methods of strength calculation do not take into account the manufacturer's tolerances for thickness and these shall be added as special allowances to the design thickness values.

At the request of PRS, additional stresses due to external loads (axial forces, bending moments, torques) exerted on the calculated parts (particularly loads due to dead mass, the mass of attached parts, etc.) shall be taken into account separately.

8.2.1.2 The dimensions of structural components of boilers, pressure vessels and heat exchangers for which no strength calculation methods are given in the present Part of the *Rules*, shall be determined on the basis of experimental data and approved theoretical calculations, and are subject to special consideration by PRS in each particular case.

8.2.2 Design Pressure

8.2.2.1 The hydrostatic pressure shall be taken into account in design pressure calculations when it exceeds 0.05 MPa.

8.2.2.2 For uniflow and forced-circulation boilers, the design pressure shall be determined with due consideration for the hydrodynamic resistances in boiler components at the boiler rated capacity.

8.2.2.3 For flat walls subjected to pressure from both sides, the design pressure shall be taken equal to the highest of the acting pressures. Walls of curved surfaces subjected to pressure from both sides shall be calculated for the highest outer and inner pressures. Where the pressure on one side of the wall of flat or curved surface is below the atmospheric pressure, the design pressure shall be taken equal to the highest pressure on the other side of the wall increased by 0.1 MPa

8.2.2.4 The design pressure for economizers shall be taken equal to the sum of the working pressure in the boiler steam drum and the hydrodynamic resistances in the economizer, tubing, mountings and fittings at boiler rated capacity.

8.2.3 Design Temperature

8.2.3.1 For the purpose of determining the allowable stresses depending on the temperature of the medium and heating conditions, the design wall temperature shall be taken not lower than indicated in Table 8.2.3.1

Table 8.2.3.1

Item	Components of boilers, pressure vessels and heat exchangers and their operating conditions	Design temperature of the wall
1	Components exposed to radiant heat	
1.1	Boiler tubes	T_m + 50 °C
1.2	Economizer tubes	T_m + 50 °C
1.3	Corrugated furnaces	T_m + 75 °C
1.4	Plain furnaces, headers, chambers, combustion chambers	T_m + 90 °C
2	Components exposed to hot gases, protected from radiant heat[1)]	
2.1	Ring segments, ends, headers, chambers, tube plates and tubes	T_m + 30 °C
2.2	Headers and tubes of steam superheaters at steam temperature up to 400 °C	T_m + 35 °C
2.3	Headers and tubes of steam superheaters at steam temperature above 400 °C	T_m + 50 °C
2.4	Utilization boilers with mechanical cleaning of heated surface	T_m + 30 °C
2.5	Utilization boilers with burner for burning out the contamination of heated surface	T_v
3	Components heated with steam or liquid	T_v
4	Non-heated components[2)]	T_m

Notes to Table 8.2.3.1:

[1)] – see 8.2.3.4;

[2)] – see 8.2.3.3;

T_m – maximum temperature of heated medium, [°C];

T_v – maximum temperature of heating medium, [°C].

8.2.3.2 The design temperature for steam superheater tubes working at steam temperatures over 400 °C, as well as for tubes and headers of superheaters exposed to radiant heat shall be determined by calculation and is subject to special consideration by PRS in each particular case.

8.2.3.3 The wall is considered to be non-heated where one of the following conditions applies:
– the wall is separated from the combustion space or uptake by fire-resistant insulation, the distance between wall and insulation being 300 mm or more;
– the wall is protected with fire-resistant insulation not exposed to radiant heat.

8.2.3.4 The wall is considered to be protected from radiant heat effect where one of the following conditions applies:
– it is protected with fire-resistant insulation;
– it is protected by a closely spaced row of tubes (with a maximum clearance between the tubes in the row not greater than 3 mm);
– it is protected by two staggered rows of tubes with a longitudinal pitch equal to the maximum of two outside tube diameters or by three or more staggered rows of tubes with a longitudinal pitch equal to the maximum of two and a half outside tube diameters.

8.2.3.5 The design temperature of heated boiler walls and non-heated steam space walls of boilers shall be assumed not less than 250 °C.

8.2.3.6 Non-insulated boiler walls, exceeding 20 mm in thickness, heated by hot gas, may be used only at gas temperature up to 800 °C. If, with wall thickness of less than 20 mm and hot gas temperature exceeding 800 °C, there are areas unprotected by insulation or by tube rows, exceeding in length 8 tube diameters, the design wall temperature shall be determined by thermal stress analysis.

For wall protection from radiant heat – see 9.1.8.

8.2.3.7 The design temperature for walls of tanks and heat exchangers operating under coolant pressure shall be taken equal to 20 °C, if higher temperatures are not likely to occur.

8.2.4 Strength Characteristics of Materials and Allowable Stresses

8.2.4.1 The strength of steels with $(R_e/R_m) \leq 0.6$ shall be assumed equal to physical yield point or proof stress R_e^t or $R_{0.2}^t$, as well as an 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 be also taken into account

For steel loaded in the creep conditions (temperature exceeding 450 °C), irrespective of (R_e/R_m) ratio, the average creep strength $R_{1/100\,000/t}$ with 1% permanent elongation, after 100 000 h, at design temperature t, shall be taken into account

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 the calculations.

8.2.4.2 For materials the stress-strain curve of which does not show a specified yield stress, the value taken for calculations shall be tensile strength at the design temperature.

8.2.4.3 For cast iron and non-ferrous metal alloys, the minimum value of ultimate tensile strength in normal temperature shall be taken for calculations.

8.2.4.4 When using non-ferrous metals and their alloys, it shall be taken into account that their heating during processing and welding tends to relieve them of the strengthening effects realized under cold conditions. Therefore the strength characteristics to be used for strength calculations of components and assemblies manufactured of such materials and alloys shall be those applied to their annealed condition.

8.2.4.5 Allowable stresses σ assumed for strength calculations shall be determined as the lowest of the following three values:

$$\sigma = \frac{R_m^t}{\eta_m}, \quad \sigma = \frac{R_e^t}{\eta_e} \quad \text{or} \quad \sigma = \frac{R_{0.2}^t}{\eta_e}$$

$$\sigma = \frac{R_{z/100000/t}}{\eta_z}, \quad \sigma = \frac{R_{1/100000/t}}{\eta_p}$$

where:

η_m – safety factor for tensile strength R_m^t

η_z – safety factor for creep strength $R_{z/1000000/t}$

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 8.2.5.

8.2.5 Safety Factors

8.2.5.1 For components made of forged steel pieces or rolled steel, exposed to inner pressure, the safety factors are not to be less than:

$$\eta_e = \eta_z = 1.6; \ \ \eta_m = 2.7 \ \text{ i } \ \eta_p = 1.0$$

For components exposed to outer pressure, the safety factors η_e, η_z and η_m shall be increased by 20%.

8.2.5.2 For components of boilers, heat exchangers and pressure vessels of classes II and III, made of steels with $(R_e/R_m) \le 0.6$, the safety factors my be reduced, but they cannot be less than:

$$\eta_e = \eta_z = 1.5; \ \ \eta_m = 2.6$$

8.2.5.3 For components of boilers, heat exchangers and pressure vessels made of cast steel and exposed to inner pressure, the safety factors are not to be less than:

$$\eta_e = \eta_z = 2.2; \ \ \eta_m = 3.0 \ \text{ i } \ \eta_p = 1.0$$

For components exposed to outer pressure, the safety factors shall be increased by 20% (η_z remains unchanged).

8.2.5.4 Safety factors η_e and η_z for thermal loaded essential parts of boilers shall be taken not less than:
3.0 – for corrugated furnaces;
2.5 – for plain furnaces, combustion chambers, stay combustion tubes, as well as long and short stays;
2.2 – for gas uptake branch pipes subjected to pressure and other similar exhaust gas heated walls.

8.2.5.5 Safety factor η_m for components made of cast iron shall be taken not less than 4.8 – for inner and outer pressure.

This factor for non-ferrous metals – shall be not less than 4.6 for inner pressure and not less than 5.5 for outer pressure. For the conical walls, in the latter case, η_m shall be taken not less than 6.0.

8.2.6 Efficiency Factors

8.2.6.1 Efficiency factors of weld seams φ shall be taken from Table 8.2.6.1-1 depending on the type of weld joint and method of welding. For particular classes of boilers, pressure vessels and heat exchangers (see Table 8.1), the value of efficiency factor φ shall be not less than that determined in Table 8.2.6.1-2.

Table 8.2.6.1-1

Welding	Type of joint	Type of weld	φ
Automatic	Butt joints	Double-sided Single-sided on backing strip Single-sided without backing strip	1.0 0.9 0.8
	Overlapping joints	Double-sided Single-sided	0.8 0.7
Semi-automatic and manual	Butt joints	Double-sided Single-sided on backing strip Single-sided without backing strip	0.9 0.8 0.7
	Overlapping joints	Double-sided Single-sided	0.7 0.6

Notes to Table 8.2.6.1-1:
1. The full penetration shall be done in each case.
2. For welded joints made with electroslag method, $\varphi = 1$ shall be taken.

Table 8.2.6.1-2

Equipment	Factor φ		
	Class I	Class II	Class III
Boilers, steam superheaters and reservoirs	0.9	0.8	–
Steam-heated steam generators	0.9	0.8	–
Pressure vessels and heat exchangers	0.9	0.7	0.6

8.2.6.2 Ligament efficiency factor of cylindrical walls weakened by holes identical in diameter shall be taken equal to the lowest of the following three values:

.1 the ligament efficiency factor of cylindrical walls weakened by a longitudinal row or a field of equally-pitched holes (Fig. 8.2.6.2-1), as determined from the formula:

$$\varphi = \frac{a-d}{a} \qquad (8.2.6.2.1)$$

.2 the ligament efficiency factor, reduced to the longitudinal direction, of cylindrical walls weakened by a transverse row or a field of equally-pitched holes (Fig. 8.2.6.2-1), as determined from the formula:

$$\varphi = 2\frac{a_1-d}{a_1} \qquad (8.2.6.2.2)$$

.3 the ligament efficiency factor, reduced to the longitudinal direction, of cylindrical walls weakened by a field of equally-pitched staggered holes (Figs. 8.2.6.2-2 and 8.2.6.2-3), as determined from the formula:

$$\varphi = k\frac{a_2-d}{a_2} \qquad (8.2.6.2.3\text{-}1)$$

where:

φ – ligament efficiency factor of walls weakened by holes;

d – diameter of the hole for expanded tubes or inner diameter of welded-on tubes and drawn nozzles, [mm];

a – pitch between two adjacent hole centres arranged along the wall, [mm];

a_1 – spacing between two adjacent hole centres in the transverse (or circumferential) direction, taken as mean circumference arc, [mm];

a_2 – spacing between two adjacent hole centres in the staggered rows, [mm], determined from the formula:

$$a_2 = \sqrt{l^2 + l_1^2}\,, \quad \text{[mm]} \qquad (8.2.6.2.3\text{-}2)$$

l – centre-to-centre distance between two adjacent holes in the longitudinal direction (see Figs. 8.2.6.2-2 and 8.2.6.2-3), [mm];

l_1 – centre-to-centre distance between two adjacent holes in the transverse or circumferential direction (see Figs. 8.2.6.2-2 and 8.2.6.2-3), [mm];

k – factor depending on the ratio l_1/l taken from Table 8.2.6.2.3.

Table 8.2.6.2.3

$\frac{l_1}{l}$	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 k shall be determined by interpolation

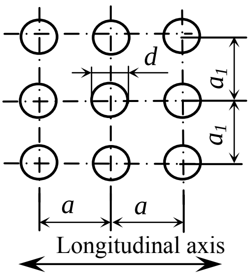


Fig. 8.2.6.2-1

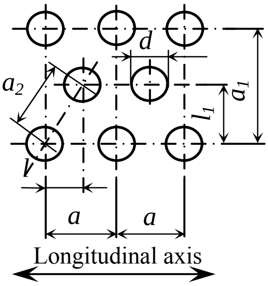


Fig. 8.2.6.2-2

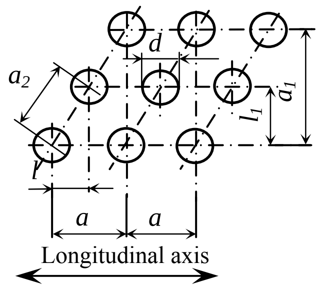


Fig. 8.2.6.2-3

8.2.6.3 Where rows or fields of equally pitched holes contain holes of different diameters, the value d in the formulae for ligament efficiency factor determination (8.2.6.2.1, 8.2.6.2.2, 8.2.6.2.3-1, 8.2.6.2.3-2) shall be assumed as a value equal to the arithmetic mean of two largest adjacent holes diameters. In the case of unequal pitch between holes of equal diameters, the formulae for ligament efficiency factor determination shall be used with the lowest values of a, a_1 or a_2, respectively.

8.2.6.4 Where welded seams have holes, the ligament efficiency factor shall be taken as equal to the product of the ligament efficiency factor of the joint and of the wall weakened by holes.

8.2.6.5 For cylindrical component walls not weakened by welded seam or row/field of holes, the ligament efficiency factor shall be taken as equal to 1.0. In no case is the factor to be taken higher than 1.0.

8.2.6.6 The ligament efficiency factor of walls weakened by holes for expanded tubes, as determined from formulae 8.2.6.2.1, 8.2.6.2.2, 8.2.6.2.3, shall be taken not less than 0.3. Calculations where the value of this factor is less will be specially considered by PRS in each particular case.

8.2.6.7 Where walls of cylindrical components shall be made of sheets of different thickness, interconnected by longitudinal welded joint, then the thickness calculation shall be made separately for each sheet, taking existing weakenings into account.

8.2.6.8 For welded tubes with longitudinal weld seam, the joint factor will be specially considered by PRS in each particular case.

8.2.6.9 Methods of determination of the ligament efficiency factor for walls weakened by openings, to be fully or partially reinforced. are given in 8.2.19.

8.2.6.10 The ligament efficiency factor of flat tube plates shall be determined for tangential and radial pitches from formula 8.2.6.2.1. The less of thus obtained values shall be used for calculating the tube plate thickness.

8.2.7 Design Thickness Allowances

8.2.7.1 In all cases where the design wall thickness allowance, *c*, is not expressly specified, it shall be taken as equal to at least 1 mm. For steel walls over 30 mm in thickness, as well as for walls manufactured of corrosion-resistant non-ferrous or high alloy materials, and for materials adequately protected against corrosion, for instance, by plating or plastic lining, on agreement with PRS, allowance *c* need not be provided for the design thickness value.

8.2.7.2 For pressure vessels and heat exchangers inaccessible for internal inspection, and these with walls heavily affected by corrosion or wear – PRS may demand increase of allowance *c*.

8.2.8 Cylindrical and Spherical Elements and Tubes Subjected to Internal Pressure

8.2.8.1 The requirements given below cover the following conditions:

$\frac{D_a}{D} \le 1.6$ – for cylindrical elements,

$\frac{D_a}{D} \le 1.7$ – for tubes,

$\frac{D_a}{D} \le 1.2$ – for spherical components.

Cylindrical elements with $D_a \le 200$ mm shall be considered as tubes.
For D_a , D – see 8.2.8.2.

8.2.8.2 The thickness of cylindrical elements and tubes is not to be less than that calculated from the formula:

$$s = \frac{D_a p}{2\sigma\varphi + p} + c \,, \quad [\text{mm}] \qquad (8.2.8.2\text{-}1)$$

or

$$s = \frac{Dp}{2\sigma\varphi - p} + c \,, \quad [\text{mm}] \qquad (8.2.8.2\text{-}2)$$

s – wall thickness, [mm];

p – design pressure, [MPa];
D_a – outer diameter, [mm];
D – inner diameter, [mm];
φ – ligament efficiency factor (see 8.2.6);
σ – allowable stress (see 8.2.4.5), [MPa];
c – design thickness allowance (see 8.2.7), [mm].

8.2.8.3 Spherical wall thickness is not to be less than those obtained from the formula:

$$s = \frac{D_a p}{4\sigma\varphi + p} + c, \quad [\text{mm}] \qquad (8.2.8.3\text{-}1)$$

or

$$s = \frac{Dp}{4\sigma\varphi - p} + c, \quad [\text{mm}] \qquad (8.2.8.3\text{-}2)$$

The symbols used are the same as in 8.2.8.2.

8.2.8.4 Irrespective of the values obtained from formulae 8.2.8.2-1, 8.2.8.2-2, 8.2.8.3-1 and 8.2.8.3-2, the wall thickness of spherical and cylindrical elements as well as tubes is not to 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 those shown in Table 8.2.8.4 – for tubes.

The thickness of tube walls heated by gas of temperature exceeding 800 °C is not to be greater than 6 mm.

Table 8.2.8.4

Outside tube diameter, [mm]	≤ 20	>20 ≤30	>30 ≤38	>38 ≤51	>51 ≤70	>70 ≤95	>95 ≤102	>102 ≤121	>121 ≤152	>152 ≤191	>191
Minimum tube 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:
Decrease in wall thickness due to expanding or bending shall be compensated by allowances.

8.2.8.5 The minimum thickness of walls of pipes made of non-ferrous alloys and corrosion-resistant steel may be less than those specified in 8.2.8.4, when agreed upon with PRS, but not less than those determined from formulae 8.2.8.2 and 8.2.8.3.

8.2.9 Elements Subjected to External Pressure

8.2.9.1 The requirements specified below refer to walls of cylindrical elements with:

$$\frac{D_a}{D} \le 1.2$$

The values of the thickness for tubes with $D_a \leq 200$ mm shall be determined in accordance with 8.2.8.2.

8.2.9.2 Plain walls of cylindrical elements, with or without stiffening members, including plain furnaces of boilers shall have a thickness not less than that determined by the formula:

$$s = \frac{50\left(B + \sqrt{B^2 + 0.04AC}\right)}{A} + c, \quad [\text{mm}] \qquad (8.2.9.2\text{-}1)$$

where:

$$A = 200\frac{\sigma}{D_m}\left(1 + \frac{D_m}{10l}\right)\left(1 + \frac{5D_m}{l}\right) \qquad (8.2.9.2\text{-}2)$$

$$B = p\left(1 + \frac{5D_m}{l}\right) \qquad (8.2.9.2\text{-}3)$$

$$C = 0.045 \cdot p \cdot D_m \qquad (8.2.9.2\text{-}4)$$

s – wall thickness, [mm];
p – design pressure (see 8.2.2), [MPa];
D_m – mean diameter, [mm];
σ – allowable stress (see 8.2.4.5), [MPa];
c – design thickness allowance (see 8.2.7), [mm];
l – design length of cylindrical portion between stiffening members, [mm].

End plates, furnace connections to end plates and combustion chamber as well as reinforcing rings (Fig. 8.2.9.2) or similar structures may be assumed as stiffening members.

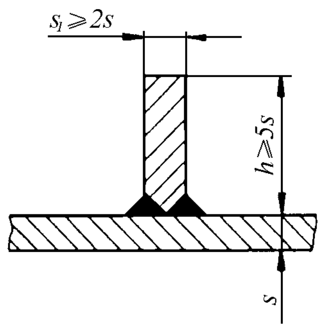


Fig. 8.2.9.2

8.2.9.3 Corrugated furnaces shall have a thickness not less than that determined by the formula:

$$s = \frac{pD}{2\sigma} + c \qquad (8.2.9.3)$$

s – wall thickness, [mm];
D – minimum inner diameter of the furnace corrugated portion, [mm];

p – design pressure (see 8.2.2), [MPa];
σ – allowable stress (see 8.2.4.5), [MPa];
c – design thickness allowance (see 8.2.7), [mm].

8.2.9.4 Where the length of the straight portion of a corrugated furnace from the front-end wall to the commencement of the first corrugation exceeds the corrugation length, the wall thickness over this portion shall be not less than that calculated in accordance with formula 8.2.9.2-1.

8.2.9.5 The thickness of plain furnaces is not to be less than 7 mm nor more than 20 mm. The thickness of corrugated furnaces is not to be less than 10 mm and not more than 20 mm.

8.2.9.6 Plain furnaces up to 1400 mm in length need not be fitted with reinforcing rings. Where a boiler has two or more furnaces, the reinforcing rings of adjacent furnaces should be arranged in alternate planes.

8.2.9.7 Holes and openings in walls of cylindrical and spherical elements shall be compensated for according to requirements of 8.2.19.

8.2.9.8 Thickness s_1 of the vertically loaded ring formed by connection of combustion chamber with vertical boiler shell (see Fig. 8.2.9.8), is not to be less than that determined by the formula:

$$s_1 = \frac{3.7p}{\sigma}\sqrt{D_1(D_1 - D_0)} + 1, \quad \text{[mm]} \qquad (8.2.9.8)$$

p – design pressure, [MPa].

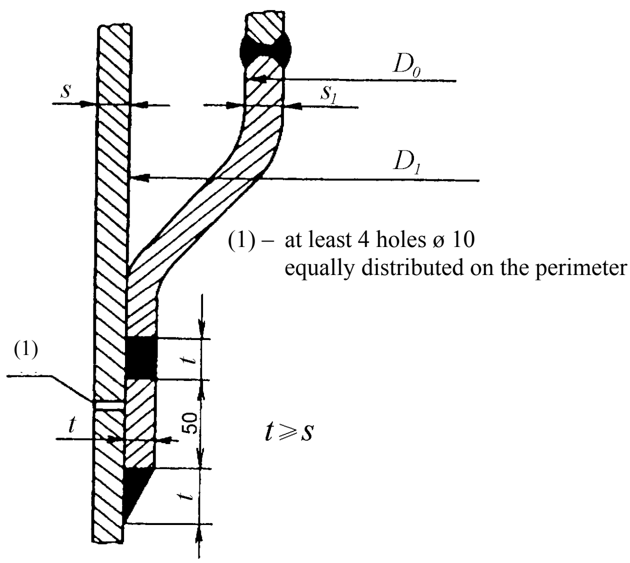


Fig. 8.2.9.8

8.2.10 Conical Elements

8.2.10.1 The wall thickness of conical elements subjected to internal pressure is not to be less than:

.1 at α ≤ 70° – the greater of the values derived from the formula:

$$s = \frac{D_a p y}{4\sigma\varphi} + c\,, \quad \text{[mm]} \qquad (8.2.10.1.1\text{-}1)$$

and

$$s = \frac{D_a p y}{(4\sigma\varphi - p)\cos\alpha} + c\,, \quad \text{[mm]} \qquad (8.2.10.1.1\text{-}2)$$

.2 at α >70° – the value derived from the formula:

$$s = 0.3\left[D_a - (r + s)\right]\sqrt{\frac{p}{\sigma\varphi}\cdot\frac{\alpha}{90^\circ}} + c\,, \quad \text{[mm]} \qquad (8.2.10.1.2)$$

s – wall thickness, [mm];
D_c – design diameter (Figs. 8.2.10.1.2-1 through 8.2.10.1.2-4), [mm];
D_a – outer diameter (Figs. 8.2.10.1.2-1 through 8.2.10.1.2-4), [mm];
p – design pressure (see 8.2.2), [MPa];
y – shape factor (see Table 8.2.10.1);
α, α_1, α_2, α_3 – angles (Figs. 8.2.10.1.2-1 ÷ 8.2.10.1.2-4), [°];
σ – allowable stress (see 8.2.4.5), [MPa];
φ – efficiency factor (see 8.2.6). In formulae 8.2.10.1.1-1 and 8.2.10.1.2 the efficiency factor of a circumferential weld joint shall be used, while in formula 8.2.10.1.1-2 – of longitudinal weld joint. For seamless shells, as well as in situations where circumferential weld is spaced from the edge to a distance exceeding:

$0.5\sqrt{\dfrac{D_a s}{\cos\alpha}}$ it shall be assumed $\varphi = 1$;

r – radius of edge curvature (Figs. 8.2.10.1.2-1, 8.2.10.1.2-2 and 8.2.10.1.2-4), [mm];
c – design thickness allowance (see 8.2.7), [mm].

Table 8.2.10.1

α, [degs]	Shape factor y at various 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
1	2	3	4	5	6	7	8	9	10	11	12	13
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 fillet joints where the weld forms the edge of two elements (see Fig. 8.2.10.1.2-3), the shape factor y shall be determined for $r / D_a = 0.01$.

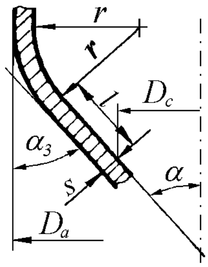


Fig. 8.2.10.1.2-1

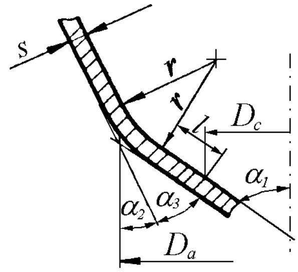


Fig. 8.2.10.1.2-2

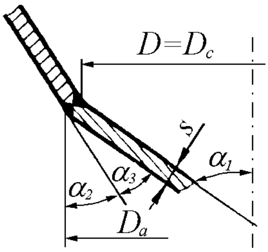


Fig. 8.2.10.1.2-3

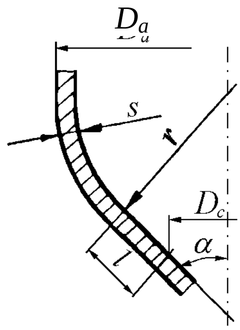


Fig. 8.2.10.1.2-4

l – distance from the edge of the wide end of conical shell, parallel to the generatrix, assumed equal to a tenfold thickness, but not larger than half the length of the conical shell generatrix (Figs. 8.2.10.1.2-1, 8.2.10.1.2-2 and 8.2.10.1.2-4), [mm];

8.2.10.2 The wall thickness of conical elements subjected to external pressure shall be determined according to 8.2.10.1, provided the following conditions are fulfilled:

.1 efficiency factor of a welded joint φ shall be equal to 1;

.2 allowance c shall be taken equal to 2 mm;

.3 design diameter D_c shall be determined by the formula:

$$D_c = \frac{d_1 + d_2}{2\cos\alpha}, \quad [\text{mm}] \qquad (8.2.10.2.3)$$

d_1, d_2 – the largest and the smallest internal diameter of the cone, respectively, [mm];

.4 when $\alpha < 45°$, it shall be proved that the walls are not subject to yield.

The pressure p_1, at which yield occurs, shall be determined by the formula:

$$p_1 = 26E10^{-6}\frac{D_c}{l_1}\left[\frac{100(s-c)}{D_c}\right]^2\sqrt{\frac{100(s-c)}{D_c}}, \quad \text{[MPa]} \qquad (8.2.10.2.4)$$

E – modulus of elasticity, [MPa];
l_1 – the maximum length of the cone or spacing between its supports, [mm].

Fulfilling the inequality $p_1 > p$ (p – design pressure, [MPa]) is the condition of the yield absence on the cone walls.

8.2.10.3 Fillet weld joints (see Fig. 8.2.10.1.2-3) are allowed only for angle values $\alpha_3 \le 30°$ and wall thickness $s \le 20$ mm. The joint shall be welded at both sides. For conical ring segments with $\alpha \ge 70°$, the joints may be welded without edge preparation if the requirements specified in 8.2.10.2 are duly observed. It is not recommended to apply fillet joints to boilers.

8.2.10.4 The areas around holes and openings in conical walls shall be strengthened according to the requirements given in 8.2.19.

8.2.11 Flat End Plates and Covers

8.2.11.1 The thickness of the flat end plates unsupported by stays, as well as of welded or bolted covers (Figs. 8.2.11.1-1 to 8.2.11.1-8 and item 1.2 in Annex 1) is not to be less than that determined by the formula:

$$s = KD_c\sqrt{\frac{p}{\sigma}} + c, \quad \text{[mm]} \qquad (8.2.11.1\text{-}1)$$

s – wall thickness , [mm];
K – design factor for structures shown in Figs. 8.2.11.1-1 ÷ 8.2.11.1-8 and items 1.1 ÷ 1.6 in Annex 1);
D_c – design diameter (Figs. 8.2.11.1-2 ÷ 8.2.11.1-7 and item 1.2 in Annex 1), [mm]. For end plates such as shown in Fig. 8.2.11.1-1 and item 1.1 in Annex 1, the design diameter shall be:

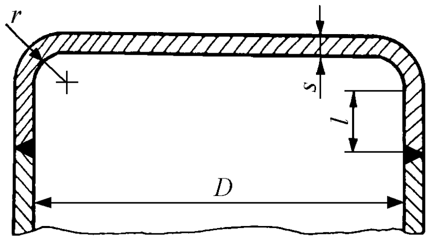
$$D_c = D - r, \quad \text{[mm]} \qquad (8.2.11.1\text{-}2)$$

For rectangular or oval covers, the design diameter shall be:

$$D_c = m\sqrt{\frac{2}{1+\left(\frac{m}{n}\right)^2}}, \quad \text{[mm]} \qquad (8.2.11.1\text{-}3)$$

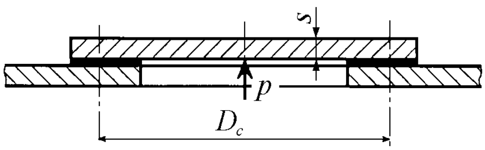
D_b – pitch circle diameter of bolts (Fig. 8.2.11.1-6), [mm];
D – inner diameter, [mm];
n and m – the maximum and minimum length of the axis or the side of the hole respectively, measured to the dividing axis of the packing (Fig. 8.2.11.1-8), [mm];
r – inner curvature radius of the dished end plate, [mm];
p – design pressure (see 8.2.2), [MPa];

σ – allowable stress (see 8.2.4.5), [MPa];
c – design thickness allowance (see 8.2.7), [mm];
l – length of cylindrical portion of end plate (Fig. 8.2.11.1-1 and item 1.1 in the Annex 1), [mm].



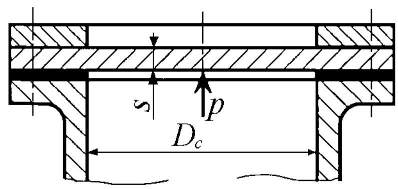
$K = 0.30$

Fig. 8.2.11.1-1



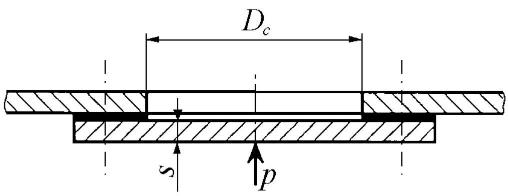
$K = 0.41$

Fig. 8.2.11.1-2



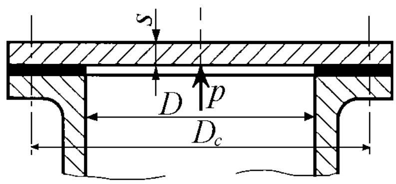
$K = 0.41$

Fig. 8.2.11.1-4



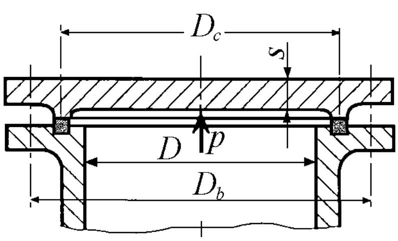
$K = 0.45$

Fig. 8.2.11.1-3



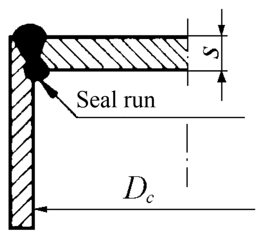
$K = 0.35$

Fig. 8.2.11.1-5



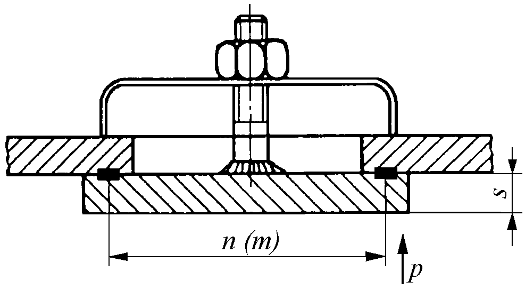
D_b/D	K
1.25	0.6
1.50	0.7
1.75	0.8

Fig. 8.2.11.1-6



$K = 0.50$

Fig. 8.2.11.1-7



$K = 0.53$

Fig. 8.2.11.1-8

8.2.11.2 The thickness of the end plates shown in item 1.2 of the Annex 1 is not to be less than that determined from formula 8.2.11.1-1. Additionally, the following conditions shall be satisfied:

.1 For circular end plates

$$0.77s_1 \geq s_2 \geq \frac{1.3p}{\sigma}\left(\frac{D_c}{2} - r\right) \tag{8.2.11.2.1}$$

.2 For rectangular end plates

$$0.55s_1 \geq s_2 \geq \frac{1.3p}{\sigma} \cdot \frac{nm}{(n+m)} \tag{8.2.11.2.2}$$

s – thickness of the end plate , [mm];
s_1 – shell thickness , [mm];
s_2 – thickness of the end plate in the relieving groove area, [mm].

For explanation of other symbols – see 8.2.11.

In no case the value of s_2 shall be less than 5 mm.

The above conditions are applicable to end plates of 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 will be specially considered by PRS in each particular case.

8.2.12 Flat Walls Reinforced by Stays

8.2.12.1 Flat walls (Figs. 8.2.12.1-2 and 8.2.12.1-3) reinforced by long and short stays, corner stays, stay tubes or other similar structures shall have a thickness not less than determined by the formula:

$$s = KD_c\sqrt{\frac{p}{\sigma}} + c \tag{8.2.12.1-1}$$

K – design factor (see Figs. 8.2.12.1-1 ÷ 8.2.12.1-3 and items 5.1 ÷ 5.3 in the Annex 1); if the part of the wall area in question is reinforced by stays having variable values of K factor, the K value in the formula shall be equal to the arithmetic mean of these factors;
D_c – assumed design diameter (Figs. 8.2.12.1-2 and 8.2.12.1-3), [mm];

with uniform distribution of stays:

$$D_c = \sqrt{a_1^2 + a_2^2} \qquad (8.2.12.1\text{-}2)$$

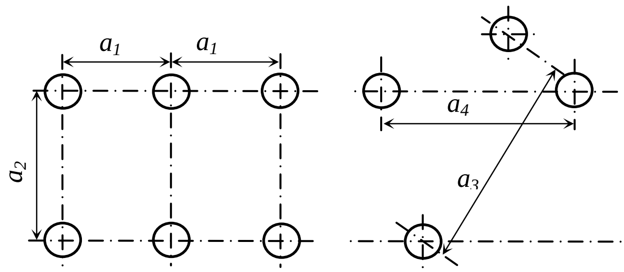
with non-uniform distribution of stays:

$$D_c = \frac{a_3 + a_4}{2} \qquad (8.2.12.1\text{-}3)$$

In all other cases, the values of D_c shall be taken as equal to the diameter of the largest circle which can be drawn through the centres of three stays or through the centres of stays and the commencement of the wall flanging curvature if the radius of the latter satisfies the requirements of 8.2.13; in this case, the flanging shall be regarded as a reinforced point. A manhole flanging is not to be regarded as a reinforced point;

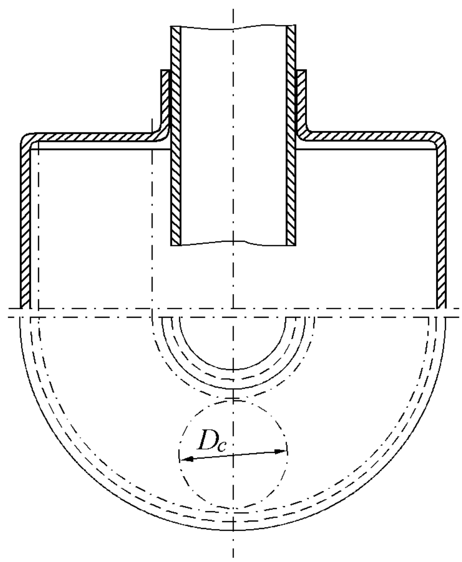
a_1, a_2, a_3, a_4 – pitch or stay-to-stay distance (Fig. 8.2.12.1-1), [mm].

For other symbols – see 8.2.11.1.



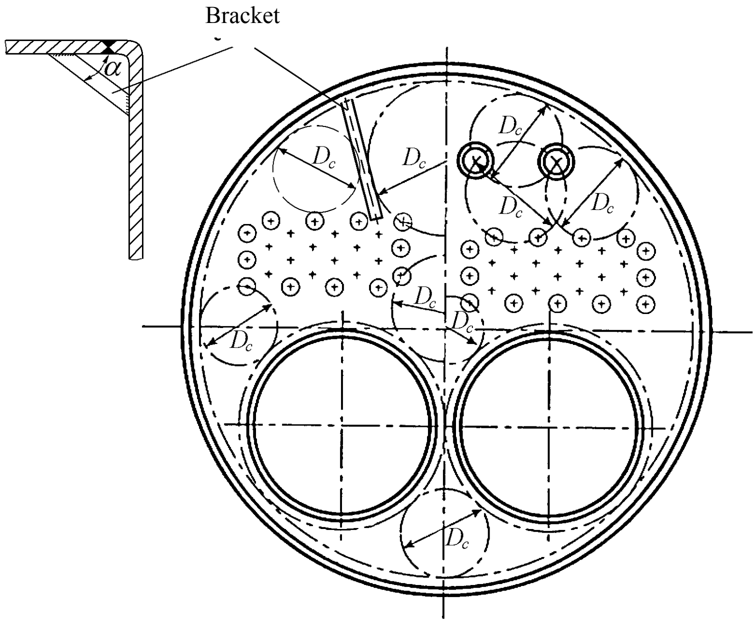
$K = 0.45$

Fig. 8.2.12.1-1



$K = 0.50$

Fig. 8.2.12.1-2



K = 0.35 (for the bracket)

Fig. 8.2.12.1-3

8.2.13 Flanged Flat Walls and End Plates

8.2.13.1 In flat wall and end plate thickness calculations, the flanging can be taken into account only when the inner flanging radius is not less than that given in Table 8.2.13.1.

Table 8.2.13.1

Outer diameter of end plate, [mm]	Flanging inner radius, [mm]
up to 350	25
above 350 to 500	30
above 500 to 950	35
above 950 to 1400	40
above 1400 to 1900	45
over 1900	50

The inner flanging radius is not to be less than 1.3 times the wall thickness.

8.2.13.2 The length of cylindrical portion of a flanged flat end plate is not to be less than: $l = 0.5\sqrt{Ds}$, (see Fig. 8.2.11.1-1).

8.2.14 Reinforcing Holes in Flat Walls

8.2.14.1 Holes in flat walls, end plates and covers, of diameters exceeding four times thickness, shall be reinforced by means of welded-on branch pieces or pads, or by increasing the design wall thickness. The holes shall be arranged at a distance not less than 0.125 times the design diameter from the design diameter outline.

8.2.14.2 If the actual wall thickness is larger than that calculated from formulae 8.2.11.1-1 and 8.2.12.1-1, the maximum diameter of a non-compensated hole shall be determined from the formula:

$$d = 8s_r\left(1.5\frac{s_r^2}{s^2} - 1\right) \qquad (8.2.14.2)$$

d – diameter of a non-compensated hole, [mm];
s_r – actual wall thickness, [mm];
s – design wall thickness obtained from the formulae 8.2.11.1-1 and 8.2.12.1-1, [mm].

8.2.14.3 Edge reinforcement shall be provided for holes of larger dimensions than those indicated in 8.2.14.1 and 8.2.14.2.

The dimensions of reinforcing elements of branch pieces shall comply with the equation:

$$s_k\left(\frac{h^2}{s_r^2} - 0.65\right) \ge 0.65d - 1.4s_r \qquad (8.2.14.3)$$

s_k – thickness of branch wall (see 8.2.14.3), [mm],
d – inner diameter of branch, [mm];
s_r – see 8.2.14.2, [mm];
h = $h_1 + h_2$,[mm], (Fig. 8.2.14.3).

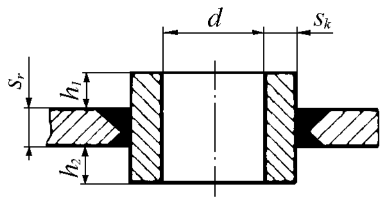


Fig. 8.2.14.3

8.2.15 Tube Plates

8.2.15.1 Flat tube plates of heat exchangers shall have the thickness s_1 not less than that determined by the formula:

$$s_1 = 0.9KD_W\sqrt{\frac{p}{\sigma\varphi}} + c \quad , \quad [\text{mm}] \qquad (8.2.15.1)$$

K – factor depending on the ratio of the shell wall thickness *s* to the tube plate thickness s_1; for the tube plates welded to the shell, *K* shall be taken from diagram 8.2.15.1 with the preliminary assumption of s_1 thickness, the calculation shall be corrected if the difference between assumed s_1 value and that obtained from formula 8.2.15.1 exceeds 5%;
for the tube plate fastened by bolts or stud-bolts between the body and cover flanges, $K = 0.5$;

D_W – inner diameter of shell, [mm];

p – design pressure (see 8.2.2), [MPa];

σ – allowable stress (see 8.2.4.5), [MPa];
σ shall be reduced by 10% in the case of heat exchangers of rigid structure where the thermal elongation factors of the body and pipe materials are different;

φ – efficiency factor of a tube plate weakened by pipe holes (see 8.2.15.2);

c – design thickness allowance (see 8.2.7), [mm].

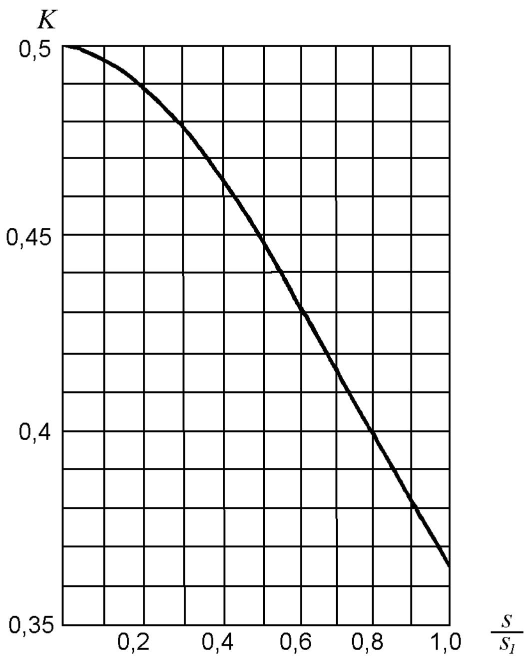


Fig. 8.2.15.1

8.2.15.2 Where $0.75 > d / a > 0.4$ and $D_W / s_1 \geq 40$, the efficiency factor of a tube plate shall be calculated from the following formulae:

– where holes are arranged according to equilateral triangle:

$$\varphi = 0.935 - 0.65\frac{d}{a} \tag{8.2.15.2-1}$$

– where holes are arranged in line or in transposition:

$$\varphi = 0.975 - 0.68\frac{d}{a_2} \tag{8.2.15.2-2}$$

d – diameter of tube plate holes, [mm];
a – spacing of axes of holes arranged in triangles, [mm];
a_2 – spacing of axes of holes arranged in line or in transposition (as well as arranged concentrically), whichever less, [mm].

8.2.15.3 When the quotient d / a = 0.75 ÷ 0.80, the thickness of the tube plate calculated by formula 8.2.15.1 shall comply with the requirement:

$$f_{\min} \geq 5d$$

$f_{\min}$ – minimum permissible cross-section of a bridge located in a tube plate, [mm^2].

In the case where the values of d / a and D_W / s_1 differ from those given above, as well as for heat exchangers of the rigid structure when the mean temperature difference exceeds 50 °C, the thickness of tube plates will be specially considered by PRS.

8.2.15.4 The thickness of tube plates with expanded tubes, except complying with the requirements of 8.2.15.1, is also to satisfy the following requirement:

$$s \geq 10 + 0.125d \tag{8.2.15.4}$$

The expanded connections of tubes with tube plates are also to comply with the requirements of 8.2.20.6, 8.2.20.7 and 8.2.20.8.

8.2.15.5 If tube plates are reinforced by the welded or expanded pipes complying with the requirements of 8.2.20, the calculations of such tubes may be made according to 8.2.12.

8.2.16 Dished Ends

8.2.16.1 Dished ends, unpierced or pierced, subjected to internal or external pressure, (see Fig. 8.2.16.1), shall have a thickness not less than that determined by the formula:

$$s = \frac{D_a p y}{4\sigma\varphi} + c \tag{8.2.16.1}$$

s – wall thickness of the dished end, [mm];
p – design pressure, [MPa];
D_a – outer diameter of the end, [mm].
The end shall be flanged within area not less than 0.1 D_a measured from the outer edge of cylindrical portion of the dished end (see Fig. 8.2.16.1);
φ – efficiency factor (see 8.2.6);
σ – allowable stress (see 8.2.4.5), [MPa];

y – shape factor selected from Table 8.2.16.1, depending on the ratio of the height of the dished end to its outside diameter and on the value of weakening by the holes; for intermediate values of h_a / D_a and $d / \sqrt{D_a s}$, the shape factor *y* may be determined by interpolation.

When using Table 8.2.16.1 for determining *y*, the preliminary value *s* is taken from the standard thickness series. The final value of *s* is not to be less than that defined by formula 8.2.16.1.

For elliptical and basket shaped ends, R_W is the maximum radius of curvature.

Table 8.2.16.1

End shape	$\frac{h_a}{D_a}$ ratio	Shape factor							
		y – for flanged area and unpierced ends	y_A – for dished part of end with non-compensated holes in respect to $\frac{d}{\sqrt{D_a s}}$						y_c – for dished part of end with compensated 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 $R_W = 0.8D_a$	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 – allowance to design thickness, [mm], to be taken equal:
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 a non-compensated hole, [mm].

Formula 8.2.16.1 is valid where the following relations are satisfied:

$$\frac{h_a}{D_a} \geq 0.18\,; \quad \frac{s-c}{D_a} \geq 0.0025; \quad R_W \leq D_a\,; \quad r \geq 0.1D_a\,; \quad l \leq 150 \text{ mm},$$

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 end elements are shown in Fig. 8.2.16.1.

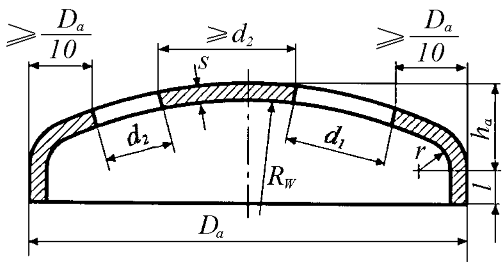


Fig. 8.2.16.1

8.2.16.2 The unpierced end means also an end where the holes of diameter not exceeding 4*s* and not greater than 100 mm are arranged at a distance of not less than $0.2D_a$ from the outer cylindrical portion of the end. Non-compensated holes with diameters less than the wall thickness and not exceeding 25 mm are allowed.

8.2.16.3 The wall thickness of dished ends in combustion chambers of vertical boilers can be calculated as for unpierced ends, also where the flue gas outlet branch passes through the end.

8.2.16.4 Dished ends subjected to external pressure, except for those of cast iron, shall be checked for shape stability using the following formula:

$$\frac{36.6E_T}{R_W^2} \cdot \frac{(s-c)^2}{100p} > 3.3 \qquad (8.2.16.4)$$

E_T – modulus of elasticity at design temperature, [MPa], for modulus of elasticity for steel – see Table 8.2.16.4, for non-ferrous metals the modulus of elasticity value shall be agreed with PRS;

R_W – maximum inner radius of curvature, [mm].

The other symbols used are the same as in 8.2.16.1.

Table 8.2.16.4

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

8.2.16.5 The minimum wall thickness of dished steel ends shall be not less than 5 mm. For ends manufactured of non-ferrous alloys and stainless steels, the minimum wall thickness may be reduced, on agreement with PRS.

8.2.16.6 The use of dished ends of welded construction will be specially considered by PRS in each particular case.

8.2.17 Flanged End Plates

Unpierced flanged end plates (see. Fig. 8.2.17) subjected to internal pressure shall have a thickness not less than that determined by the formula:

$$s = \frac{3Dp}{\sigma} + c \tag{8.2.17}$$

s – wall thickness, [mm];
p – design pressure (see 8.2.2), [MPa];
D – inner diameter of the end plate, taken equal to the internal diameter of the shell, [mm];
σ – allowable stress (see 8.2.4.5), [MPa];
c – design thickness allowance (see 8.2.7), [mm].

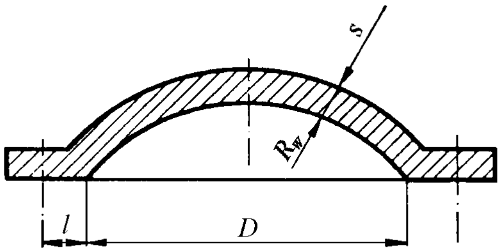


Fig. 8.2.17

Flanged end plates are allowed within a range of diameters D up to 500 mm and for design pressures not higher than 1.5 MPa. The radius of curvature of the end plate R_W is not to be greater than 1.2 D, while the distance l is not to exceed 2s.

8.2.18 Headers of Rectangular Section

8.2.18.1 The wall thickness of rectangular headers (Fig. 8.2.18.1-1) subject to internal pressure is not to be less than that determined by the formula:

$$s = \frac{pn}{2.52\sigma\varphi_1} + \sqrt{\frac{4.5Kp}{1.26\sigma\varphi_2}} \tag{8.2.18.1-1}$$

s – wall thickness, [mm];
p – design pressure (see 8.2.2), [MPa];
n – half of the inner width of the header side perpendicular to that being calculated, [mm];
m – half of the inner width of the header side being calculated, [mm];
σ – allowable stress (see 8.2.4.5), [MPa];
φ_1 and φ_2 – efficiency factors of headers, weakened by holes, determined as follows:

φ_1 – from formula 8.2.6.2.1;
φ_2 – from formula 8.2.6.2.1, if $d < 0.6$ m;

$$\varphi_2 = 1 - \frac{0.6m}{a}, \text{ if } d \geq 0.6 \text{ m}; \tag{8.2.18.1-2}$$

d – diameter of holes, [mm]. For oval holes, d shall be taken as equal to size of the holes at the longitudinal axis, but for use in formulae 8.2.6.2.1 and 8.2.18.1-2 d shall be taken as the size at the holes perpendicular to the header centre line.

Where the holes are arranged in staggered pattern, a_2 (Fig. 8.2.18.1-2) shall be substituted for a in formula 8.2.18.1-2. Where the rectangular headers have longitudinal welded joints (see Fig. 8.2.18.1-1), the efficiency factors, φ_1 and φ_2 shall be assumed to be equal, respectively, to the joint factors of weld seams selected as per 8.2.6.

Longitudinal welded joints shall be arranged, as far as possible, within the area l_1, for which $K = 0$. Where the header wall is weakened in several different places, the calculations shall be based on the lowest efficiency factor value.

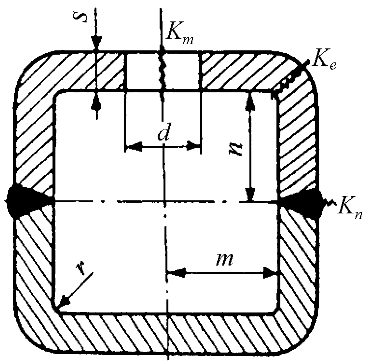


Fig. 8.2.18.1-1

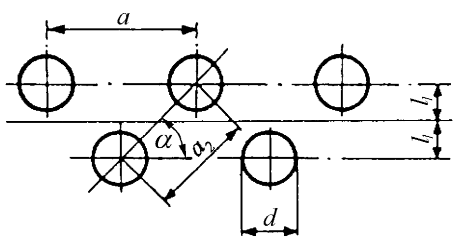


Fig. 8.2.18.1-2

K – design factor for bending moment at the centre of side wall or at the centre line of the row of holes, calculated from the formulae:

for the centre line of the header side wall

$$K_m = \frac{m^3 + n^3}{3(m+n)} - \frac{m^2}{2}, \quad [\text{mm}^2] \qquad (8.2.18.1\text{-}3)$$

for rows of holes or longitudinal welded joints

$$K_n = \frac{m^3 + n^3}{3(m+n)} - \frac{m^2 - l_1^2}{2}, \quad [\text{mm}^2] \qquad (8.2.18.1\text{-}4)$$

If the above formulae give negative values, the absolute numerical values shall be used; where the holes are arranged in a staggered pattern, factor K shall be multiplied by cos α;

α – angle between diagonal pitch line of holes and header axis, [°];

l_1 – distance between the row of holes under consideration and the centre line of header wall (Fig. 8.2.18.1-2), [mm].

8.2.18.2 Where fillet weld joints are allowed in rectangular headers by special approval of PRS, the wall thickness of such headers is not to be less than that determined by the formula:

$$s = \frac{p\sqrt{m^2 + n^2}}{2.52\sigma\varphi_1} + \sqrt{\frac{4.5K_e p}{1.26\sigma\varphi_2}} \qquad (8.2.18.2\text{-}1)$$

K_e – the design factor for bending moment at the edges, [mm^2], determined from the formula:

$$K_e = \frac{m^3 + n^3}{3(m+n)} \qquad (8.2.18.2\text{-}2)$$

For other symbols used – see 8.2.18.1.

8.2.18.3 The inner radius of curvature of rectangular header side edges is not to be less than 0.33 times the wall thickness and never less than 8 mm. The minimum thickness of header wall designed to accommodate expanded tubes is not to be less than 14 mm. The width of ligaments between holes is not to be less than 0.25 times the pitch between hole centres. The wall thickness in the area of curvature is not to be less than that determined by formulae 8.2.18.1-1 and 8.2.18.2-1.

8.2.19 Holes in Cylindrical, Spherical, Conical Walls and in Dished Ends

8.2.19.1 Regions of holes shall be compensated. The following compensations are allowed:

.1 holes compensated by wall thickness exceeding the design value (Figs. 8.2.19.1-1 and 8.2.19.1-2);

.2 holes compensated by means of disk-shaped reinforcing plates attached by welding to the wall to be reinforced (Figs. 8.2.19.1-3 and 8.2.19.1-4);

.3 holes compensated by means of welded-on tubular elements, such as nozzles, sleeves, branch pieces, etc. (Figs. 8.2.19.1-5 ÷ 8.2.19.1-7).

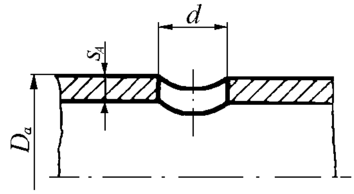


Fig. 8.2.19.1-1

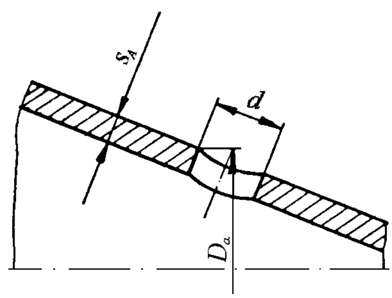


Fig. 8.2.19.1-2

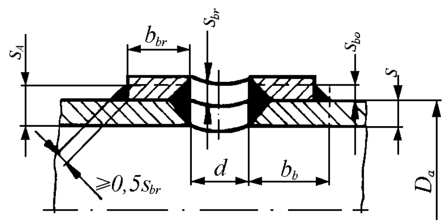


Fig. 8.2.19.1-3

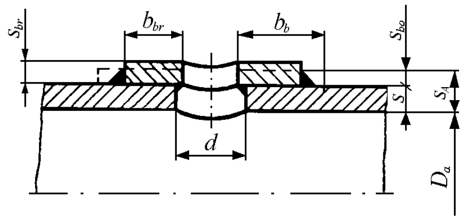
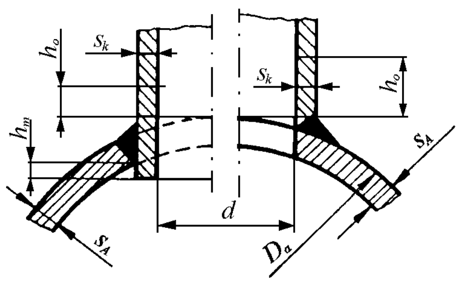


Fig. 8.2.19.1-4



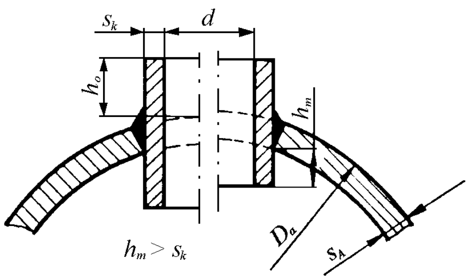
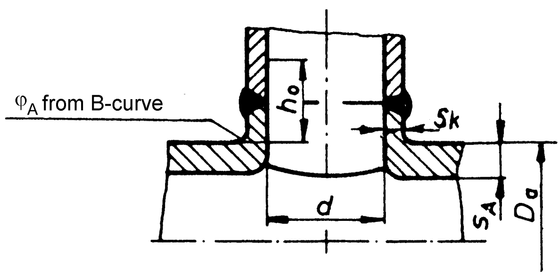


Fig. 8.2.19.1-5 Fig. 8.2.19.1-6



φ_A from B-curve

Fig. 8.2.19.1-7

The compensating elements for holes, as shown in Figs. 8.2.19.1-5 to 8.2.19.1-7, are recommended for welding with use of removable backing strip or some other technique ensuring adequate penetration of the weld joint.

8.2.19.2 The thickness of walls having holes shall be in accordance with requirements 8.2.8 and 8.2.9 for cylindrical walls, 8.2.10 – for conical walls and 8.2.16 – for dished ends.

8.2.19.3 The materials used for the wall to be reinforced and for the reinforcing elements shall have identical strength characteristics as far as possible. Where use is made of reinforcing materials with inferior strength characteristics as compared to material of the wall to be reinforced, the area of the reinforcing sections shall be increased accordingly.

The compensating elements shall be reliably attached to the wall to be reinforced.

8.2.19.4 Holes in the walls shall be located at a distance of at least three wall thicknesses, but not less than 50 mm away from the welded joints. The arrangement of holes at the distance less than 50 mm from welded joints will be specially considered by PRS.

8.2.19.5 The dimension of reinforced holes is not to exceed 500 mm. The use of holes measuring over 500 mm and their compensation method will be specially considered by PRS in each particular case.

8.2.19.6 The thickness of tubular elements (branches, sleeves or nozzles) attached by welding to the walls of boilers, pressure vessels and heat exchangers is generally to be taken as not less than 5 mm. Use of the said elements with thickness below 5 mm will be specially considered by PRS.

8.2.19.7 A hole may be compensated for by increase of the wall thickness above its design value. In such a case the increased thickness s_A is not to be less than the value obtained from the following formulae:

for cylindrical walls

$$s_A = \frac{pD_a}{2\sigma\varphi_A + p} + c \qquad (8.2.19.7\text{-}1)$$

for spherical walls

$$s_A = \frac{pD_a}{4\sigma\varphi_A + p} + c \qquad (8.2.19.7\text{-}2)$$

for conical walls

$$s_A = \frac{pD_a}{(2\sigma\varphi_A - p)\cos\alpha} + c \qquad (8.2.19.7\text{-}3)$$

s_A – required wall thickness without compensating elements, [mm];

φ_A – efficiency factor of the wall weakened by a hole and to be reinforced, determined depending upon the nondimensional parameter $\frac{d}{\sqrt{D_a\left(s_A - c\right)}}$ from curve *A* (see Fig. 8.2.19.7); it shall be noted that the value of s_A in the expression for this nondimensional parameter shall be obtained from formulae 8.2.19.7-1 ÷ 8.2.19.7-3;

d – diameter of the hole (inner diameter of a nozzle, sleeve, or branch) or the size of an oval or elliptical hole along the longitudinal axis, [mm].

For other symbols – see 8.2.8.2 and 8.2.10.1.

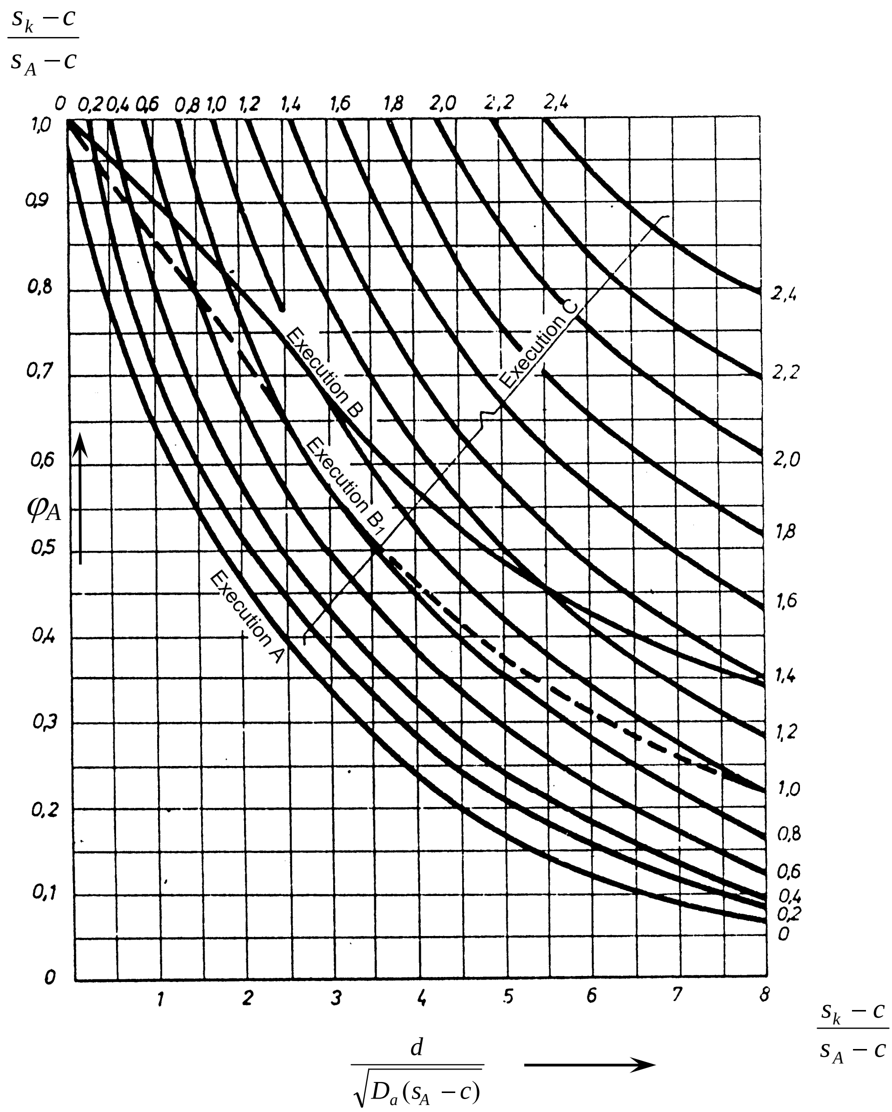


Fig 8.2.19.7

8.2.19.8 Where disc-shaped reinforcing plates are used as compensation for holes in cylindrical, spherical or conical walls, the dimensions of the reinforcing plates shall be determined by the following formulae:

$$b_b = \sqrt{D_a\left(s_A - c\right)} \qquad (8.2.19.8\text{-}1)$$

$$s_{bo} \geq s_A - s_r \qquad (8.2.19.8\text{-}2)$$

b_b – maximum effective width of the plate (see Fig. 8.2.19.1-3 and 8.2.19.1-4), [mm];

s_{bo} – height (thickness) of the plate (see Figs. 8.2.19.1-3 and 8.2.19.1-4), [mm];
s_A – total thickness of the wall to be reinforced and of the reinforcing plate, determined as per 8.2.19.7, [mm];
s_r – actual thickness of the wall to be reinforced, [mm].
For other symbols used – see 8.2.19.7.

In the case where actual width of reinforcing plate is lower than resulting from formula 8.2.19.8-1, thickness of the plate shall be increased respectively according the formula:

$$s_{br} \geq s_{bo} \frac{1 + \dfrac{b_b}{b_{br}}}{2} \tag{8.2.19.8-3}$$

s_{br} – actual height (thickness) of the plate, [mm];
b_{br} – actual width of the plate, [mm].

The height of weld seam joining the reinforcing plate to the wall is not to be less than 0.5 s_{br} (Fig. 8.2.19.1-3).

8.2.19.9 Where welded tubular elements are used as compensation for holes in cylindrical, spherical and conical walls, the dimensions of the said elements are not to be less than the values obtained as follows:

.1 The wall thickness s_k of a tubular element (nozzle, sleeve, etc.), [mm], shall be determined depending on the nondimensional parameter

$$\frac{d}{\sqrt{D_a\left(s_A - c\right)}}$$

and the efficiency factor φ_A taken from curve C represented in Fig. 8.2.19.7. The values φ_r and s_r shall be substituted for φ_A and s_A shown in Fig. 8.2.19.7:
s_r – actual wall thickness, [mm];
φ_r – actual efficiency factor of a wall having thickness s_r, as determined by formulae 8.2.8.2-1, 8.2.8.2-2, 8.2.8.3-1, 8.2.8.3-2 and 8.2.10.1.2 by solving the equations of the said formulae for φ.

The ratio:

$$\frac{s_k - c}{s_A - c}$$

obtained from Fig. 8.2.19-7, shall be used for determining the minimum thickness s_k [mm] of a nozzle, sleeve, etc. It will be noted that in the above ratio, the actual thickness s_r shall be substituted for s_A.

.2 The minimum design height h_0 [mm] of a tubular stiffener (nozzle, sleeve, tube) shall be determined by the formula:

$$h_0 = \sqrt{d\left(s_k - c\right)} \tag{8.2.19.9.2-1}$$

If the actual height of a tubular stiffener h_r is less than that determined by the formula 8.2.19.9.2-1, the thickness s_k shall be increased respectively according to the formula:

$$s_{kr} = s_k \frac{h_0}{h_r} \qquad (8.2.19.9.2\text{-}2)$$

8.2.19.10 Holes in dished ends shall be compensated for as follows:

.1 For holes reinforced by increasing wall thickness, the factor y in formula 8.2.16.1 shall be replaced by factor y_A obtained from Table 8.2.16.1.

.2 For holes reinforced by means of disk-shaped reinforcing plates, the plate dimensions shall be determined in compliance with 8.2.19.8, the overall thickness of the wall to be reinforced s_A being calculated from the formula:

$$s_A = \frac{p(R_W + s)y_0}{2\sigma\varphi_A} + c \qquad (8.2.19.10.2)$$

R_W – inner radius of the end curvature in the way of the hole, [mm];
y_0 – shape factor selected from Table 8.2.16.1.
For other symbols – see 8.2.16.1 and 8.2.19.7.

.3 For openings with tubular stiffeners, the stiffener dimensions shall be determined according to 8.2.19.9 except that the value $2(0.5D_a + s)$ shall be substituted for D_a in the nondimensional parameter

$$\frac{d}{\sqrt{D_a(s-c)}}$$

and the actual efficiency factor φ for the ends having the thickness s shall be determined by formula 8.2.16.1 for φ, assuming $\varphi = \varphi_A$, $y = y_0$ and $s = s_A$ (see 8.2.16.1).

8.2.19.11 For through tubular stiffeners with the inward projecting portion $h_m \geq s_r$ (Fig. 8.2.19.1-5 and 8.2.19.1-6), the thickness of the tubular stiffener may be reduced by 20%, but it is not to be less than that required for the design pressure.

8.2.19.12 The ratio of the wall thickness of a tubular stiffener s_k to the thickness of the wall to be reinforced s is not to be greater than 2.4. If, for reasons of design, this ratio is taken as more than 2.4, the thickness value for the tubular stiffener s_k equal to not more than 2.4 times the thickness of the wall to be reinforced shall be used in the calculation.

8.2.19.13 Disk-shaped reinforcing plates and tubular stiffeners can be used in combinations as compensation for holes (Fig. 8.2.19.13). In such a case the dimensions of the compensating elements shall be determined with requirements for the disk-shaped and tubular stiffeners being simultaneously taken into account.

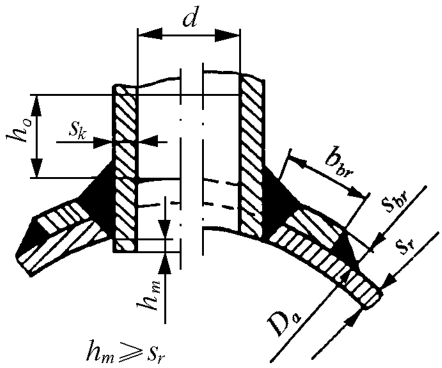


Fig. 8.2.19.13

8.2.19.14 For branches drawn of the reinforced wall (Fig. 8.2.19.1-7), the thickness s_A is not to be less than that determined by formulae 8.2.19.7-1 ÷ 8.2.19.10-2.

The efficiency factor φ_A of the wall weakened due to provision of a drawn branch shall be determined from diagram 8.2.19.7 as follows:

for $\dfrac{d}{D_a} \le 0.4$, from the curve *B*,

for $\dfrac{d}{D_a} = 1.0$, from the curve B_1,

for $0.4 < \dfrac{d}{D_a} < 1.0$, by interpolation of curves B and B_1.

The thickness of the shoulder of a drawn branch s_k is not to be less than that determined by the formula:

$$s_k \ge s_A \frac{d}{D_a} \quad , \quad [\text{mm}] \tag{8.2.19.14}$$

and not less than that required for the design pressure.

8.2.19.15 The effect of adjacent holes may be disregarded provided that:

$$\left(l + s_{kr1} + s_{kr2}\right) \ge 2\sqrt{D_a\left(s_r - c\right)} \tag{8.2.19.15-1}$$

$\left(l + s_{kr1} + s_{kr2}\right)$ – distance between two adjacent holes (Figs. 8.2.19.15-1 and 8.2.19.15-2), [mm];

D_a – outer diameter of the wall to be reinforced, [mm];

s_r – actual thickness of the wall to be reinforced, [mm];

c – design thickness allowance (see 8.2.7), [mm].

If the distance $\left(l + s_{kr1} + s_{kr2}\right) < \sqrt{D_a\left(s_r - c\right)}$, a check shall be made to ascertain the stress occurring in the section between the holes due to design pressure.

The stresses involved are not to exceed, both longitudinally and laterally, the allowable values obtained from the formula:

$$\frac{F}{f_c} \le \sigma \qquad (8.2.19.15\text{-}2)$$

σ – allowable stress (see 8.2.4.5), [MPa];
F – load exerted by the design pressure upon the section between the holes (see 8.2.19.16), [N];
f_c – cross-sectional area between the holes (see 8.2.19.17), [mm²].

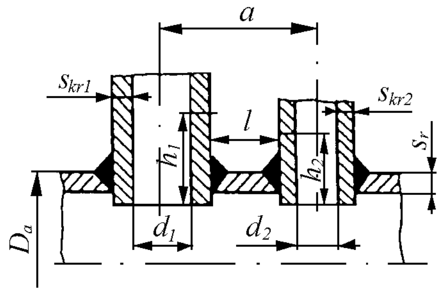


Fig. 8.2.19.15-1

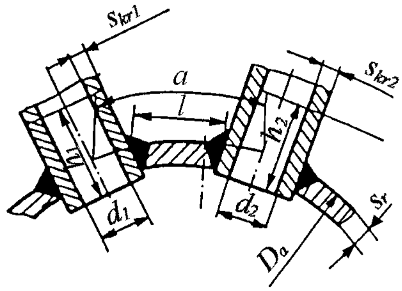


Fig. 8.2.19.15-2

8.2.19.16 The load exerted by the design pressure affecting the sectional area between two holes shall be determined as follows:

.1 for holes arranged longitudinally along a cylindrical wall:

$$F_a = \frac{Dpa}{2}\ , \quad [\mathrm{N}] \qquad (8.2.19.16.1)$$

.2 for openings arranged circumferentially in cylindrical or conical walls, as well as in the spherical walls:

$$F_b = \frac{Dpa}{4}\ , \quad [\mathrm{N}] \qquad (8.2.19.16.2)$$

.3 for openings in dished ends:

$$F_b = \frac{R_B pay}{2}\ , \quad [\mathrm{N}] \qquad (8.2.19.16.3\text{-}1)$$

a – spacing (pitch) between two adjacent holes, measured at the outer circumference, as shown in Fig. 8.2.19.15-2, [mm];
D – inner diameter (for conical walls measured at the centre of the opening), [mm];
p – design pressure, [MPa];
R_B – inner radius of curvature (see 8.2.19.10), [mm];
y – shape factor (see 8.2.16.1).

Where holes are arranged in cylindrical walls with a diagonal pitch, the load F involved shall be determined from appropriate formula, the results obtained from this formula being multiplied by a factor:

$$K = 1 + \cos^2\alpha \qquad (8.2.19.16.3\text{-}2)$$

α – angle of inclination of the line of the opening centres to the longitudinal axis, [degs].

8.2.19.17 The sectional area, f_c, [mm^2] of the wall between two adjacent holes shall be determined for tubular stiffeners as shown below:

$$f_c = l(s-c) + 0.5[h_1(s_{kr1}-c) + h_2(s_{kr2}-c)], \text{ [mm}^2\text{]} \quad (8.2.19.17\text{-}1)$$

h_1 and h_2 – height of the stiffeners, [mm], determined from the formulae:

for blind stiffeners

$$h_{1,2} = h_0 + s \quad (8.2.19.17\text{-}2)$$

for through stiffeners

$$h_{1,2} = h_0 + s + h_m \quad (8.2.19.17\text{-}3)$$

l – width of the ligament between two adjacent holes (Figs. 8.2.19.15-1 and 8.2.19.15-2), [mm];
s – thickness of the wall to be reinforced, [mm];
s_{kr1} and s_{kr2} – thicknesses of the tubular stiffener walls (Figs. 8.2.19.15-1 and 8.2.19.15-2), [mm];
c – design thickness allowance, [mm], (see 8.2.7);
h_0 – design height of the tubular stiffener (formula 8.2.19.9.2-1), [mm];
h_m – design height of the tubular stiffener protruding inwards (see Fig. 8.2.19.1-5, 8.2.19.1-6 and 8.2.19.13), [mm].

For holes to be reinforced by other methods (combined or disc-shaped stiffeners, etc.), the values of f_c shall be determined in a similar manner.

8.2.19.18 For drawn branches arranged in a row, it will be necessary to check that the efficiency factor of the walls weakened by holes φ, as determined by formula 8.2.6.2.1 for this row, is not less than the efficiency factor φ_A, determined from curves B and B1 in Fig. 8.2.19.7. If $\varphi < \varphi_A$, the value of φ shall be used for determination of the wall thickness according to 8.2.19.14.

This requirement is also applicable to welded nozzles arranged in a row, the thickness of which is determined only on the basis of the internal pressure effect.

8.2.20 Stays

8.2.20.1 The cross-sectional area of long and short stays, corner stays and stay tubes, subject to tensile or compressive stresses is not to be less than that calculated from the formula:

$$f = \frac{pf_s}{\sigma \cos\alpha} \quad (8.2.20.1)$$

f – cross-sectional area of the single stay, [mm^2];
p – design pressure (see 8.2.2), [MPa];
σ – allowable stress (see 8.2.4.5), [MPa];
α – angle between the corner stay and the wall to which the stay is attached, [degs], (Fig. 8.2.12.1-3);

f_s – maximum surface area of the wall to be reinforced per stay, [mm^2]. This area is bounded by straight lines passing at right angles through the centres of the lines interconnecting the centre line of stay with the adjacent reinforced points (stays). The cross-sectional area of the stays and tubes within this area may be calculated from the surface area per stay.

8.2.20.2 For stays subject to longitudinal bending, the allowable bending stresses shall be taken with safety factor not less than 2.25.

8.2.20.3 Where use is made of end plates with a single reinforcing stay (Fig. 8.2.20.3), the latter shall be so designed that it may take up at least half the load upon the end plate. An end plate of this type shall have a thickness in compliance with the requirements of 8.2.12.1

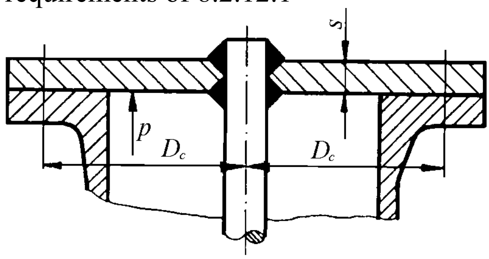


Fig. 8.2.20.3

8.2.20.4 Stay and ordinary fire tubes shall have a wall thickness not less than the values given in Table 8.2.20.4.

The thickness of stay fire tubes with diameter over 70 mm is not to be less than:
6 mm – for peripheral tubes,
5 mm – for tubes arranged inside the tube nest.

Table 8.2.20.4

Outside diameter of tubes, [mm]	Tube wall thickness, [mm]			
	3.0	3.5	4.0	4.5
	Maximum working pressure, [MPa]			
50	1.10	1.85	–	–
57	1.00	1.65	–	–
63.5	0.90	1.50	2.10	–
70	0.80	1.35	1.90	–
76	0.75	1.25	1.75	2.25
83	–	1.15	1.60	2.10
89	–	1.05	1.50	1.90

8.2.20.5 The cross-sectional area of welded joints of stays shall be such as to satisfy the following condition:

$$\frac{\pi d_a e}{f} \geq 1.25 \qquad (8.2.20.5)$$

d_a – stay diameter or outside diameter for tubes, [mm];

e – weld joint thickness (items 5.1 ÷ 5.3 in the Annex 1), [mm];
f – cross-sectional area of the stay (see 8.2.20.1); [mm^2].

8.2.20.6 For expanded tubes, the length of the expansion belt in the tube plate is not to be less than 12 mm. Expansion joints for working pressures above 1.6 MPa shall be made with sealing grooves.

8.2.20.7 Expansion joints shall be checked for secure seating of the tubes in the tube plates by axial testing loads. The tubes may be considered securely seated, if the value obtained from the formula:

$$\frac{pf_s}{20sl} \qquad (8.2.20.7)$$

does not exceed:
15 – for joints of plain tubes,
30 – for joints with sealing grooves,
40 – for joints with flanging of tubes;
s – thickness of the tube, [mm];
l – length of the expansion belt, [mm];
The other symbols used are the same as in 8.2.20.1.
The length of expansion belt of tubes *l* is not to be taken greater than 40 mm.

8.2.20.8 The expansion belt of plain pipes shall be not less than that determined by the formula:

$$l = \frac{pf_s K_r}{q}, \quad [\text{mm}] \qquad (8.2.20.8\text{-}1)$$

where:
$K_r = 5.0$ – safety factor of expanded joint;
p, f_s – see 8.2.20.1.
q – strength of pipe joint over 1 mm of expansion belt, evaluated experimentally from the below formula, [N/mm]:

$$q = \frac{F}{l_1}, \quad [\text{N/mm}] \qquad (8.2.20.8\text{-}2)$$

where:
F – axial force necessary for drawing the expanded tube from the tube plate, [N];
l_1 – length of expansion belt used for experimental evaluation of value *q*.

8.2.21 Top Girders

The section modulus of top girders of rectangular section is not to be less than that determined by the formula:

$$W = \frac{1000M}{1.3\sigma Z} \qquad (8.2.21\text{-}1)$$

W – section modulus for single girder, [mm^3];
σ – allowable stress (see 8.2.4.5), [MPa];

Z – coefficient of rigidity of the wall to be reinforced, for the structure shown in Fig. 8.2.21, Z=1.33;

M – bending moment of the single girder, [Nm], for a rectangular section the moment shall be determined by the formula:

$$M = \frac{pal^2}{8000} \qquad (8.2.21\text{-}2)$$

l – design length of the girder, [mm], (Fig. 8.2.21);

p – design pressure, [MPa];

a – spacing between axes of adjacent girders, [mm];

s_1 – width of the girder, [mm], (Fig. 8.2.21);

h – height of the girder which is not to exceed 8 s_1 [mm], (Fig. 8.2.21).

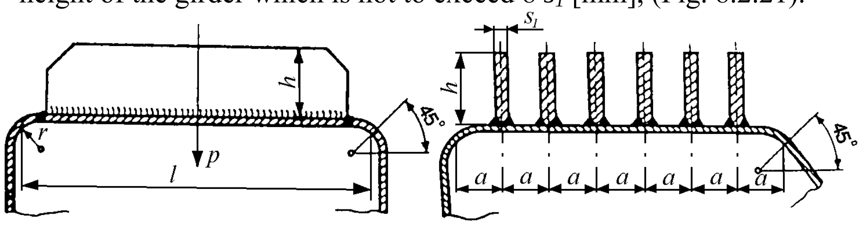


Fig. 8.2.21

8.3 Supervision, Testing and Certificates

8.3.1 The following essential parts of boilers, steam superheaters, economisers and water heated steam generators are subject to supervision in the process of construction for compliance with the approved documentation:

- ring segments, end plates, tube plates, drums, headers and chambers [M)],
- heated and non-heated tubes [M)],
- furnaces and elements of combustion chambers[M)],
- long and short stays, girders[M)],
- bodies of mountings and fittings for working pressure 0.7 MPa and more [M)].

8.3.2 The following essential parts of pressure vessels and heat exchangers are subject to supervision in the process of construction for compliance with the approved documentation:

- shells, distributors, end plates, headers and covers [M)],
- tube plates[M)],
- tubes[M)],
- long and short stays, fastenings[M)],
- bodies of valves for working pressure 0.7 MPa and more, 50 mm and over in diameter [M)].

Notes and explanations:

[M)] – material for parts of boilers, pressure vessels and heat exchangers of classes I and II (see 8.1) should be accepted by PRS.

9 BOILERS

9.1 Boiler Design

9.1.1 Boilers shall be designed so as to ensure their reliable operation in the conditions specified in sub-chapter 1.16 of *Part VI – Machinery Installations and Refrigerating Plants*.

9.1.2 The tubes thickness reduced in the process of bending is not to be less than the design thickness.

9.1.3 The use of long and short stays and of stay tubes in places where they are exposed to bending or shearing stresses, shall be avoided. Stays, strength walls, stiffeners, etc., shall have no abrupt changes in cross sections.

Inspection holes shall be provided at short-stay ends, as shown in item 5.3 of the Annex 1.

9.1.4 For walls reinforced by short stays and exposed to flame and high-temperature gases, the distance between stay centres is not to be greater than 200 mm.

9.1.5 In fire-tube boilers, corner stays shall be arranged at a distance of not less than 200 mm from the furnaces. Where flat walls are stiffened with welded girders, this should be done so that the load involved is transferred, as far as possible, directly onto the boiler shell or the most rigid of its parts.

9.1.6 Branch pieces installed in the boiler shall be of rigid construction and of minimum length sufficient for fixing and dismantling boiler mountings and fittings without removing the insulation. Branch pieces are not to be subjected to excessive bending stresses and shall be reinforced by stiffening fins if so required.

9.1.7 Flanges intended for installation of mountings, fittings and piping, as well as branches and sleeves passing through the entire thickness of the boiler wall shall be attached by welding to boiler shell, preferably from both sides. Branch pieces may also be welded from one side, using backing strip or by some other method that ensures penetration throughout the entire thickness of the boiler wall.

9.1.8 Boiler drums and headers of wall thickness greater than 20 mm, as well as steam superheater headers shall be protected from direct heat radiation, unless conditions specified in 8.2.3.4 are met. It is recommended that the gas uptake pipes of vertical fire-tube boilers passing through the steam space of the boiler be protected from direct exposure to hot gases.

9.1.9 Where non-metal sealing gaskets are used for closures of manholes and other openings, the design should prevent the possibility of gaskets being forced out.

9.1.10 Suitable design provisions shall be made to prevent steam formation in economizers of boilers.

9.1.11 A name-plate including all principal particulars of the boiler shall be provided in a visible place on the boiler.

9.1.12 The fastening elements on boilers, except for the elements not being under load, are not to be welded directly to the boiler shells but shall be attached to the welded pads.

9.1.13 Expanded tubes on headers and tube plates shall be of seamless type.

9.1.14 Boilers with finned pipes should be accessible for inspection from the flame side and shall be fitted with effective soot blowers.

9.2 Boilers Mountings, Fittings and Gauges – General Requirements

9.2.1 All boiler mountings shall be fitted on special welded branches, nozzles or pads, and be secured to these, as a rule, by flanged joints. The studs shall have a full thread holding in the pad for a length at least one external diameter of the thread. Screwed joints are allowed for mountings and fittings in a range of bores up to 15 mm.

The design of welded pads, branches and nozzles shall comply with the requirements of 8.2.19.

9.2.2 The valve covers shall be secured to valve casings by studs or bolts. Valves with bore diameters of 32 mm and less may have screwed joints provided that there are reliable means preventing them from being loosened.

9.2.3 The valve covers and cocks shall be fitted with ”on” and ”off” position indicators. Position indicators are not required where the design allows to see without difficulty whether the fittings are open or shut.

All valves shall be so designed as to be capable of being shut with clockwise motion of the wheels.

9.3 Feed Valves

9.3.1 Each auxiliary boiler for essential services (see sub-chapter 1.2, *Part VI – Machinery Installations and Refrigerating Plants*) shall be equipped with at least two feed valves. Auxiliary boilers for other services, and also exhaust-gas heated steam boilers may have one feed valve each.

9.3.2 The feed valves shall be of non-return type. A shut-off valve shall be installed between the feed valve and the boiler. The non-return and shut-off valves may be housed in one casing. The shut-off valve shall be fitted directly to the boiler.

9.3.3 The requirements concerning the feed water system are given in Chapter 18, *Part VI – Machinery Installations and Refrigerating Plants*.

9.4 Water Level Indicators

9.4.1 Every boiler with a free water surface (evaporating surface) shall be provided with at least two independent water level indicators with reflecting glass (see 9.4.3).

Subject to agreement with PRS, one of them may be replaced by:

– suitable safety and indication means of lower and upper water level; (safety and indication sensors shall be independent), or
– remote, placed at lower position, independent water level indicator of an approved type.

Boilers of a capacity below 750 kg/h, as well as all steam heated steam generators and exhaust-gas heated steam boilers with free water evaporating surface and steam receivers (steam separators) may be provided with single water level indicators with reflecting glass.

9.4.2 Forced circulation boilers shall be provided, instead of water level indicators, with two independent alarms to signal a shortage of water flow. The second alarm is not nedeed where the control systems are fitted on the boiler to satisfy the requirements of 4.2 and 4.3 of Table 21.3.1-1, *Part VIII – Electrical Installations and Control Systems*. This requirement does not apply to exhaust-gas heated steam boilers.

9.4.3 Flat prismatic reflecting glass shall be used in water level indicators. For boilers having a working pressure of 3.2 MPa and upwards, sets of mica sheets shall be used instead of glass, or else plain glass with a mica layer to protect the glass from water and steam effects, or some other materials resistant to destructive action of the boiler water.

9.4.4 The water level indicators shall be fitted vertically on the boiler front, possibly close to the boiler shell and at an equal and possibly shortest distance from the vertical centre plane of the boiler drum.

9.4.5 Water level indicators shall be provided with shut-off valves both on the water and steam side. The design of the shut-off valve shall provide for the safe cut-off of water flow in case of glass crack.

9.4.6 Water level indicators shall have the possibility of separate blowing-off the water and steam spaces. Blow-down valves shall have an inside diameter of not less than 8 mm. The design of water level indicator head shall prevent the gasket material from being forced into the ducts by the boiler pressure and shall allow for replacing the glasses while the boiler is in operation.

9.4.7 Water level indicators shall be so installed that the lower edge of slot in the indicator frame is positioned at least 50 mm below the lowest water level in the boiler, the centre line of indicator frame slot (centre of sight) being above the lowest water level.

9.4.8 Water level indicators shall be connected to the boiler by means of independent branch pipes. No tubes leading to these branches are allowed inside the boiler. The branches shall be protected from exposure to hot gases, radiant heat and intense cooling.

If the gauge glasses are fitted on hollow casings, the space inside such gauges shall be divided by partitions.

Water indicators and their branch pipes are not allowed to carry nozzles or branch pieces to be used for other purposes.

9.4.9 The branch pieces for attachment of water level indicators to boilers shall have an inside diameter not less than:
20 mm – for bent branches of auxiliary boilers,
15 mm – for straight branches of auxiliary boilers.

9.4.10 The design, dimensions, number, location and lighting of water level indicators shall provide for adequate visibility and reliable control of the boiler water level. Where water level visibility is inadequate, irrespective of the height of water level indicator location, or where the boilers are remotely controlled, provision shall be made in boiler control positions for reliable remote water level indicators (placed at lower position) or other types of water indicators approved by PRS. This requirement does not apply to exhaust-gas heated steam boilers and their steam receivers.

9.4.11 The indication error of the remote water level indicators is not to be greater than ±20 mm as compared to the indications of water level indicators fitted directly on the boiler. The difference in their simultaneous indications at the maximum possible rate of level changes is not to exceed 10% of the distance between the lower and the upper level.

9.5 Markings of the Lowest Water Level and Highest Heating Surface Points

9.5.1 Each boiler with free water surface (evaporating surface) shall have its lowest water level marked on the boiler water level indicator with a reference line drawn on the gauge frame or body. Additionally, the lowest water level shall be marked on a special plate in the form of horizontal reference line with inscription „lowest water level". The plate shall be fitted to the boiler shell near to the water level indicators.

The lowest level reference line, as well as the plate are not to be covered with boiler insulation.

9.5.2 The lowest water level in the boiler shall be situated not less than 150 mm above the highest heating-surface point. This distance is also to be maintained with the ship list up to 5° to any side and under all possible trims in normal service conditions.

In the case of boilers with design capacity less than 750 kg/h, the said minimum distance between the lowest water level and the highest point of heating surface may be reduced to 125 mm.

9.5.3 The position of the upper ends of the uppermost downcomers is assumed to be the highest point of heating surface of water-tube boilers.

For vertical fire-tube boilers with the fire tubes and gas uptake pipes passing through the steam space of the boiler, the determination of the highest heating-surface point will be specially considered by PRS in each particular case.

9.5.4 Each fire-tube boiler shall be fitted with a position indicator for the highest heating-surface point, which shall be attached to the boiler wall close to the lowest water-level plate, and to have an inscription ”highest heating-surface point”.

9.5.5 The requirements concerning the position of the highest heating-surface point and the relevant position indicator do not apply to exhaust-gas heated steam boilers, forced circulation boilers, economizers and steam superheaters.

9.6 Pressure Gauges and Thermometers

9.6.1 Each boiler shall have at least two pressure gauges connected to the steam space by separate pipes fitted with stop valves or stop cocks. Three-way valve or cock shall be provided between the pressure gauge and pipe, thus making it possible to shut off the pressure gauge from the boiler, blow off the connecting pipe with boiler steam and install the control pressure gauge.

9.6.2 One of the pressure gauges shall be installed on the front of the boiler and the other in main engine control station.

9.6.3 Boilers with design capacity below 750 kg/h and exhaust-gas heated steam boilers are allowed to have one pressure gauge.

9.6.4 A pressure gauge shall be provided at the feed water economizer.

9.6.5 Pressure gauges shall have a scale sufficient to allow for boiler hydraulic testing.

The pressure gauge scale shall have a red line to mark the working pressure in the boiler.

9.6.6 Pressure gauges shall be installed on the boilers in such a way that they are suitably protected from the heat emitted by non-insulated boiler surfaces.

9.6.7 Steam superheaters and economizers shall be equipped with thermometers of suitable scale range.

Where remote temperature control is installed, the local thermometers are also to be fitted.

9.7 Safety Valves

9.7.1 Each boiler shall have not less than two spring-loaded safety valves of identical construction and equal diameter of cross-sectional area, to be installed on the boiler drum, as a rule, on a common branch piece; additionally one valve shall be fitted on the steam superheater outlet header. The superheater safety valve shall be so adjusted as to open before the safety valve installed on the drum.

Safety valves of non direct-acting type are recommended for steam boilers having a working pressure of 4 MPa and more.

One safety valve is sufficient for steam boilers with design capacity below 750 kg/h, as well as for steam reservoirs (steam separators).

9.7.2 The aggregate cross-sectional area of safety valves is not to be less than that determined by the formula:

for saturated steam

$$f = K\frac{G}{10.2\,p_w + 1}\ , \quad [\text{mm}^2] \qquad (9.7.2\text{-}1)$$

for superheated steam

$$f = K\frac{G}{10.2\,p_w + 1}\sqrt{\frac{v_H}{v_s}}\ , \quad [\text{mm}^2] \qquad (9.7.2\text{-}2)$$

f – aggregate cross-sectional area of safety valves, [mm^2];
G – design capacity of boiler steam, [kg/h];
p_w – working pressure, [MPa];
v_H – specific volume of superheated steam at the appropriate working pressure and temperature, [m^3/kg];
v_S – specific volume of saturated steam at the appropriate working pressure, [m^3/kg];
K – factor as per Table 9.7.2.

Table 9.7.2

Valve lift	*K* factor
$\frac{d}{20} \le h < \frac{d}{16}$	22
$\frac{d}{16} \le h < \frac{d}{12}$	14
$\frac{d}{12} \le h < \frac{d}{4}$	10.5
$\frac{d}{4} \le h < \frac{d}{3}$	5.25
$\frac{d}{3} \le h$	3.3

d – minimum diameter of valve, [mm];

h – valve lift, [mm].

Safety valves are not to be less than 32 mm or more than 100 mm in diameter.

If specially approved by PRS, the use of safety valves with smaller areas than required by formulae 9.7.2-1 and 9.7.2-2 may be allowed, provided it will be proved experimentally that each of these valves has a discharge capacity not lower than the design steam capacity of the boiler.

9.7.3 The cross-sectional area of the safety valve installed on the permanently fixed superheater may be included in the aggregate area of the valves to be determined by formulae 9.7.2-1 and 9.7.2-2. This area is not to amount to more than 25% of the aggregate cross-sectional area of the valves.

9.7.4 The safety valves shall be so adjusted that the valve operating pressure does not exceed the design pressure. Safety valves of auxiliary boilers of essential services, after being lifted shall stop the steam escape before the pressure falls below 0.85 of the working pressure.

9.7.5 Each flue gas heated economizer shall be provided with spring-loaded safety valve not less than 15 mm in diameter.

9.7.6 Where safety valves are fitted on a common branch, the cross-sectional area of the branch is not to be less than 1.1 times the aggregate cross-sectional area of the valves installed thereon.

9.7.7 The cross-sectional area of the outlet steam branch of each safety valve, as well as of the pipe connected thereto, is not to be less than twice the aggregate cross-sectional area of the valves.

9.7.8 To remove the condensate, a drain pipe without any stopping devices shall be provided on the valve body or on the outlet steam pipe if it is located below the valve.

9.7.9 The safety valves shall be connected directly to the boiler steam space without any stopping devices.

No pipes leading to the safety valves may be installed inside the boiler, nor can any provision be made on the safety valve bodies or their branches for steam extraction devices or for other devices.

9.7.10 The safety valves shall be so arranged that they can be lifted by a special hand-operated easing gear. The easing gear shall be operated from the boiler room, and from the upper deck or any other readily accessible place outside the boiler room. The safety valves of steam superheaters, exhaust gas heated steam boilers and their steam receivers (separators) may be controlled only from the boiler room.

9.7.11 The safety valves shall be so designed that they can be sealed or provided with an equivalent safeguard to make it impossible for the valves to be readjusted by unauthorised persons.

The springs of the safety valves shall be protected from direct exposure to steam; these springs, as well as the sealing surfaces of seats and valves shall be made of heat- and corrosion-resistant materials.

9.8 Shut-off Valves

9.8.1 Each boiler shall be separated from all pipelines leading to it by means of shut-off valves secured directly to the boiler.

9.8.2 Where there is only one auxiliary boiler of essential services installed on board, the superheater and economizer shall be so arranged as to be capable of being shut-off from the boiler.

9.9 Blow-down and Scum Valves

9.9.1 Boilers shall be fitted with blow-down and scum arrangements and, where necessary, with drain valves.

Blow-down, scum and drain valves shall be fitted directly to the boiler shell. For boilers of working pressure below 1.6 MPa, these valves may be installed on welded-on branch pieces.

Steam superheaters, exhaust gas heated economizers and steam headers shall be provided with blow-down valves or drain valves.

9.9.2 The inside diameter of desludging valves and pipes is not to be less than 20 mm but not more than 40 mm. For boilers with design capacity below 750 kg/h, the inside diameter of the valves and pipes may be reduced down to 15 mm.

9.9.3 Boilers with a free water surface (evaporating surface) shall be provided with scum arrangements ensuring scum and sludge removal from the entire evaporating surface.

9.10 Salinometer Valves

Each boiler shall be provided with at least one salinometer valve or cock. Installing such valves or cocks on pipes and branches intended for other purposes is not allowed.

9.11 Deaeration Valves

Boilers, superheaters and economizers shall be equipped with sufficient number of valves or cocks for deaeration.

9.12 Openings for Internal Inspection

9.12.1 Boilers shall be provided with manholes for inspection of all internal surfaces. Where the arrangement of manholes is not possible, provision shall be made for sight holes.

9.12.2 Manhole opening dimensions shall be not less than:
300 × 400 mm – for oval openings, or
400 mm – for round openings.

In individual cases, if specially approved by PRS, the dimensions of manhole openings may be reduced to 280 x 380 mm for oval and 380 mm for round openings.

The oval manhole openings in cylindrical shells shall be positioned in a way that the minor axis of the manhole is arranged longitudinally.

9.12.3 Vertical fire-tube boilers shall have at least two sight holes arranged opposite to each other in the area of the water working level.

9.12.4 All boiler parts such as may prevent or hinder free access to and inspection of internal surfaces shall be of removable type.

9.13 Oil Fuel Installations of Boilers

9.13.1 All the components of oil fuel installation such as pumps, fans, quick-closing valves and electric drives, shall be of type approved by PRS.

Electric equipment, adjustment, control, safety and alarm systems shall be made in compliance with applicable requirements of *Part VIII – Electrical Installations and Control Systems.*

Piping systems and fittings of oil fuel installation shall comply with applicable requirements of *Part VI – Machinery Installations and Refrigerating Plants.*

9.13.2 The requirements of the present sub-chapter concern the equipment for firing the boilers with fuel oil of flash point not less than 60 °C.

9.13.3 The burners shall be so designed as to ensure the possibility of controlling the size and shape of the flame jet.

9.13.4 In the case of variable-delivery burners, controlling the amount of combustion air shall be ensured.

9.13.5 A design solution shall be applied excluding the possibility of turning and removal of the burners from their positions before cutting-off the fuel supply.

9.13.6 Where the fuel is atomized by means of steam or air, the design should preclude the possibility of penetration of steam or air to fuel oil and vice versa.

9.13.7 Where fuel preheating is applied, the design solution should preclude the possibility of excessive fuel heating when the boiler capacity is reduced or burners are cut off.

9.13.8 Proper drip trays shall be provided where fuel leaks may be expected.

9.13.9 Proper sight glasses shall be provided for observation of combustion process in the furnace. Devices preventing flame and hot air outburst in the case of burner dismantling shall be provided.

9.13.10 Proper arrangement for storage and smothering the manual ignition torch shall be provided.

It is recommended that the inlets of boiler fans be protected against penetration of moisture and solids.

9.13.11 Additional Requirements for Permanently Attended Boilers with Automatic Firing Control

9.13.11.1 The firing installations of boilers shall be interlocked for fuel supply to the boiler furnace to be possible only under the following conditions:

.1 the burner is in the operating position,
.2 all electrical equipment is connected to electric power supply,
.3 air is fed to the boiler furnace,
.4 the pilot burner is alight or electrical ignition switched on,
.5 the water level in boiler is normal.

The shutdown of fuel supply is, in general, to be effected by two self-closing valves operating in succession. Where the daily service tank is situated below the furnace, one such valve is sufficient.

9.13.11.2 The firing installations of boilers shall be fitted with non-detachable protective devices to operate within 1 second maximum (in the case of a pilot burner within 10 seconds maximum) and automatically shut off fuel supply to the burners in the case of:

.1 loss or low combustion air pressure or decay of combustion air flow,
.2 burner flame failure,
.3 water level in the boiler reaching its lower limit.

The operation of protective devices shall actuate visual and audible alarms.

9.13.11.3 Firing installations shall be equipped with a burner flame jet monitor. Such monitor shall respond only to the flame of the burner under control.

9.13.11.4 The capacity of the pilot burner shall be such that the burner by itself is not capable of maintaining the boiler under working pressure even with the steam consumption stopped.

If the pilot burner and the main burner are simultaneously in operation and the safety system is activated under conditions specified in 9.13.11.2, the both burners shall stop their functioning at the same time.

9.13.11.5 Firing installation of the auxiliary boilers of essential services shall be capable of being started up and controlled manually. The manual control devices shall be located as near to the boiler as possible**.**

While the firing installation is manually controlled, all the automatic devices listed in 9.13.11.1 and 9.13.11.2 shall be operative.

9.13.11.6 Provision shall be made for the firing installation to be shut off from two different stations, one of which shall be situated outside the boiler room.

9.14 Control, Safety and Alarm Systems of Boilers

9.14.1 The requirements of the present Chapter apply to permanently attended boilers.

The requirements for control systems of not permanently attended boilers are included in sub-chapter 20.7 and Chapter 21 of *Part VIII – Electrical Installations and Control Systems*.

9.14.2 Control Systems

9.14.2.1 The auxiliary water-tube boilers of essential services shall be provided with feed and combustion automatic control systems.

It is recommended that other boilers be also provided with such control systems.

9.14.2.2 The control systems shall be capable of maintaining the water level and other variable parameters within the predominated limits over the entire load range and to ensure quick changes of boiler load.

9.14.3 Safety Systems

9.14.3.1 The boilers shall be provided with non-detachable system ensuring that the water level in the boiler (see 9.5) does not fall beneath the lowest permissible level.

9.14.3.2 The boilers with automatic control of combustion shall be provided with a safety system in accordance with 9.13.11.

9.14.4 Alarm Systems

9.14.4.1 Boilers with automatic control of feed and firing shall be provided with audible and visual alarm at the control stand.

9.14.4.2 The audible and visual alarms shall be activated in the case of:
- the water level reaching its lowest limit,
- the water level reaching its highest limit,
- failures in the automatic control and safety systems,
- failures in the boiler firing installations (see 9.13.11.3),
- salinity of feed water exceeding the permissible level (see also 18.2.4 of *Part VI – Machinery Installations and Refrigerating Plants*).

9.14.4.3 The lowest water level alarms of the auxiliary boilers of essential services shall be activated prior to the activation of the safety system.

9.14.4.4 Provision shall be made for the audible alarm to be switched off manually after announcing the signal.

9.15 Incinerating Boilers

9.15.1 These provisions apply to naval ships' auxiliary boilers used for incinerating garbage and oil wastes of flash point above 60 °C.

9.15.2 Automatic control systems of not permanently manned incinerating boilers and the systems elements shall comply with the requirements of Chapter 20 of *Part VIII – Electrical Installations and Control Systems*.

9.15.3 Special furnace chamber shall be provided for incineration of garbage and oil wastes; the chamber shall comply with the following requirements:

.1 the chamber shall be entirely separated from the boiler furnace and lined with material resistant to chemical effects of incinerated products;

.2 ducts interconnecting the furnace with chamber shall have sufficient cross-sectional area. In all the cases the working pressure in the chamber is not to exceed the furnace pressure by more than 10%;

.3 a safety device, activated when the working pressure is exceeded by 0.02 MPa, shall be provided preventing outburst of flame into the boiler-engine room;

.4 aggregated free cross-sectional area of the safety device shall be not less than 115 cm^2 per 1 m^3 of the chamber volume and not less than 45 cm^2;

.5 the chamber charging device shall be such as to prevent the simultaneous opening of the chamber and furnace. Any limitations concerning the garbage incinerating shall be posted on the warning plate;

.6 chambers provided for incineration of garbage only can be installed in the boiler furnace;

.7 if no garbage dump bunker is provided, the chute cover shall be provided with locking device preventing its opening in case the temperature inside the chamber could cause self-ignition of the garbage.

9.15.4 Oil wastes are, in general, to be incinerated in special system designed for this purpose. It is possible to use for this purpose the boiler firing system including the burner, provided that smokeless incineration is ensured as far as possible.

9.15.5 Incinerating boilers shall be provided with effective system of soot removal.

9.16 Thermal Oil Heaters

9.16.1 Thermal oil heaters are, in general, to be installed in separate spaces, equipped with exhaust ventilation system, capable to perform at least 6 air changes per hour.

9.16.2 Thermal oil heaters shall be so designed as to eliminate a possibility of thermal oil overheating above its upper permissible temperature limit in the case its burners and thermal oil circulating pumps are stopped.

The maximum working temperature of given thermal oil shall be maintained at least 50 °C below its upper permissible temperature limit.

9.16.3 The construction of combustion chambers and burners shall secure a uniform heat distribution.

Only such non-uniformity of heat distribution may be admitted at which temperature in thermal oil boundary layer at any place of the heating surface does not exceed the upper permissible temperature limit for the thermal oil used.

The construction of combustion chamber and location of burners shall preclude a direct exposure of the heater surface to the flames. The burner shall be so designed as to eliminate the heat delivery above its nominal rate.

The combustion chambers of thermal oil heaters with the capacity of 1000 kW and more shall be provided with hermetisation devices and a separate smothering system of type approved by PRS.

9.16.4 Each thermal oil heater shall be fitted with:
– shut-off valves at inlet and outlet of thermal oil. Such valves shall be controlled from outside the compartment in which they are situated. Alternatively an arrangement for quick gravity drainage of the thermal oil, contained in the oil system, into a draining tank is acceptable (see also 14.6.2 of *Part VI – Machinery Installations and Refrigerating Plants*);
– pressure gauge;
– at least two spring-loaded safety valves of closed type, of identical construction and dimensions, the capacity of each one being not less than the capacity of circulating pump. The cross-sectional area of safety valves is not to be less than that equivalent to diameter 32 mm and not greater than that equivalent to diameter 100 mm;
– arrangements for taking samples of thermal oil;
– inspection openings according to 9.12.

9.16.5 Each thermal oil heater shall be equipped with effective means for soot removal.

9.16.6 Thermal oil heater tubes shall be connected to headers and chambers by welding.

9.16.7 Bellows type valves shall be applied to thermal oil boilers. Application of gland type fittings will be subject to special consideration by PRS.

9.16.8 Thermal oil heaters shall be provided with alarm and safety system activated at reaching limit temperatures of thermal oil and exhaust gas, fitted at the outlet of the heaters.

9.16.9 Thermal oil heaters shall be provided with automatic combustion control, audible and visual alarm, interlock device provided in 9.13.11.1, as well as protective device complying with 9.13.11.2.

9.16.10 Additional requirements for exhaust gas heated thermal oil heaters

9.16.10.1 Exhaust gas heated thermal oil heaters shall be fitted with devices closing the supply of hot gas to the heater in the case of safety system activation.

9.16.10.2 The heater shall be so designed and installed that all tubes can be easily and readily inspected for any signs of corrosion and leakage.

9.16.10.3 The heater shall be fitted with temperature sensor(s) and fire detection alarm system.

9.16.10.4 A fixed fire extinguishing and cooling systems shall be fitted. A sprinkler system of sufficient capacity may be accepted.

The exhaust duct below the exhaust gas heater shall be so arranged for adequate collection and drainage of any fluid as to prevent it from flowing into the diesel engine. The collected fluid shall be properly drained.

9.16.10.5 Only one safety valve may be installed on exhaust gas heated thermal oil heaters.

9.17 Water Heating Boilers

Design and construction materials of water heating boilers shall comply with the requirements for steam boilers.

10 PRESSURE VESSELS AND HEAT EXCHANGERS

10.1 Design of Pressure Vessels and Heat Exchangers

10.1.1 The components of pressure vessels and heat exchangers which come in contact with sea water or other corrosive media shall be made of materials resistant to corroding effects of the medium. If other materials are used, their protection against corrosion is subject to special consideration by PRS.

10.1.2 The design of pressure vessels and heat exchangers shall ensure their reliable operation under the conditions specified in sub-chapter 1.6 of *Part VI – Machinery Installations and Refrigerating Plants.*

10.1.3 Pressure vessels and heat exchangers shall comply with the requirements of 9.1.2, 9.1.3, 9.1.4, 9.1.7, 9.1.8, 9.1.10 as well as 8.2.14 and 8.2.19.

10.1.4 Where necessary, pressure vessels and heat exchangers shall be so designed as to allow for a thermal expansion of their shells and components

10.1.5 Supports shall be provided for attaching the shells of heat exchangers and pressure vessels to their foundations. Upper fastenings shall be provided if necessary.

When designing the fastening of pressure vessels and heat exchangers to the foundations, the requirements of sub-chapter 1.11 of *Part VI – Machinery Installations and Refrigerating Plants* shall be also taken into account.

10.2 Fittings and Gauges

10.2.1 Each pressure vessel and heat exchanger or their inseparable set shall be fitted with fixed safety valve. Where there are several non-interconnected spaces, safety valves shall be provided for each space. Hydrophore tanks shall be fitted with safety valves, which shall be installed on the waterside.

In particular cases, PRS may waive the above-mentioned requirements.

10.2.2 Safety valves are generally to be of a spring-loaded type. Safety diaphragms of a type approved by PRS are allowed for use in fuel and oil heaters, provided they are installed on the fuel and oil side.

10.2.3 The discharge capacity of safety valves shall be such that under no conditions the working pressure is exceeded by more than 10 %.

10.2.4 The safety valves shall be so designed as to be capable of being sealed or fitted with an equivalent safeguard to prevent the possibility of valve adjustment without the knowledge of the personnel in charge. The materials used for springs and sealing surfaces of valves shall be resistant to corroding effect of the medium.

10.2.5 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 design and shall have an adequate protection. Flat glass plates shall be used for level indicators containing steam, fuel, oil or refrigerants. Flat sight glasses shall be used in liquid indicators for steam, fuel, oil and refrigerants.

10.2.6 The design of pressure vessels and heat exchangers shall provide for flanges or flanged branch pieces for fastening the fittings.

Threaded branch pieces can be applied to hydrophore tank designs.

10.2.7 Pressure vessels and heat exchangers shall be equipped with blow-down and drainage devices.

10.2.8 Manholes shall be provided to enable inspection of the internal surfaces of pressure vessels and heat exchangers. Where provision of manholes is not possible, sight holes shall be fitted in suitable places. Where the pressure vessels or heat exchangers are over 2.5 m in length, sight holes shall be provided at both ends.

Manholes or sight holes are not required where the pressure vessel or heat exchanger is of dismountable construction or where possibility of corrosion and contamination of internal surfaces is completely ruled out.

Manholes or sight holes are not necessary in pressure vessels and heat exchangers whose construction excludes the possibility of inspection.

For the dimensions of manholes openings – see 9.12.2.

10.2.9 Each pressure vessel and heat exchanger, as well as their permanently connected units shall be equipped with a pressure gauge or a compound pressure gauge. Where heat exchangers have several spaces, a pressure gauge or a compound pressure gauge shall be provided for each space.

Pressure gauges shall comply with the requirements set forth in 9.6.1 and 9.6.5.

10.2.10 Fuel heaters where the fuel temperature may exceed 220 °C shall be fitted – apart from the temperature controls – also with sensor warning about high temperature or stopped flow of fuel.

For electric heaters – see also 15.4 of *Part VIII – Electrical Installations and Control Systems*.

10.3 Requirements for Particular Types of Pressure Vessels and Heat Exchangers

10.3.1 Air Receivers

10.3.1.1 The safety valves of starting air receivers of main and auxiliary diesel engines, as well as of fire protection systems, after being lifted, shall completely stop the air escape before the pressure inside the receiver falls below 0.85 of the working pressure.

10.3.1.2 Where the compressors, reducing valves or pipes from which air is supplied to receivers are provided with safety valves so adjusted that it is impossible to supply the receivers with air of pressure higher than the working pressure, safety valves need not be fitted on such air receivers. In such a case, fusible plugs shall be fitted on the receivers instead of the safety valves.

10.3.1.3 The fusible plugs shall have a fusion temperature within the range of 100-130 °C. The fusion temperature shall be punched out 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.

10.3.1.4 Each air receiver shall be equipped with a water-draining device. In the case of air receivers arranged horizontally, the water draining devices shall be provided at both ends of the receiver.

10.3.2 Cylinders for Compressed Gases

10.3.2.1 Cylinders for compressed gases are called portable pressure vessels specially manufactured for the storage of compressed gaseous refrigerants or CO_2, which are stored on board the ship for her operational purposes, but cannot be filled by means of the ship's equipment.

10.3.2.2 The strength calculations shall be made taking into account the requirements specified in 8.2.8 and the following:
– design pressure shall be not less than the pressure which may rise at temperature 45 °C, at the predetermined filling level;
– allowed stresses σ are determined according to 8.2.4, while the safety margin K to 8.2.5.1;
– allowance c for cylinders being exposed to corrosion shall be taken not less than 0.5 mm.

Use of steel with the yield stress greater than 750 MPa but not exceeding 850 MPa is possible only upon agreement with PRS.

10.3.2.3 Non-disconnectable safety devices of approved design shall be provided to prevent a dangerous pressure rise in the cylinder in case of temperature increase. The use of 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, is allowed.

10.3.2.4 Cylinders shall be permanently marked to include the following information:

.1 name of the manufacturer,
.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.

10.3.2.5 Cylinders shall be hydraulically tested to the pressure equal to 1.5 times the working pressure.

10.3.2.6 Cylinders which are manufactured specifically for storing the compressed gases, refrigerants or extinguishing agents shall be approved by PRS.

10.3.3 Condensers

10.3.3.1 The construction of condensers and their location on board shall be such as to enable tube replacement.

The condenser shells are generally to be of steel welded construction.

Baffles shall be provided inside the condenser, at excess pressure steam inlets, to protect the tubes from the direct steam impact.

The tube attachments shall be so designed as to prevent sagging and dangerous vibration of the tubes.

10.3.3.2 The covers of the condenser water chambers shall be fitted with manholes in number and position as may be required for ensuring access to the tubes for the purposes of their expansion, packing replacement or plugging of any tube.

Cathodic protection shall be provided for the water chambers, tube plates and tubes to prevent electrolytic corrosion.

10.3.3.3 The condenser shall be of a design enabling the fastening of monitoring and measuring devices.

10.3.4 Pressure Vessels and Heat Exchangers of Refrigerating Installations

In case of the pressure vessels and heat exchangers of the refrigerating and fire extinguishing installations, the requirements of 10.1, 10.2, 10.3.2 i 10.3 shall be applied with the exception of the requirements of 10.3.3.3 and 10.3.3.4, whereas the requirements of 10.2.1 may be considered as the guidelines.

These pressure vessels and heat exchangers are also to comply with applicable requirements of *Part V – Fire Protection* and the requirements of Chapter 22 of *Part VI – Machinery Installations and Refrigerating Plants.*

10.4 Filters and Coolers

10.4.1 Filters and coolers of the main and auxiliary engines shall comply with the requirements for heat exchangers and pressure vessels, concerning the materials and design.

10.4.2 Oil fuel filters installed in parallel that enables their cleaning without cutting off the fuel oil supply to engines (duplex filters) shall be provided with arrangements minimising the possibility of a filter under pressure being opened by mistake.

10.4.3 The filters or filter chambers shall be provided with suitable 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.

11 THRUSTERS

11.1 Application

11.1.1 The requirements of the Chapter 11 shall be applied to all devices which are used to drive and control the naval ship or for manoeuvring and are referred to in this chapter as devices.

The requirements of this chapter cover in particular the following:

– azimuthing thrusters,
– cycloidal propellers,
– foldable and retractable devices,
– devices used for dynamic positioning of the ship,
– water jet drive.
– tunnel propellers.

11.1.2 The main thrusters (also referred to as „main devices") are considered devices dedicated for main propulsion and steering and for dynamic positioning of the ship. All other thrusters are considered as auxiliary.

11.2 General Requirements

11.2.1 If the ship is propelled by thrusters only, at least two separate devices with independent power supply shall be used. Failure of one of the devices should not render the other device inoperable.

A possibility of application of single water jet drive device shall be each time considered separately by PRS.

11.2.2 Devices shall withstand all loads at constant and temporary working conditions.

11.2.3 Those components of devices with rotating column that transfer torque or a force associated with rotation shall be calculated assuming maximal torque exerted by the hydraulic motor (which rotates the column) with a maximal pressure differential of hydraulic liquid or calculated to a maximum starting torque of electric motor which rotates the column. The components considered shall withstand the condition where the column rotation is blocked.

11.2.4 The suitable and reliable means of preventing sea water from penetrating into the internal parts of the device and the ship's hull shall be provided.

11.2.5 The dynamic seals, which prevent seawater from penetrating into the device or the ship's hull shall be of the type approved by PRS.

11.2.6 Inspection openings, which allow carrying out necessary periodical survey of the main parts of the thruster shall be provided.

11.2.7 Thrusters, which are mounted inside the vessel's hull in a way that allows their protrusion or rotation, shall be placed in a separate watertight compartment unless double sealing is provided according to 11.2.5 to prevent water ingress into the ship's hull. Alarm of water ingress between the seals and possibility to carry out the seals survey during docking the ship shall be ensured.

11.2.8 The nozzle design shall comply with the applicable requirements of *Part III – Hull Equipment*.

11.2.9 In case of the devices with a propeller fitted on the rotating column where reverse manoeuvre is effected by 180° column rotation, the time for such rotation should not exceed 30 s.

11.2.10 The main thrusters shall provide the thrust vector control from all the main drive remote control positions and from the thruster compartment. In each of the positions a possibility of indication of the propeller pitch and direction of the thrust vector, as well as means suitable to stop the drive immediately and communications with all other positions shall be provided. The means for immediate stopping the drive shall be independent of remote control system of the device.

11.3 Drive

11.3.1 Diesel engines driving thrusters directly shall comply with the requirements of the Chapter 2 of the *Rules.* The installations which serve the diesel engines shall comply with appropriate chapters of *Part VI – Machinery Installations and Refrigerating Plants*.

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

11.3.3 For main thrusters, a permanently connected spare hydraulic oil storage tank of capacity suitable for full oil exchange in at least one device shall be provided.

11.3.4 The electric motors (in whole power range) that are used for drive of the main thrusters shall be supervised by PRS during the manufacturing.

11.3.5 Electrically driven main thrusters are also to comply with the applicable requirements of Chapter 17 of *Part VIII – Electrical Installations and Control Systems*.

11.4 Gears and Bearings

11.4.1 The gears in main devices shall comply with the requirements of Chapter 4 of the present *Rules*.

11.4.2 The gears of auxiliary devices intended for short time operation may be calculated for a limited number of working hours. The calculations of these gears, done in accordance with valid standards, are subject to separate consideration by PRS.

11.4.3 The ultimate L10 life of roller bearings in main devices should be at least 20 000 hours.

11.4.4 The ultimate L10 life of roller bearings in auxiliary devices should be at least 5 000 hours.

11.4.5 The bearing of rotating column should allow axial load transfer in both directions.

11.5 Propulsion Shafts

11.5.1 Propulsion shafts shall comply with the requirements of Chapter 2, *Part VI – Machinery Installations and Refrigerating Plants* including the requirements for ice strengthening if they apply.

11.5.2 In case of torsional vibrations of propulsion shafts, the requirements of the sub-chapter 4.1, *Part VI* apply.

11.6 Propellers

11.6.1 Fixed pitch propellers and controllable pitch propellers shall comply with the requirements of the Chapter 3, *Part VI – Machinery Installations and Refrigerating Plants.*

11.6.2 Screw propellers of non-conventional shape and propellers of other types shall be subject of a separate consideration by PRS.

11.7 Water Jet Drive

11.7.1 The water jet drive design shall enable:

.1 putting the outlet nozzle by 30° to each side and putting the nozzle over within this range in not more than 8 seconds;

.2 readjusting deflectors from ”full ahead” to ”full astern” position within not more than 10 seconds.

11.7.2 The device water inlet shall be so designed as to preclude the possibility of stopping the water jet and cavitation on the inlet surface.

The inlet channel shall be positioned so as to prevent the air from penetrating into the channel.

11.7.3 The pump drive shaft shall be mounted so high as to ensure device start at the minimum draught of the ship.

11.7.4 Cavitation on the pump impeller vanes, conducive to damage of the device elements, shall be prevented for every standard operation mode of the device.

11.7.5 Design analysis of free vibrations of the pump impeller vanes shall be performed. In the case of exceeding permissible stresses due to resonance, the prohibited rotational speeds are not to occur within the range of 0.7 to 1.1 of maximum rotational speed of the impeller.

11.7.6 Deflectors shall be provided with mechanical shields preventing them from damage after ship's hitting the berth or a floating object.

11.7.7 Each water jet drive should be provided with its own hydraulic system independent of other ship's hydraulic systems.

11.8 Control Systems

11.8.1 Remote control systems of the thrusters shall comply with the requirements of 20.2, *Part VIII – Electrical Installations and Control Systems.*

11.8.2 In the case of the main thrusters, the requirements of 20.5.1 to 20.5.3; 20.5.8 and 20.5.10 to 20.5.15 of the *Part VIII* are obligatory. It is recommended that all other requirements of the Chapter 20, *Part VIII* to be considered.

11.9 Monitoring Systems

11.9.1 Indicating system shall comply with the requirements given in sub-chapter 20.4.3, *Part VIII. – Electrical Installations and Control Systems.*

11.9.2 Indicating system shall provide at remote control positions at least:
– display of direction of the rotation and its value for devices with fixed pitch propeller;
– display of the pitch and r.p.m. value in the case of controllable pitch propellers;
– display of thrust direction.

11.9.3 Alarm system shall comply with the requirements of sub-chapter 20.4.1 of the *Part VIII – Electrical Installations and Control Systems* and the requirements given in Table 11.9.3. The alarm system of auxiliary devices of installed power below 200 kW is a subject of separate consideration by PRS.

Table 11.9.3
Alarm system of thrusters

Item	Component installation, system	Parameter	Alarm system: parameter value signalised	Remarks
1.	Hydraulic drive of: – propeller, – device rotation, – propeller pitch change	level in spare hydraulic oil tank	minimal	–
2.		hydraulic oil pressure	minimal	–
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 a gravity tank	minimal	
6.	Electrical motor drive of: – propeller, – device rotation, – propeller pitch change	acc. to *Part VIII – Electrical Equipment and Automation,* Table 21.3.1-1, item 2.6		–
7.	Monitoring system of the thruster	alarm system power supply	minimal	–
8.		remote control system power supply	minimal	–
9.		emergency stop means acc. to 11.2.10	emergency stop	–
10.	Thruster compartment	fire detection	fire	–
11.		bilge water level in drainage well*	high level	–

* Water leak alarm system shall be used accordingly where possible and applicable.

11.10 Survey, Testing and Certificates

11.10.1 Thrusters shall be type approved by PRS.

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

In case of delivery of a single device, PRS may agree, after consideration of technical documentation, to application of the device, which has no type approval certificate.

11.10.3 Each thruster, mentioned in 11.10.1 and 11.10.2, shall be surveyed by PRS during its manufacturing and testing according to the requirements of 11.10.5 and 11.10.7.

11.10.4 The scope of survey of auxiliary devices with power installed below 200 kW shall be subject of separate consideration by PRS.

11.10.5 The survey during manufacturing and testing of the device covers:
- checking the conformity of the materials and technologies used with approved documentation,
- conformity check of manufacturing with the approved documentation,
- testing the device including pressure tests of housings, piping and fittings and the factory running tests. Testing must be conducted according to an approved tests programme. Factory running tests shall be done in the presence of the surveyor to PRS.

11.10.6 The following essential parts of thrusters are subject to supervision in the process of construction for compliance with the approved documentation:
- movable and stationary casings$^{M)}$,
- columns$^{M)}$,
- propeller shaft and intermediate shafts^{M},
- propellers$^{M)}$,
- nozzles,
- fastening elements, keys,
- pipings and fittings.

Notes and explanations:

$^{M)}$ – material approved by PRS. In case of auxiliary thrusters with rated power below 200 kW, the certificate of the material manufacturer is accepted. The material shall be inspected by the surveyor to PRS and the hardness test shall be carried out in his presence.

11.10.6.1 The pressure tests of housings shall be carried out according to 1.5.2.1. In case of hydrostatic pressure acting internally and/or externally, the working pressure *p* taken for calculations shall be the highest hydrostatic pressure acting on one side in the lowest point of the housing.

11.10.6.2 Factory tests shall be carried out on a test stand which allows to carry out the testing of the device at nominal r.p.m. and with propeller shaft and column, if applicable, loaded with full rotational torque. PRS may consider carrying out operational testing partly or wholly onboard the ship.

Operational testing includes:

.1 Start and stop testing of the drive, reversing tests;

.2 Operational testing of the device as a steering unit;

.3 Control systems testing.

11.10.6.3 After oprerational tests, the visual examinations of the whole device shall be carried out and, in justified cases, visual inspection of internal parts shall be conducted and, in particular, gears shall be examined.

The lubricating oil sample is also to be checked for traces of metallic and non-metallic particles.

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

11.10.7 The sea trials of a thruster shall be carried out according to an approved test schedule.

The ability of the device to provide the drive and steering in all considered modes of sailing and manoeuvring must be presented during the sea trials.

The trials shall be performed at different operational ship speeds, various positions and power settings of the device and during rapid manoeuvres which start with the most inconvenient combinations of vessel speed and position of the device.

11.10.7.1 In the case of the devices installed onboard for the first time, PRS may request to carry out measurements of the linear vibrations.

11.10.7.2 During the tests of the monitoring systems, compliance with the requirements defined in 11.10 shall be shown.

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

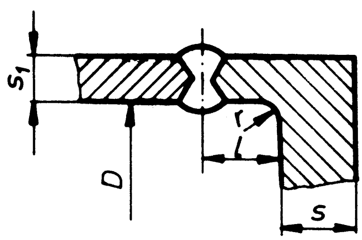
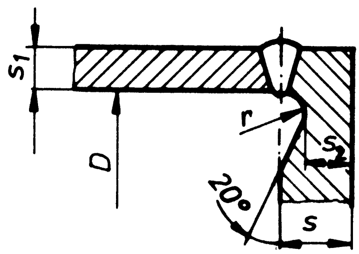
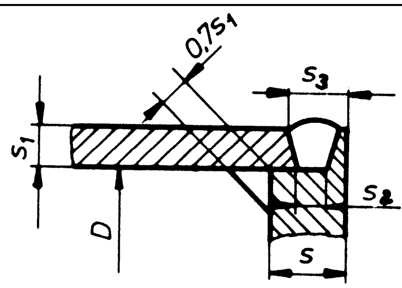
The sample of lubricating oil is also to be checked for content of solid metallic and non-metallic particles.

Annex 1

EXAMPLES OF WELDED JOINTS USED IN BOILERS, PRESSURE VESSELS AND HEAT EXCHANGERS

The scantlings of parts for welding and dimensions of welds shall be assumed in conformity with the national standards, corresponding to the welding method. The examples of the most frequently used welded joints are given in the tables of the present Annex.

According to the mechanical properties of the materials used and owing to the progress in welding technology, other welding techniques may be also used for performing the welded joints. The type of welded joint used in such cases and the necessary modifications of the joints shown in the tables shall be agreed with PRS.

Item	Drawing (example)	Application
1	2	3
1	**Flat end plates and covers**	
1.1.		$K = 0.38$ $r \geq \frac{s}{3}$, but not less than 8 mm $l \geq s$
1.2		$K = 0.45$ $r \geq 0.2\ s$, but not less than 5 mm $s_2 \geq 5$ mm (see Note 1)
1.3		$K = 0.5$ $s_2 \leq s_1$, but not less than 6.5 mm $s_3 \geq 1.24\ s_1$ (see Note 1)

Annex 2

SPARE PARTS

1 GENERAL REQUIREMENTS

1.1 The amounts and kinds of spare parts presented in tables are given for guidance. When determining the true number of spare parts, recommendations of manufacturers of the equipment installed onboard shall be taken into account.

1.2 The spare parts together with appropriate tools, materials, instruments and gauges shall be properly secured in easily accessible places and protected against corrosion.

1.3 It is recommended that the set of spare parts should provide one set of flexible joints of each type and size used onboard.

2 LIST OF SPARE PARTS

Table 2.1
Diesel engines [1)]

Item	Spare parts	Main engines		Auxiliary engines	
		Naval ships of unrestricted service	Naval ships of restricted service	Naval ships of unrestricted service	Naval ships of restricted service
1	2	3	4	5	6
1	Main bearings or their shells of each type and size fitted, complete with shims, studs (bolts) and nuts	1 set per bearing	–	1 set per bearing	–
2	Main thrust block (see Table 2.4)				
3	Cylinder liner complete with valves, joint rings and gaskets	1	–	Only joint rings and gaskets 1 set	–
4	Cylinder cover complete with valves, joint rings and gaskets. For engines without covers – the respective valves	1	–	Only joint rings and gaskets 1 set	–
4.1	Studs with nuts for securing cylinder covers	½ set per cover	–	–	–
5	Valves				
5.1	Exhaust valves (casings, seats, springs and other fittings)	1 set per cylinder	1 set per cylinder	1 set per cylinder	–

1	2	3	4	5	6
5.2	Air inlet valves complete with casings, seats, springs and other fittings	1 set per cylinder	1 set per cylinder	1 set per cylinder	–
5.3	Starting air valve complete with casings, seats, springs and other fittings	1	–	1	–
5.4	Relief valve, complete	1	–	1	–
5.5	Fuel injection valves of each type and size fitted, complete with all fittings	1 set per engine[2)]	1	½ set per engine	–
6	Connecting rod bearings				
6.1	Bottom end bearings or shells of each type and size fitted, complete with shims, bolts and nuts	1 set per cylinder	–	1 set per cylinder	–
6.2	Top end (crosshead) bearings or shells of each type and size fitted, complete with shims, bolts and nuts	1 set per cylinder	–	1 set per cylinder	–
7	Piston of each type and size fitted, complete with skirt, rings, studs and nuts	1	–	Gudgeon pin with bushes only per cylinder	–
8	Piston rings	1 set per cylinder	1 set per cylinder	1 set per cylinder	–
9	Hinged or telescopic cooling pipes of pistons with packings and other fittings	1 set per cylinder	–	1 set per cylinder	–
10	Lubricator of the largest size, complete with drive	1	–	–	–
11	Fuel pumps				
11.1	Fuel pump complete or, if parts are replaceable onboard, complete set of parts for one pump member (plunger, sleeve, valves, springs, etc.)	1	–	1	–

1	2	3	4	5	6
11.2	High pressure fuel pipe complete with unions of each size and type fitted	1	–	1	–
12	Turboblowers [3)]				
12.1	Rotor complete, shaft, nozzles and bearings	1 set	–	–	–

Notes to Table 2.1:

[1)] For monotypic engines (”monotypic engines” means engines whose similar parts are interchangeable), the tabulated minimum of spare parts is valid irrespective of the number of engines installed onboard.

[2)] For engines with one or two fuel injection valves in one cylinder – full number of complete fuel injection valves for one engine.
For engines with three and more fuel injection valves in one cylinder – two complete fuel injection valves for one cylinder, and for the remaining number of fuel injection valves of the engine – all parts except casings are recommended.

[3)] Not recommended for engines complying with the requirements of 2.4.1.

Table 2.2
Turbines for main drive [1)]

Item	Spare parts	Number of spare parts per ship	
		Ships of unrestricted service	Ships of restricted service
1	Injection valve, complete	1 set	1 set
2	Starting arrangement	1 set	1 set
3	Oil and fuel filter inserts	1 set	1 set

Notes to Table 2.2:

[1)] For monotypic turbines (”monotypic turbines” means turbines whose similar parts are interchangeable), the tabulated minimum of spare parts is valid irrespective of the number of turbines installed onboard.

Table 2.3
Gears and clutches [1)]

Item	Spare parts	Number of spare parts per ship	
		Ships of unrestricted service	Ships of restricted service
1	Main bearing bushes or shells of each type and size	1 set per bearing	–
2	Pads of the gear thrust bearing with a set of liners or thrust rings of each type and size with a set of liners for one bearing face [2)]	1 set	–
3	Inner or outer race with rollers (where fitted)	1 set	–

Notes to Table 2.3:

1) For monotypic gears and clutches (”monotypic gears and clutches” means gears and clutches whose similar parts are interchangeable), the tabulated minimum of spare parts is valid irrespective of the number of gears and clutches installed onboard.

2) When the pads of one face differ from those of the other one, complete set of pads shall be provided for each face.

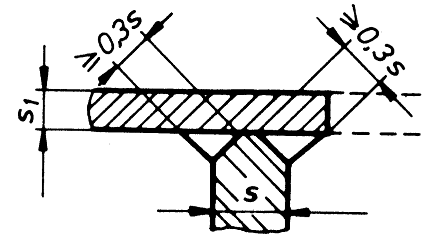
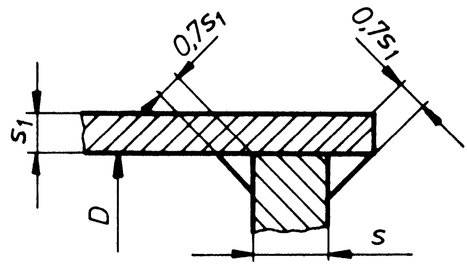
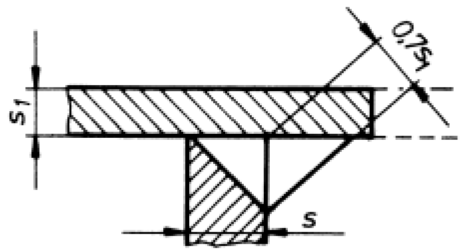
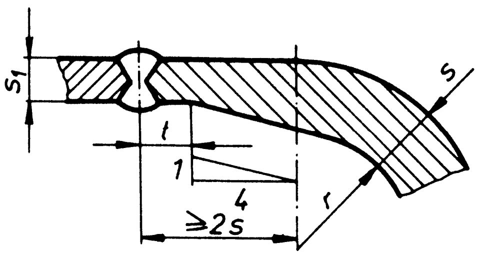
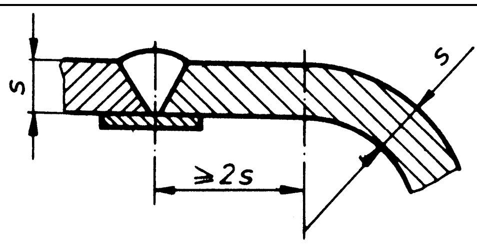
Table 2.4
Shafting and propellers

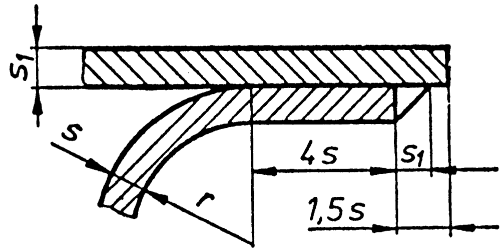
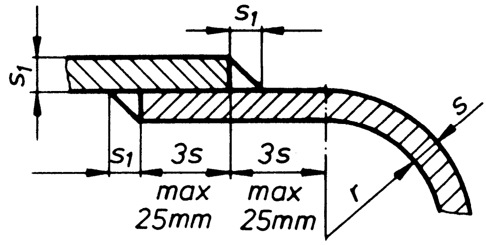
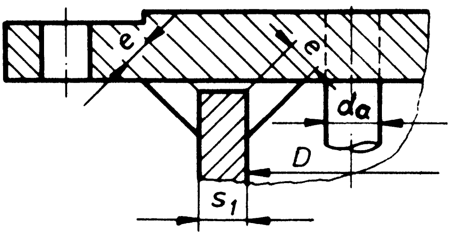
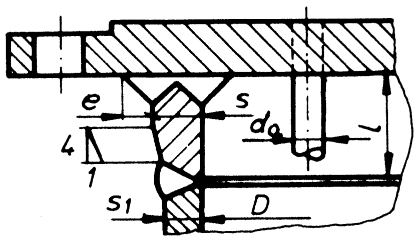
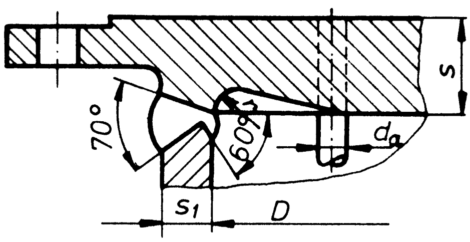
Item	Spare parts	Number of spare parts per ship	
		Ships of unrestricted service	Ships of restricted service
1	2	3	4
1	Shafting thrust bearings[1)]		
1.1	Thrust bearing pads for ahead running where pad type bearings are fitted	1 set	–
1.2	Thrust collars for ahead running where multiple collar bearings are fitted	1 set	–
1.3	Roller bearing where anti-friction type bearings are fitted	1 set	–
2	Propellers [2)]		
2.1	Cycloidal propeller blades complete with fastening elements	2 pcs per propeller	–
2.2	Bearings of blades, parts of pitch control gear and packing (rings, collars) for CP propellers and cycloidal propellers	1 set per propeller	–
2.3	Spare parts for gear of CP propellers, those of the cycloidal propellers and for units to serve systems other than specified under 2.1 and 2.2, depending on propeller construction	On agreement with PRS	–

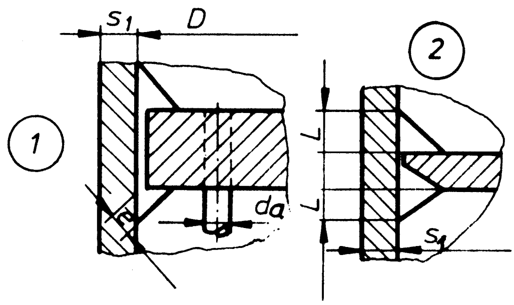
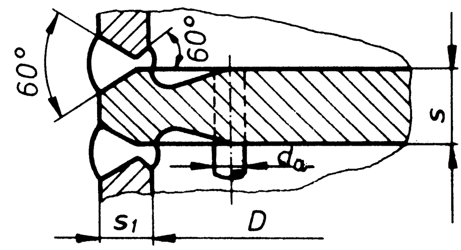
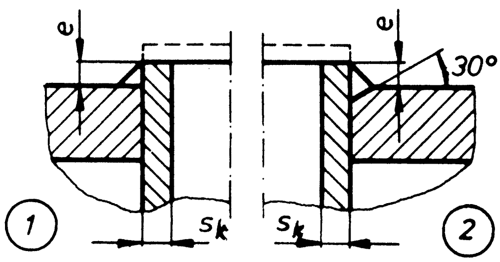
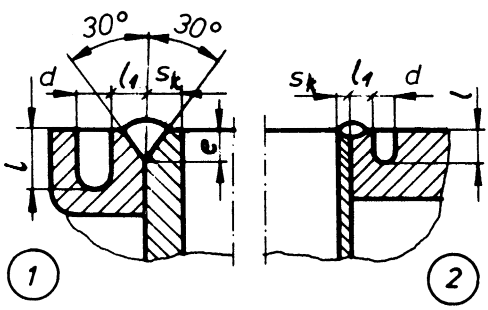
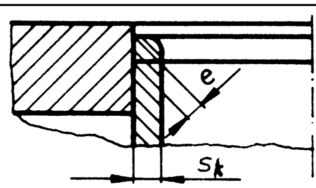
Notes to Table 2.4:

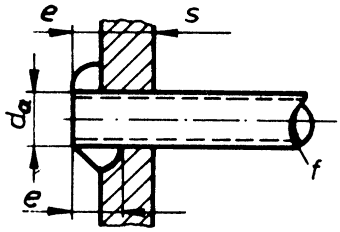
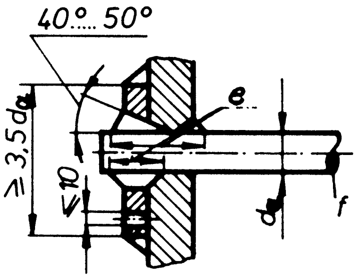
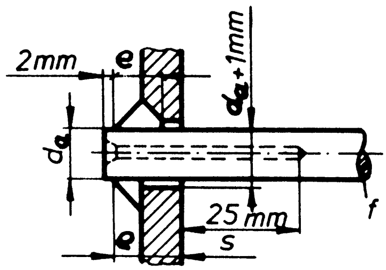
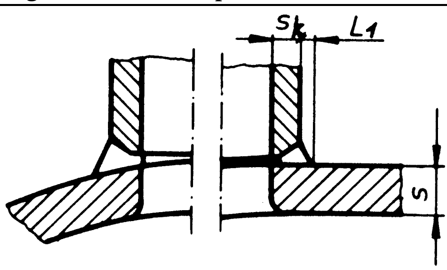
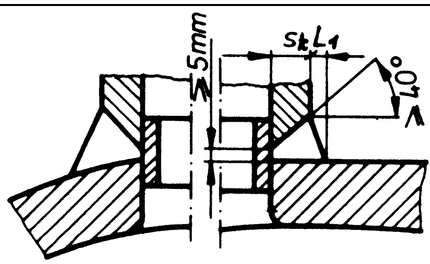
1) For bearings of one type the tabulated minimum of spare parts is valid irrespective of the number of bearings fitted onboard.

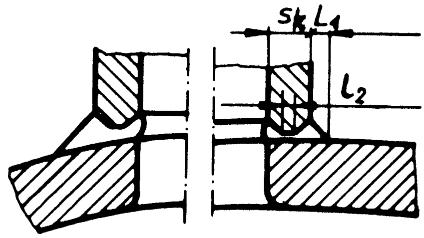
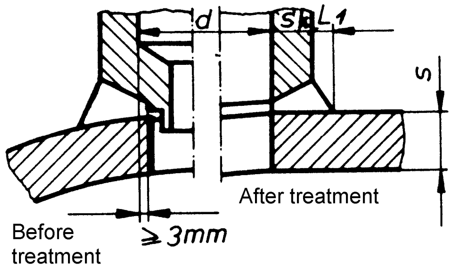
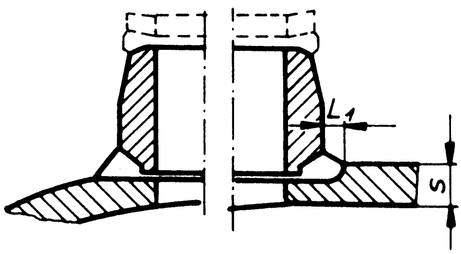
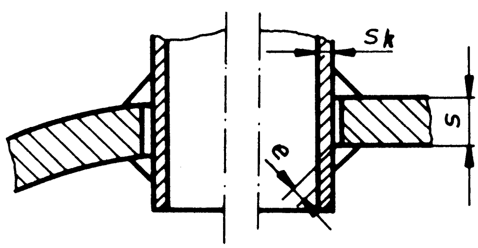
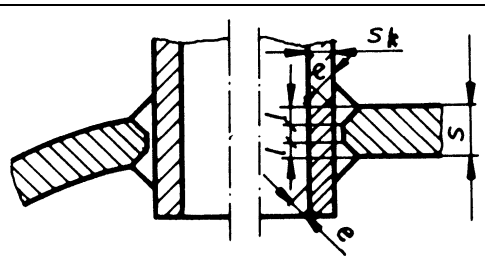
2) For naval ships with ice strengthenings.

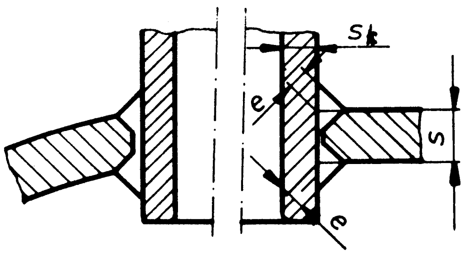
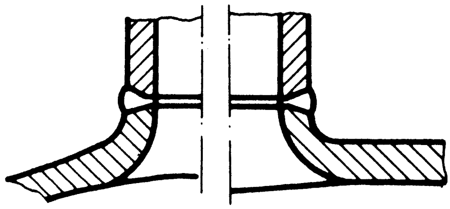
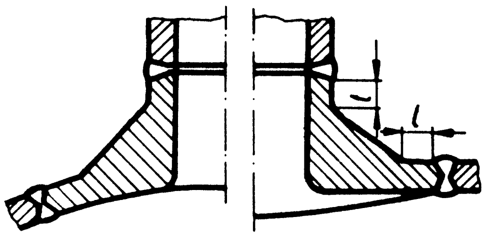
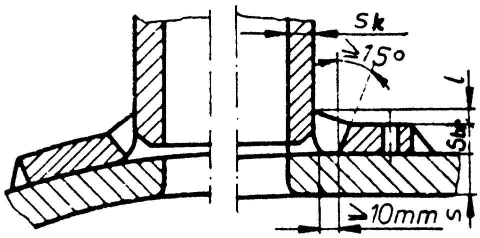
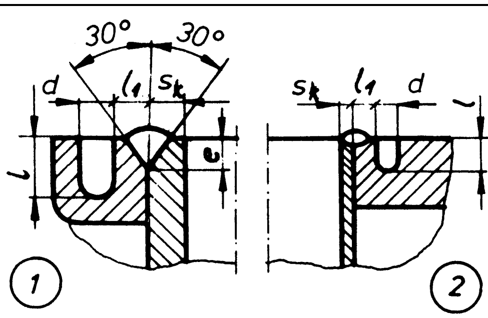
1	2	3
1.4		*K* = 0.45 (see Note 1)
1.5		*K* = 0.55 (see Note 1)
1.6		*K* = 0.57
2	**Dished ends**	
2.1		The joint may be used in boilers and pressure vessels of Class I, II and III (see Notes 2 and 17)
2.2		The joint may be used in boilers and pressure vessels of Classes II and III

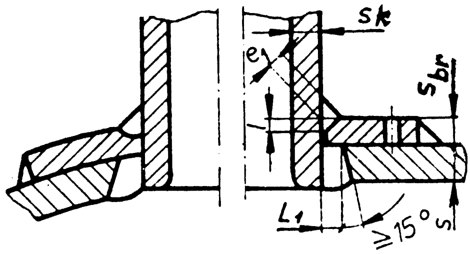
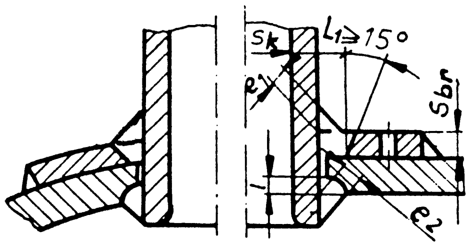
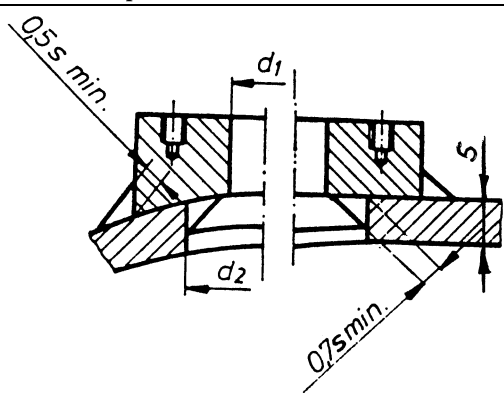
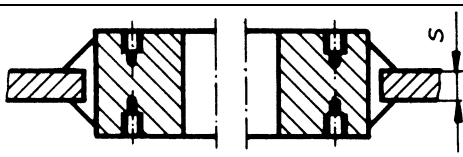
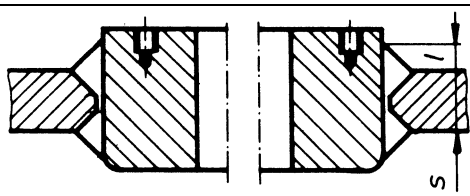
<table>
<tr><td>1</td><td>2</td><td>3</td></tr>
<tr><td>**2.3**</td><td></td><td>Not recommendable joint – it may be only used for pressure vessels of Class II not exposed to corrosion
$s_1 \leq 16$ mm
$D \leq 600$ mm</td></tr>
<tr><td>**2.4**</td><td></td><td>The joint may only be used for pressure vessels of Class III
$s_1 \leq 16$ mm
$D \leq 600$ mm</td></tr>
<tr><td>**3**</td><td colspan="2">**Tube plates**</td></tr>
<tr><td>**3.1**</td><td></td><td>$K = 0.45$
$e = 0.7\ s_1$
$s_1 \leq 16$ mm
(see Notes 3 and 4)</td></tr>
<tr><td>**3.2**</td><td></td><td>$K = 0.45$
$e = \frac{1}{3} s_1$
$e > 6$ mm
$s_1 > 16$ mm
(see Notes 5 and 6)</td></tr>
<tr><td>**3.3**</td><td></td><td>$K = 0.45$
$r \geq 0.2\ s$,
but at least 5 mm</td></tr>
</table>

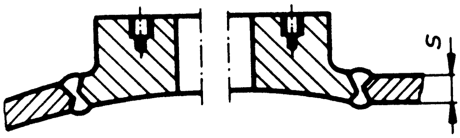
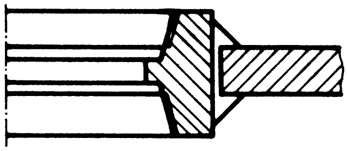
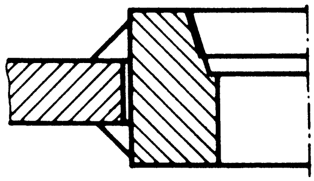
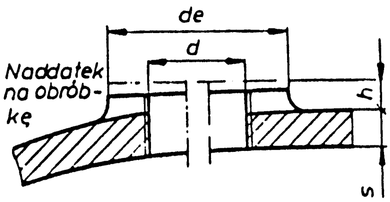
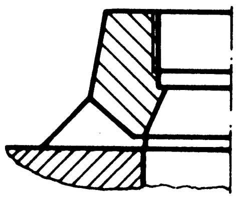
1	2	3
3.4		$k = 0.45$ $e \geq 0.7\ s_1$; when $L > 13$ mm the alternative 2 is recommendable, where $L = \frac{1}{3} s_1$ and $L \geq 6$ mm (see Note 7)
3.5		$K = 0.45$ $r \geq 0.2\ s$, but at least 5 mm
4	**Tubes**	
4.1		$e = s_k$ $e \geq 5$ mm $s_k \geq 2.5$ mm (see Notes 8, 9 and 10)
4.2		$d = s_r$; $l_1 = s_r$ $1.5\ s_r < l < 2\ s_r$ 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)

1	2	3
5	**Long and tube stays**	
5.1		$K = 0.42$
5.2		$K = 0.34$
5.3		$K = 0.38$ For short stays – see 9.1.3
6	**Branch pieces and connectors**	
6.1	**Non-through welded branch pieces**	
6.1.1		$s_k \le 16$ mm $L_1 = \frac{1}{3}s_k$, but not less than 6 mm
6.1.2		$L_1 = \frac{1}{3}s_k$, but not less than 6 mm (see Note 13)

1	2	3
6.1.3		$L_2 = 1.5 \div 2.5$ mm $L_1 \geq \frac{1}{3} s_k$, but not less than 6 mm (see Note 14)
6.1.4		$L_1 \geq \frac{1}{3} s_k$, but not less than 6 mm (see Notes 15 and 16)
6.1.5		$L_1 = 10 \div 13$ mm (see Note 15)
6.2	**Through welded branch pieces**	
6.2.1		The joint used, in general, for the values $s_k < \frac{1}{2} s$ $e = s_k$
6.2.2		The joint used, in general, for the values $s_k \cong \frac{1}{2} s$ $e = 6 \div 13$ mm $e + l = s_k$

1	2	3
6.2.3		The joint used, in general, for the values $s_k > \frac{1}{2}s$ $e \geq \frac{1}{10}s$, but not less than 6 mm
6.3	**Offset branch pieces**	
6.3.1		
6.3.2		see Note 17
6.4	**Branch pieces with reinforcing rings**	
6.4.1		$l \geq \frac{1}{3}s_k$, but not less than 6 mm
6.4.2		$l \geq \frac{1}{3}s_k$, but not less than 6 mm $L_1 \geq 10$ mm

1	2	3
6.4.3		$e + l = s_k$ or s_{br} (whichever less) $L_1 \geq 10$ mm
6.4.4		$e_2 + l \geq s_k$ $L_1 \geq 10$ mm $2s_k \leq (e_2 + l)$ plus $(s_{br} + e_1)$ or L_1 (whichever less of the latest)
6.5	**Pads and branch pieces with bolt holes**	
6.5.1		$d_2 \leq d_1 + 2s_{\min}$ (see Note 18)
6.5.2		$s \leq 10$ mm (see Notes 19 and 20)
6.5.3		$L \geq 6$ mm $s \leq 20$ mm

6.5.4		$s \geq 20$ mm
1	2	3
6.6	**Shell pads and branch pieces for screw joints**	
6.6.1		
6.6.2		
6.6.3	Allowance for machining 	$d \leq s$ $d_e = 2\ d$ $h \leq 10$ mm $h \leq 0.5\ s$ (see Note 21)
6.6.4		

Notes to the drawings:

1. The joint may be used in boilers of not more than 610 mm in diameter and in such pressure vessels 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 not more than 16 mm in thickness the joints may be single-side welded. The breadth of the ring is not to be less than 40 mm.
7. The distance between the internal shell diameter and the external tube plate-diameter shall be as small as possible.
8. The end of the tube projecting beyond the weld shall be milled or ground.

9. The spacing of the tubes is not to 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 is not to 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 is not to 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 possible and is in no case to 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 vessel shell and deposited welded material shall be sufficient for necessary number of thread turns.