Polski Rejestr Statków

RULES
FOR CLASSIFICATION AND CONSTRUCTION OF SEA-GOING SHIPS

CZĘŚĆ VII

MACHINERY, BOILERS AND PRESSURE VESSELS

2014

GDAŃSK

Polski Rejestr Statków

RULES
FOR CLASSIFICATION AND CONSTRUCTION OF SEA-GOING SHIPS

CZĘŚĆ VII

MACHINERY, BOILERS AND PRESSURE VESSELS

2014

GDAŃSK

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

Part I – Classification Regulations
Part II – Hull
Part III – Hull Equipment
Part IV – Stability and 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 IX – Materials and Welding.

Part VII – Machinery, Boilers and Pressure Vessels – 2014 was approved by the PRS Board on 30 December 2013 and enters into force on 1 January 2014.

From the entry into force, the requirements of *Part VII – Machinery, Boilers and Pressure Vessels* apply, in full, to new ships.

For existing ships, the requirements of *Part VII – Machinery, Boilers and Pressure Vessels* are applicable within the scope specified in *Part I – Classification Regulations.*

The requirements of *Part VII – Machinery, Boilers and Pressure Vessels* are extended by the below-listed Publications:

Publication No. 4/P – Inspection of Mass Produced Internal Combustion Engines,
Publication No. 5/P – Inspection of Mass Produced Exhaust Driven Turboblowers,
Publication No. 8/P – Calculation of Crankshafts for Diesel Engines,
Publication No. 28/P – Tests of I.C. Engines,
Publication No. 68/P – Type Testing Procedure for Crankcase Explosion Relief Valves.
Publication No. 69/P – Marine Diesel Engines – Control of Nitrogen Oxides Emission,
Publication No. 98/P – Guidelines Regarding the Requirements for Marine Diesel Engines Fitted with NO_x Selective Catalytic Reduction (SCR) Systems
Publication No. 103/P – Guidelines for Energy Efficiency of Ships.

PRS/OP, 01/2014
ISBN 978-83-7664-143-0

CONTENTS

1 GENERAL

1.1 Application

1.1.1 The requirements of this Part of the Rules – *Part VII – Machinery, Boilers and Pressure Vessels* apply to engines, machinery, boilers, pressure vessels and heat exchangers of sea-going ships classed by PRS.

1.1.2 The requirements for machinery apply to:

.1 I.C. engines and turbines for main propulsion;
.2 reduction gears, disengaging and flexible couplings of main propulsion system;
.3 I.C. engines and turbines of power generating sets and complete power generating sets;
.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.

1.1.3 The requirements for boilers, pressure vessels and heat exchangers apply to:

.1 steam boilers including exhaust gas boilers and steam superheaters of working pressure 0.07 MPa or more;
.2 heating oil boilers;
.3 water boilers with water temperature exceeding 115 °C;
.4 boiler economisers of working pressure 0.07 MPa or more;
.5 liquid fuel firing equipment of boilers;
.6 evaporators of main boilers and of important auxiliary boilers;
.7 condensers of main engines and auxiliary machinery;
.8 pressure vessels and heat exchangers containing in working conditions 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 or more;
.9 coolers and fuel, oil and water filters for main and auxiliary engines;
.10 air coolers and air heaters of working pressure in the air space 0.07 MPa or more.

1.1.4 Ergonomie Requirements

1.1.4.1 Engines, machinery components, boilers, pressure vessels and heat exchangers installed on board the ships classed with PRS covered with the requirements of this *Part* shall be so designed and arranged and shall be operated so as to ensure the compliance with occupational health and safety requirements and to ensure the seafarer comfort and capabilities with respect to ventilation, vibration, noise, means of access and egress taking account of the ambient conditions.

1.1.4.2 Detailed recommendations in this respect are contained in IACS publication Rec. No. 132 *Human element recommendations for structural design of lighting, ventilation, vibration, noise, access & egress arrangements.*

1.2 Definitions and Explanations

Definitions and explanations relating to general terminology of the *Rules for Classification and Construction of Sea-going Ships* (hereinafter referred to as the *Rules*) are specified in *Part I – Classification Regulations.* Where in *Part VII* definitions from other Rule parts are used, cross-reference to those parts is made.

For the purpose of *Part VII*, the following definitions have been adopted:

D e s i g n w a l l t e m p e r a t u r e – a temperature in the midst of the wall thickness used in calculation of allowable stress according to the ambient temperature and the heating conditions.

B o i l e r d e s i g n c a p a c i t y – a maximum hourly amount of steam that can be generated by the boiler at design parameters on continuous-duty runs.

1.3 Technical Documentation

1.3.1 General Requirements

The technical documentation including items listed below shall be forwarded to the Head Office of PRS for consideration and approval prior to equipment construction. The documentation shall be submitted in triplicate.

1.3.2 Documentation for Approval of I.C. Engines

1.3.2.1 The following documentation of I.C. engines shall be submitted to PRS for engine type approval:

.1 Data for crankshaft calculation in accordance with *Publication No. 8/P – Calculation of Crankshafts for I.C. Engines* **W**
.2 Engine transverse sectional drawing **W**
.3 Engine longitudinal section drawing **W**
.4 Drawing of bedplate and crankcase, cast or welded with welding details and instructions [9)] **W/Z**
.5 Thrust bearing assembly drawing[3)] **W**

.6	Drawing of thrust bearing bedplate, cast or welded with welding details and instructions [9]	**W/Z**
.7	Drawings of frame/framebox, cast or welded, with welding details and instructions [1), 9)]	**W/Z**
.8	Tie rod drawings	**W**
.9	Drawing of cylinder head assembly	**W**
.10	Drawing of cylinder liner	**W**
.11	Drawing of crankshaft with details for each number of cylinders	**Z**
.12	Drawing of crankshaft assembly for each number of cylinders	**Z**
.13	Drawing of thrust or intermediate shaft (if integral with the engine)	**Z**
.14	Drawing of coupling bolts	**Z**
.15	Drawing of counterweights with connecting bolts (if not integral with the crankshaft)	**Z**
.16	Drawing of connecting rod	**W**
.17	Connecting rod, assembly	**W**
.18	Drawing of crosshead assembly[2]	**W**
.19	Drawing of piston rod assembly[2]	**W**
.20	Drawing of piston assembly	**W**
.21	Drawing of camshaft drive assembly	**W**
.22	Material specifications of main parts with information on non-destructive material tests and pressure tests [8]	**Z**
.23	Arrangement of foundation bolts (for main engines only)	**Z**
.24	Schematic layout or other equivalent documents of starting air system on the engine [6]	**Z**
.25	Schematic layout or other equivalent documents of fuel oil system on the engine [6]	**Z**
.26	Schematic layout or other equivalent documents of lubricating oil system on the engine [6]	**Z**
.27	Schematic layout or other equivalent documents of cooling water system on the engine [6]	**Z**
.28	Schematic diagram of engine control and safety system on the engine [6]	**Z**
.29	Assembly drawing of shielding and insulation of exhaust pipes	**W**
.30	Shielding of high pressure fuel pipes, assembly [4]	**Z**
.31	Arrangement of crankcase explosion relief valve [5]	**Z**
.32	Operation and service manuals [7]	**W**
.33	Schematic layout or other equivalent documents of hydraulic system (for valve lift) on the engine	**Z**
.34	Type test program and type test report	**Z**
.35	High pressure parts for fuel oil injection system [10]	**Z**

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) All engines.
5) Only for engines of a cylinder diameter of 200 mm or more or a crankcase volume of 0.6 m^3 or more. Detailed requirements concerning documentation of explosion relief valves are given in paragraph 2.2.6.
6) And the system documentation so far as supplied by the engine manufacturer. Where engines incorporate electronic control systems, a failure mode and effects analysis (FMEA) shall be submitted to demonstrate that failure of an electronic control system will not result in the loss of essential services for the operation of the engine and that operation of the engine will not be lost or degraded beyond an acceptable performance criteria of the engine.
7) Operation and service manuals shall contain maintenance requirements (for servicing and repair), including details of special tools and gauges that shall be used with their fittings/settings together with any test requirements on completion of maintenance.
8) For comparison with PRS requirements for material, non-destructive testing (NDT) and pressure testing as applicable.
9) The weld procedure specification shall include details of pre- and post-weld heat treatment, weld consumables and fit-up conditions.
10) The documentation shall contain specification of pressures, pipe dimensions and materials.

Notes:

1. The documentation with code **Z** shall be approved by PRS.
2. The documentation with code **W** shall be submitted for reference but it may be the subject of certain requirements by PRS.
3. In case of the documentation with code **W/Z** the first letter is applicable to the cast structure and the second to the welded one.

1.3.2.2 Documentation of turbo-blowers, air coolers, etc. – see sub-chapters 1.3.3 and 1.3.5.

1.3.2.3 Updated documentation of the engine type is the basis for PRS survey of the engine manufacturing.

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

1.3.3 Documentation for Approval of Turbines

1.3.3.1 The following documentation shall be supplied to PRS for Turbine Type Approval:

.1 Technical description and basic technical specification, for gas turbines including diagram of power and rotations versus inlet air temperature **Z**

.2 Assembly drawings and sectional drawings with mounting dimensions, **Z**

.3 Drawings of casings, rotors, vanes and vane seals, attachments, seals bearings, burners and combustion chambers, heat exchangers integral with the turbine together with specification of the materials **Z**

.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 specification must be provided **W**
.5 Technical details of heat treatment for essential parts **W**
.6 Drawings of thermal insulation **Z**
.7 Foundation and attachment drawings **Z**
.8 Diagram of temperature field in the turbine at rated nominal power and at a maximum allowable short time power **W**
.9 Strength calculations of rotors, vanes and vane attachments **Z**
.10 Torsional vibration analysis[1)] and if applicable vanes vibration calculations **Z, tg**
.11 Turbine strength analysis for the whole operational life period of components that are highly loaded and work in highest temperature and taking into account creep behaviour and strength and high temperature corrosion **W, tg**
.12 Diagrams of rotation speed control system, alarm and safety system **Z**
.13 Detailed information regarding speed governors and safety controller **Z**
.14 Diagrams of lubrication and fuel system **Z**
.15 Rotor balancing procedure **W, tg**
.16 Failure analysis and analysis of safety system effectiveness **W, tg**
.17 Turbine Type Test Program [2)] **Z**
.18 Turbine Test Program [3)] **Z**
.19 Operation Manual including Manual for Emergency Situation Measures **Z, tg**
.20 Instruction for Preventive Maintenance **Z, tg**

References:

1) See 1.3.3.3.
2) Test program ought to have acceptance criteria defined. In case of turbines produced as single units separate test programs are not required.

Notes:

1. The documentation with code **Z** shall be approved by PRS.
2. The documentation with code **W** shall be submitted for reference but it may be subject of certain requirements by PRS.
3. The documentation with code **tg** is requested for gas turbines only (turbochargers excluded).
4. In case of turbines with a power below 100kW and for turbines dedicated for auxiliary purposes, the scope of the required documentation for approval may be lowered after agreeing it with PRS.

1.3.3.2 Documentation for heat exchangers see p.1.3.5.

1.3.3.3 Updated documentation of the Turbine Type and Torsional Vibration Calculations for the particular drive system is the base for PRS supervision.

1.3.3.4 If the turbine is being built under licence and engine manufacturer does not posses *Type Approval Certificate* for the Engine, he ought to provide the documentation in a scope given in 1.3.3.1 with detailed listing of all introduced changes with the reference to the approved type and design. PRS may request confirmation of presented changes by the licence holder – who is in possession of the *Type Approval Certificate*.

1.3.4 Documentation for Approval of Machinery

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

.1	Technical description and basic technical specification	**Z**
.2	General arrangement with cross section and dimensional data	**Z**
.3	Drawings of foundations, crankcases, columns and casings showing all details and welding procedures	**W/Z**
.4	Drawings of cylinder heads and cylinder liners	**W**
.5	Drawings of piston rods, connecting rods assemblies and pistons	**W**
.6	Drawings of rotors of blowers and compressors	**W**
.7	Drawings of crankshafts and other shafts transmitting torque moments	**Z**
.8	Drawings of pinions and toothed gear wheels (see also p. 4.2.1.2)	**Z**
.9	Drawings of disengaging and flexible couplings (see also p. 4.3.1.2)	**Z**
.10	Drawings of the main gear unit thrust bearing, unless built-in	**Z**
.11	Drawings of torsional vibrations dampers	**Z**
.12	Diagrams of control, alarm and safety systems within the machinery installation	**Z**
.13	Diagrams of the fuel oil, lubricating oil, cooling water and hydraulic system pipelines within the machine – including the information on flexible hoses applied	**Z**
.14	Thermal insulation drawings, including exhaust pipes	**W**
.15	Strength calculations of fastening components for fixing anchor winches to the deck including requirements of chapter 6.3.7 if applicable	
.16	Drawings of foundations of the auxiliary devices[2)], gears, steering gears, windlasses, mooring and towing winches	**Z**
.17	Material specification for essential parts with all details of non-destructive testing, pressure testing and special technologies used during manufacturing	**Z**
.18	Test program. [1)]	**Z**

References:

[1)] The Type Test Program and Product Test Program shall be provided where applicable.

[2)] In the scope agreed with PRS

Notes:

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

1.3.5 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 *Part VII* and arrangement of all welds with dimensions;

.2 Drawings of all boiler, pressure vessels and heat exchangers parts to be surveyed with the exception of charging air coolers whose dimensions are given in *Part VII*;

.3 Arrangement of valves and fittings including their specification;

.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 program;

.9 Boiler Operation and Service Manual.

1.3.6 Documentation for Approval of Thrusters

1.3.6.1 For approval of thrusters, the following documentation shall be submitted to PRS:

.1	Technical description and basic technical specification	**Z**
.2	Assembly drawing in cross section with dimensions	**Z**
.3	Drawings of casings, shafts and gears	**Z**
.4	Drawings of the nozzle and the propeller or other propulsion device	**Z**
.5	Drawings of pitch control device or vanes of cycloidal type propellers	**Z**
.6	Drawings of bearings, dynamic seals of the propeller shaft and the rotating column of propeller	**Z**
.7	Hydraulic, electrical, and pneumatic diagrams together with specification of the components	**Z**
.8	Diagrams of lubricating and cooling system if applicable	**Z**
.9	Diagram showing variation of the starting torque of the motor causing rotation of the propeller column	**W**
.10	Material specification for all essential parts as specified in .3, .4 and .5 with all details of non-destructive testing, pressure testing and special technologies used during manufacturing	**Z**
.11	Torsional vibrations calculations	**Z**
.12	Gears’ and roller bearings’ calculations	**W**

.13 Operation and Service Manual **W**
.14 Type testing program[1)] **Z**
.15 Test program[1)]. **Z**

References:

[1)] Test program shall include acceptance criteria. In the case of production of a single unit a separate test program for Type and Product is not required.

Notes:

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

1.3.6.2 Updated documentation is a base for PRS supervision during manufacturing of the thrusters.

1.3.6.3 If the device is being built under licence and device manufacturer does not posses *Type Approval Certificate* for the product type, he ought to provide the documentation in the scope specified in paragraph in 1.3.6.1 with a detailed listing of all introduced design changes with the reference to the approved type. PRS may request confirmation of presented changes by the licence holder – who is in possession of the *Type Approval Certificate*.

1.3.7 Documentation of Machinery Installations of Energy Efficient Ships with Additional Mark EF in the Symbol of Class

1.3.7.1 Documentation of respective machinery installations required at each stage of design as well as the documentation prepared after the sea trials for the final verification of attained EEDI shall be submitted to PRS for consideration and approval.

1.3.7.2 For ships intended to be assigned an additional mark EF in the symbol of class, the documents to be submitted are specified in the *Guidelines on Survey and Certification of EEDI* [IMO Res. MEPC.214(63)] and the *Industry Guidelines on Calculation and Verification of EEDI* both contained in *Publication 103/P – Guidelines for Energy Efficiency of Ships.*

1.4 Scope of Survey

1.4.1 General provisions concerning the survey of construction of engines, machinery, boilers, pressure vessels and heat exchangers covered in *Part VII* are contained in *Part I – Classification Regulations* and in *PRS Supervision Activity Regulations*.

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

Exempted from the survey during manufacturing are also compressed gas bottles produced in accordance with national standards and under the survey of competent technical inspection body recognised by PRS.

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

.1 Internal combustion engines (not in series production):
- crankshaft [M)];
- detachable crankshaft coupling flanges [M)];
- built-up crankshaft coupling bolts [M2)];
- steel piston crowns [M2)];
- piston rods [M2)];
- connecting rods with bearing covers [M)];
- crossheads [M2)];
- steel parts of cylinder liners [M1)];
- steel cylinder covers [M1)];
- main bearing supports of welded design, steel plates and transverse support – forged or cast steel transverse [M)];
- engine bed plate and crankshaft casing of welded design [M)];
- engine load bearing structure of welded design [M)];
- tie bolts [M)];
- turbocharger shaft and rotor including vanes (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, crossheads, main and connecting rod bearings;
- steel gear wheels for camshaft drive.

Supervision for compliance with the approved documentation for engines which are a subject of mass production is a subject of separate consideration by PRS

.2 Steam turbines:
- turbine casing [M)];
- manoeuvring gear casing and nozzle box [M)];
- shaft, rotor and rotor disk;
- blades [M)];
- shroud and lashing wire;
- nozzles, diaphragms [M)];
- gland-seals;
- rigid coupling;
- bolts connecting parts of the rotor, parts of turbine casing, couplings.

.3 Gears, disengaging and flexible couplings:
- casings;
- shafts [M)];
- pinions, tooth wheels, tooth rims [M)];

- torque transmitting parts of couplings:
  - rigid parts M) ;
  - flexible parts;
- connecting bolts.

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

.5 Centrifugal pumps, fans, air blowers and turbochargers:
- shafts;
- rotors;
- casings.

.6 Steering gears:
- tillers of main and emergency gear M);
- rudder quadrant M);
- rudderstock yoke M);
- pistons with piston rods M);
- ram cylinders M);
- drive shafts M);
- gear wheels tooth rims M).

.7 Windlasses, mooring and towing winches:
- drive, intermediate and output drive shafts M);
- gear wheels, tooth rims;
- sprockets;
- claw clutches;
- brake bands.

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

.9 Fuel and oil separators:
- shaft;
- bowl body, bowl disks;
- gear wheels.

.10 Boilers, steam superheaters and economisers as well as water heated steam generators:
- ring segments, end plates, tube plates, drums, headers and chambers M3);
- heated and non-heated tubes M3);
- furnaces and elements of combustion chambers M3);

- long and short stays and girders [M3)];
- bodies of mountings and fittings for working pressure 0.7 MPa and more [M3)].

.11 Pressure vessels and heat exchangers:
- shells, distributors, end plates, headers and covers [M3)];
- tube plates [M3)];
- tubes [M3)];
- long and short stays and girders, fastenings [M3)];
- bodies of valves for working pressure 0.7 MPa and more, 50 mm and over in diameter [M3)].

.12 Gas turbines:
- 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 linking components of the rotor, casing, clutches [M)];
- dynamic seals
- pipings and fittings [M)];

.13 Thrusters
- movable and stationary casings [M4)];
- columns [M4)];
- propeller shaft and intermediate shafts [M4)];
- propellers [M4)]
- nozzles;
- fastening elements, keys;
- pipings and fittings.

Notes and index explanations:

M) – material shall be PRS accepted.
M1) – material for parts of engines of cylinder bore 300 mm and more shall be PRS accepted.
M2) – material for parts of engines of cylinder bore 400 mm and more shall be PRS accepted.
M3) – material for parts of boilers, pressure vessels and heat exchangers of class I and II (see 8.1) shall be PRS accepted.
M4) – material approved by PRS. In case of auxiliary drive-steering devices with rated power below 200 kW, the certificate of the material manufacturer is accepted. The material shall be inspected by PRS surveyor and the hardness tests shall be carried out in his presence.

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

1.4.4 The survey of the mass production of internal combustion engines and turboblowers is carried out in accordance with the 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.*

1.4.5 Upon completion of assembly, adjustment and running in, each engine and piece of machinery shall be subjected to running tests at manufactures works, in accordance with the test program agreed with PRS.

The tests of internal combustion engines shall be performed taking into consideration the requirements specified in *PRS Publication No. 28/P – Tests of I.C. Engines.*

1.4.6 Type tests of engines and machinery shall be performed in accordance with the program which ensures checking the reliability and long operational suitability of individual parts, assemblies as well as of entire engines and machinery items. I.C. Engine Type testing shall be performed taking into consideration the requirements specified in PRS *Publication No. 28/P – Tests of I.C. Engines.*

Type testing may be waived when the number of engine cylinders has been changed, however, it is assumed, in accordance with paragraph 2.1.3, that the engine is of the same type.

1.4.7 If the rated power has been increased up to 10%, then PRS may consider dispensing with the type re-tests, provided that the engine at the increased power fulfils the requirements specified in sub-chapter 2.3.

1.5 Pressure Tests

1.5.1 Parts of Internal Combustion Engines

The components of internal combustion engines shall be subject to pressure tests in accordance with Table 1.5.1.

Table 1.5.1

<table>
<tr><th>Item</th><th colspan="2">Component name</th><th>Test pressure [MPa]</th></tr>
<tr><td>1</td><td colspan="2">2</td><td>3</td></tr>
<tr><td>1</td><td colspan="2">Cooling space of cylinder cover[1)]</td><td>0.7</td></tr>
<tr><td>2</td><td colspan="2">Cylinder liner over the whole length of the cooled space</td><td>0.7</td></tr>
<tr><td>3</td><td colspan="2">Cooling space of cylinder block</td><td>1.5 p not less than 0.4</td></tr>
<tr><td>4</td><td colspan="2">Exhaust valve cooling space</td><td>1.5p not less than 0.4</td></tr>
<tr><td>5</td><td colspan="2">Cooling space of the piston crown (where the cooling space is sealed by piston rod or by rod and skirt, test after assembling)[1)]</td><td>0.7</td></tr>
<tr><td rowspan="3">6</td><td rowspan="3">High pressure fuel injection system</td><td>Fuel injection pump body, pressure side</td><td>1.5 p or p+30 whichever less</td></tr>
<tr><td>Fuel injection valve</td><td>1.5 p or p+30 whichever less</td></tr>
<tr><td>Fuel injection pipes</td><td>1.5 p or p+30 whichever less</td></tr>
<tr><td>7</td><td colspan="2">Hydraulic system (high pressure hydraulic tubes, pumps, servomotors, etc.for exhaust valve control)</td><td>1.5 p</td></tr>
<tr><td>8</td><td colspan="2">Scavenging pump cylinder</td><td>0.4</td></tr>
</table>

1	2		3
9	Turbocharger, cooling space		1.5 *p* not less than 0.4
10	Exhaust pipe, cooling space		1.5 *p* not less than 0.4
11	Engine driven compressors (cylinders, covers, air coolers)	Air space	1.5 *p*
		Water space	1.5 p, not less than 0.4
12	Coolers, at both sides [2)]		1.5 p, not less than 0.4
13	Working spaces of engine driven pumps (lub. oil, water, fuel and bilge pumps)		1.5 p, not less than 0.4

Notes:

1) For cylinder covers and piston crowns 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 supercharging air coolers may be upon PRS consent tested at the water side only.

3) *p* – maximum working pressure for the given part.

1.5.2 Machinery Parts and Fittings

1.5.2.1 The parts of machinery working under pressure shall 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 shall 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 those values given in Table 1.5.2.1, then the test pressure shall 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
1	2	3	4	5	6	7	8	9	10	11	12
Carbon and carbon manganese steel	*p*, [MPa], up to	without limit	20	20	20	20	10	10	–	–	–
	K	0	0	1	3	5	8	11	11	–	–
Molybdenum and molybdenum-chromium steel with molybdenum contents 0.4% and more	*p*, [MPa], up to	without limit				20	20	20	20	20	20
	K	0	0	0	0	0	1	2	3.5	6	11

1	2	3	4	5	6	7	8	9	10	11	12
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 Pressure tests of machinery parts can be performed separately for each space, applying the test pressure according to working pressure and temperature in the particular 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 shall be tested for tightness applying procedure agreed with PRS. In welded structures, only welded joints shall 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, shall be pressure tested in accordance with Table 1.5.3.1.

Table 1.5.3.1

Item	Specification	Test pressure p_h, [MPa]	
		upon completion of construction or assembling of strength members of the shell elements, 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[1] and parts thereof, operating at temperature exceeding 350 °C and the following pressure: – up to 15 MPa	$1.5\, p_w \dfrac{R_e^{350}}{R_e^t}$	–

1	2	3	4
	– 15.0 MPa and more[2)]	$1.35\, p_w \frac{R_e^{350}}{R_e^t}$	–
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
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:

1) Pressure testing shall be done separately for each side of the heat exchanger. Testing of coolers of the I.C. engines – see Table 1.5.1.

2) For pressure p_w = 15 to 16.6 MPa, constant value of p_w = 16.6 MPa shall 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 ship, shall be steam tested under the working pressure.

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

1.6 Materials and Welding

1.6.1 Materials applied for construction of parts of internal combustion engines, pieces of machinery, boilers, pressure vessels and heat exchangers covered with these *Rules* shall fulfil the relevant requirements specified in *Part IX – Materials and Welding*.

1.6.2 Butt joints are generally to be used. Structures using fillet joints or joints affected by bending stress will be specially considered by PRS in each particular case.

The examples of welded joints used are given in the Annex to this Part of the *Rules*.

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

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

1.6.5 Materials used for the parts of steam 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.

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

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

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

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

1.6.8 The use of steel alloys for the construction of boilers, pressure vessels and heat exchangers is subject to PRS acceptance in each particular case. This requires the data concerning the mechanical properties and creep strength of the steel and welded joints at the design temperature, technological properties, welding procedure and heat treatment to be submitted for consideration.

1.6.9 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 blow-down valves, may be manufactured of ferritic nodular cast iron complying with the requirements specified in Chapter 15 of *Part IX – Materials and Welding*.

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

In other situations, the use of cast iron is subject to PRS acceptance in each particular case.

1.6.11 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 situations the use of copper alloys is subject to PRS acceptance in each particular case.

1.6.12 Seamless pipes shall be generally used for parts being the subject of this Part of the Rules. However, unless any special reservations have been expressed, upon agreement of PRS, longitudinally or spiral welded pipes may be used, provided their equivalence with seamless pipes has been proven.

1.6.13 Usage of materials that contain asbestos in installations, including spare parts, is prohibited for all ships according to SOLAS Regulation II-1/3-5, IACS UI SC 249 as well as MSC.1/Circ.1374 and MSC.1/Circ.1379.

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 the case of heat treatment applied to the welded parts, procedures shall fulfil 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 Class I (see Table 8.1) made of steel, of wall thickness exceeding 20 mm;
.3 boilers, heat exchangers and pressure vessels 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 shall 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 cast iron parts of built-up crankshafts;
.4 cast steel or cast iron parts of semi built-up crankshafts;
.5 connecting rods;
.6 piston rods;
.7 steel piston crowns;
.8 tie bolts;
.9 bolts subjected directly to variable loads (bolts of main bearings, big end 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 shall be subjected to ultrasonic tests confirmed by report signed by the manufacturer:

– parts of internal combustion engines of cylinder bore up to 400 mm specified in .1, .2, .3, .4, .7 and .10 under 1.8.1,
– parts of internal combustion engines of cylinder bore exceeding 400 mm specified in .1 through .7 and .10 under 1.8.1,
– rotor blades 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 PRS Surveyor shall be applied to:

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

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

1.8.4 Welded seams inspection with use of 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 performed 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 shall be performed in accordance with the requirements specified in *Part IX – Materials and Welding.*

1.9 General Technical Requirements

1.9.1 The design and make of machinery being the subject of *Part VII* shall ensure reliable operation thereof under environmental conditions specified in paragraph 1.6.1, *Part VI – Machinery Installations and Refrigerating Plants.*

1.9.2 Oil fuel used in internal combustion engines and boilers shall fulfil the requirements specified in paragraph 1.18.1, *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 the requirements specified in paragraph 1.9.8, *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 arrangements preventing their loosening.

1.9.5 The piping systems within engines, machinery and boilers shall fulfil the relevant requirements specified in *Part VI – Machinery Installations and Refrigerating Plants.*

1.9.6 The electrical equipment of engines, boilers and machinery shall fulfil the relevant requirements specified in *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 shall have corrosion-resistant 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 or measuring instruments and gauges is determined by the manufacturer and shall fulfil the requirements specified in sub-chapter 1.15, *Part VI – Machinery Installations and Refrigerating Plants*.

Monitoring and control instruments for engines installed in unattended machinery spaces shall fulfil the relevant requirements specified in *Chapter 21* of *Part VIII – Electrical Installations and Control Systems*.

1.9.9 Remote and automatic control systems, safety and alarm systems of engines and machinery shall fulfil the relevant requirements specified in *Part VIII – Electrical Installations and Control Systems*.

2 INTERNAL COMBUSTION ENGINES

2.1 General Requirements

2.1.1 The requirements of this Chapter apply to all internal combustion engines having rated power 55 kW and more.

Application of these requirements to diesel engines having rated power below 55 kW is subject to PRS acceptance 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 (direct or indirect);
- **.4** type of fuel (liquid, gaseous or mixed fuel);
- **.5** working cycle (four-stroke or two-stroke);
- **.6** gas exchange (naturally aspirated or supercharged);
- **.7** maximum rated power output per cylinder, rated rotational speed and maximum effective pressure;
- **.8** method of pressure charging (pulse system or constant pressure system);
- **.9** charging air cooling system (with or without intercoolers, number of inter-cooling stages);
- **.10** cylinder arrangement (in-line or V-type).

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[*)] shall be ensured at the following ambient conditions:

Table 2.1.4

Ambient conditions	For ships of unrestricted service	For ships of restricted service (outside the tropics)
Atmospheric pressure	100 kPa (750 mm Hg)	100 kPa (750 mm Hg)
Air temperature	+ 45 °C	+ 40 °C
Relative air humidity	60%	50%
Sea water temperature	+ 32 °C	+ 25 °C

2.1.5 The engines for main propulsion shall also fulfil the requirements of specified in sub-chapter 1.8, *Part VI – Machinery Installations and Refrigerating Plants*.

[*)] As the rated power is assumed the power as 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 operational test.

2.1.6 The scavenging spaces of crosshead engines having direct connection with cylinders, shall be provided with fire extinguishing system agreed with PRS which shall be independent of the machinery space fire extinguishing system.

The scavenging spaces of main engines in unattended machinery spaces are to be equipped with fire detecting installation transmitting the alarm signal in case of fire.

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

Fuel of flash point not less than 43 °C shall be used for these engines.

2.1.8 IC engines with more than 130 kW rated power output installed on ships constructed on 1 January 2000 or after that date, as well as engines subjected to substantial modification on 1 January 2000 or after that date shall fulfil the requirements specified in *Publication 69/P – Marine Diesel Engines. Control of Nitrogen Oxides Emission*.

2.1.9 Marine diesel engines fitted with Selective Catalytic Reduction (SCR) System shall fulfill requirements of regulation 13 of *MARPOL Annex VI.* Additional guidance for design, testing, surveys and certification of marine diesel engines fitted with SCR system is given in *Publication No. 98/P – Guidelines Regarding the Requirements for Marine Diesel Engines Fitted with* NO_X *Selective Catalytic Reduction (SCR) Systems.*

2.2 Engine Frame

2.2.1 The crankcase and its detachable or opened covers of openings shall be of suitable strength, the fastenings of covers shall be strong enough to prevent displacement of the covers in the case of explosion. The crankcase doors shall be fastened sufficiently securely so as not to be readily displaced by a crankcase explosion.

2.2.2 The engine frame 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 of cylinder blocks shall be fitted with drain arrangements providing for complete drying.

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

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

The turbo-blowers can be used for crankcase ventilation only for the engines with rated power not exceeding 750 kW, provided reliable oil separators are fitted.

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

2.2.4 Crankcases of engines having a cylinder bore of 200 mm and above or a volume of 0.6 m^3 and above shall be provided with safety devices (explosion relief valves) of a suitable type as follows:

.1 engines having a cylinder bore not exceeding 250 mm shall have at least one valve near each end of the crankcase; but engines having 8 cylinders or more shall have an additional valve fitted near the middle of crankcase;

.2 engines having a cylinder bore exceeding 250 mm, but not exceeding 300 mm, shall have at least one such valve in way of alternate crankthrow, with a minimum of two valves (not less than 2 devices for each engine);

.3 engines having a cylinder bore exceeding 300 shall have at least one valve in way of each main crankthrow.

2.2.5 The free area of each safety valve shall be not less than 45 cm^2. The combined free area of the valves fitted on an engine must not be less than 115 cm^2 per cubic metre of the crankcase gross volume. The volume of the fixed parts in the crankcase may be deduced in estimating the gross volume, however rotating and reciprocating components shall be included in the gross volume.

2.2.6 Crankcase safety devices (explosion relief valves) shall fulfil the following requirements:

.1 they shall be of the type approved by PRS and shall have *Type Approval Certificate* issued for a configuration that represents installation arrangements that will be used on an engine. Type approval procedure is given in *Publication 68/P – Type Testing Procedure for Crankcase Explosion Relief Valves*;

.2 the valves shall be designed and built for immediate opening of the valve at an overpressure of not more than 0.02 MPa and quick closing to prevent the inrush of air into the crankcase;

.3 crankcase safety valve discharges shall be properly shielded to provide protection for persons being near the engine against the possible danger from emission of flame;

.4 they shall be provided with lightweight spring-loaded valve discs or other quick-acting and self-closing devices to relieve a crankcase of pressure in the event of an internal explosion and to prevent the inrush of air thereafter;

.5 the valve discs in crankcase explosion relief valves shall be made of ductile material capable of withstanding the shock of contact with stoppers at the full open position;

.6 where crankcase explosion relief valves are provided with arrangements for shielding emissions from the valve following an explosion, the valve shall be type tested to demonstrate that the shielding does not adversely affect the operational effectiveness of the valve;

.7 documentation of explosion relief valves, submitted to PRS, shall include a copy of the manufacturer's installation and maintenance manual, pertinent to the size and type of the valve supplied for installation on a particular engine. The manual shall contain the following information:
- description of the valve, with details of function and design limits,
- copy of *Type Approval Certificate*,
- installation instructions,
- maintenance in service instructions, including testing and renewal of any sealing arrangements,
- actions required after a crankcase explosion.

Note:
A copy of explosion relief valves installation and maintenance manual shall be provided on board ship.

.8 they shall be provided with suitable markings that include the following information:
- name and address of the manufacturer,
- designation and size,
- month and year of manufacture,
- approved installation orientation.

2.2.7 On the both sides of the engine there shall be fitted plates or notices warning against opening the doors, covers or sight glasses for a period of time necessary for cooling down the engine parts after stopping the engine. It is accepted to place such warning on the engine control position.

2.2.8 Engines having a cylinder bore 230 mm or more shall be fitted with cylinder overpressure alarms indicating its permissible value.

2.2.9 Separate compartments of the crankcase such as gears or chain driving timing gear or similar drives, the volume of which exceeds 0.6 m^3 shall be equipped with additional explosion relief valves fulfilling the requirements specified in paragraphs 2.2.5 and 2.2.6.

2.2.10 Engine shall be provided with the following oil mist detection arrangements (or engine bearing temperature monitors or equivalent devices):
- for alarm and slow-down purposes, in the case of low speed diesel engines of 2250 kW rated power and above or having cylinders of more than 300 mm bore,
- for alarm and automatic shut-off purposes, in the case of medium and high speed diesel engines of 2250 kW rated power and above or having cylinders of more than 300 mm bore.

Oil mist detection arrangements shall be type-approved by PRS. Engine bearing temperature monitors or equivalent devices used as safety devices shall be of a type approved by PRS for such purposes.

It is recommended that the engine be fitted with a thrust bearing high temperature alarm if the thrust bearing is situated inside the engine and has a connection with the crankcase.

Note: An equivalent device is considered as measures applied to high-speed engines where specific design features to preclude the risk of crankcase explosions are incorporated.

2.2.11 To protect internal combustion engine against crankcase explosion, the following requirements shall be fulfilled:

.1 ventilation of crankcase and any arrangement which could produce a flow of external air within the crankcase is, in principle, not permitted, except for dual fuel (DFD) engines where crankcase ventilation shall be provided in accordance with the requirement of paragraph 2.12.3.2;

.2 where crankcase ventilation pipes are provided, they shall be as small as practicable to minimize the inrush of air after a crankcase explosion;

.3 if a forced extraction of the oil mist atmosphere from the crankcase is provided (e.g. for oil mist detection purposes), the vacuum in the crankcase shall not exceed 2.5×10^{-4} N/mm^2 (2.5 mbar);

.4 to avoid interconnection between crankcases and the possible spread of fire following an explosion, crankcase ventilation pipes and oil drain pipes for each engine shall be independent of any other engine;

.5 lubricating oil drain pipes from the engine sump to the drain tank must be submerged at their outlet ends;

.6 a warning notice shall be fitted either on the control stand or, preferably, on a crankcase door on each side of the engine. This warning notice shall specify that, whenever overheating is suspected within the crankcase, the crankcase doors or sight holes shall not be opened before a reasonable time, sufficient to permit adequate cooling after stopping the engine;

.7 oil mist detectors and other monitoring arrangements, fitted to the engine, shall have *Type Approval Certificate* issued by PRS or recognized classification society, shall be tested in accordance with the requirements specified in *Part VIII – Electrical Installations and Control Systems* (Table 21.3.1-1) and shall fulfil the requirements specified in sub-paragraphs .8 to .19;

.8 oil mist detection system and arrangements shall be installed in accordance with the engine designer's and oil mist detection system manufacturer's recommendations. The following particulars shall be included in oil mist detection system instructions:

– schematic layout of oil mist detection system and alarm system showing location of engine crankcase sample points and piping or cable arrangements, together with dimensions of pipes to detectors,
– evidence of study to justify the selected location of sample points and sample extraction rate (if applicable) with regard to the crankcase

arrangements and geometry, as well as the predicted crankcase atmosphere where oil mist can accumulate,
- the manufacturer' maintenance and test manual;
- information relating to type or in-service testing of the engine with engine protection system test arrangements having approved type of oil mist detection system;

.9 a copy of the oil mist detection equipment maintenance and test manual, required by .8, shall be provided on board ship;

.10 oil mist detection and alarm information shall be capable of being read from a safe location away from the engine;

.11 each engine shall be provided with its own independent oil mist arrangement and a dedicated alarm.

.12 oil mist detection system and alarm system shall be capable of being tested on the test bed and on board under engine at standstill and engine running at normal operating conditions in accordance with test procedures approved by PRS;

.13 alarms and shutdowns for the oil mist detection/monitoring system, as well as the system arrangements shall fulfil the requirements specified in Chapters 20 and 21, *Part VIII – Electrical Installations and Control Systems*;

.14 the oil mist detection arrangements shall provide alarm indication in the event of a foreseeable functional failure in the equipment and installation arrangements;

.15 the oil mist detection system shall indicate that any lenses fitted in the equipment and used in determination of the oil mist level have been partially obscured to a degree that will affect the reliability of the information and alarm indication;

.16 where oil mist detection system uses programmable electronic systems, the arrangements are subject to PRS acceptance in each particular case;

.17 plans showing details and arrangements of oil mist detection system and alarm arrangements are subject to PRS' approval in accordance with the requirements specified in paragraph 1.3.2, sub-paragraph .28;

.18 the equipment, together with detectors shall be tested when installed on the test bed and on board ship to demonstrate that the detection and alarm system functionally operates. Tests shall be performed in conditions as well as in accordance with the test programme, approved by PRS;

.19 where sequential oil mist detection arrangements are provided, the sampling frequency and time shall be as short as reasonably practicable;

.20 where alternative methods are provided for the prevention of the build-up of oil mist that may lead to a potentially explosive condition within the crankcase, detailed documentation shall be submitted to PRS for consideration and approval. The following information shall be included in the documentation:

- engine particulars – type, power, speed, stroke, bore and crankcase volume,
- details of arrangements preventing the build-up of potentially explosive conditions within the crankcase, e.g., bearing temperature monitoring, oil splash temperature, crankcase pressure monitoring, recirculation arrangements,
- documents to demonstrate that the arrangements are effective in preventing the build-up of potientially explosive conditions, together with details of in-service experience,
- operation and maintenance manual.

.21 where it is proposed to use the introduction of inert gas into the crankcase to minimize a potential crankcase explosion, documentation of the arrangements shall be submitted to PRS for approval.

Note:
The requirements, specified in paragraph 2.2.11, apply to internal combustion engines installed on ships classed with PRS when:
1) application for certification of an engine is dated on or after 1 January 2010; or
2) installed in new ships for which the date of contract for construction is on or after January 2010.

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

2.3.2 The constructions of crankshafts not covered by PRS *Publication No. 8/P* or crankshafts made of nodular cast iron with $500 \leq R_m \leq 700$ MPa are subject to PRS acceptance in each particular case, provided that complete strength calculation or the experimental data are submitted.

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

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

2.3.5 Reference marks shall be made on the outer side of the connection of the crank webs with the main journals of semi-built crankshafts.

2.3.6 Where the thrust bearing is built into the engine frame, the diameter of the thrust shaft shall not be less than that specified under 2.4 and 22.2.4, *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 the rated power.

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

2.4.3 Scavenging spaces of two-stroke engines with positive displacement type scavenging pumps, as well as the scavenging spaces with direct connection with the cylinders, shall be provided with safety valves set for the pressure exceeding that of scavenging air by not more than 50%.

The cross-sectional area of the safety valves shall not be less than 30 cm^2 per each cubic metre of the scavenging space capacity, including the volume of under-piston spaces in crosshead engines fitted with diaphragms unless these spaces are used for scavenging air compression.

2.5 Fuel System

2.5.1 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 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 for shielding purpose, these shall be of an approved type.

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

2.5.3 All surfaces whose 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 practicable) shielded or otherwise protected against fuel or fuel leak 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 requirements specified in sub-chapter 12.4 and paragraphs 1.10.1, 1.10.3, 12.8.2 and 13.2.4 of *Part VI – Machinery Installations and Refrigerating Plants* apply to the fuel and lubricating oil filters installed on the engines.

2.6 Lubrication

2.6.1 The main and auxiliary engines of output power more than 37 kW shall be equipped with alarm devices giving audible and visible signals in the case of lubricating signal failure.

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.

2.7 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 In addition to the non-return valves required in paragraph 16.3.2 of *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 electrically started engines 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 fulfil the requirements specified in sub-chapter 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 limiters of torque (fuel dose) preventing the engine load exceeding the rated torque, resulting from the power output defined in conditions specified in Table 2.1.4.

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

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

.2 the torque limiter shall be set to 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 shall be capable of withstanding a short duration overload with torque equal to 1.1 of the rated torque, at the rated engine speed. The engines of power generating sets shall be fitted with limiters of torque (fuel dose) preventing the engine against load exceeding 1.1 of the rated torque, 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 shall not exceed the values specified in 4 of Annex 2 to *Part VIII – Electrical Installations and Control Systems*.

2.10.4 The starting and reversing arrangements shall be so arranged as to preclude:

.1 engine operation in direction opposite to the desired one;

.2 reversing the engine when the fuel supply is on;

.3 starting the engine before reversal is completed;

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

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

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

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

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

In the case when a step load equivalent to the rated output of a generator is switched off, a transient speed variation in excess of 10 % of the rated speed may be acceptable, provided this does not cause the intervention of the overspeed device (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 be more than ±5% from the rated speed.

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

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

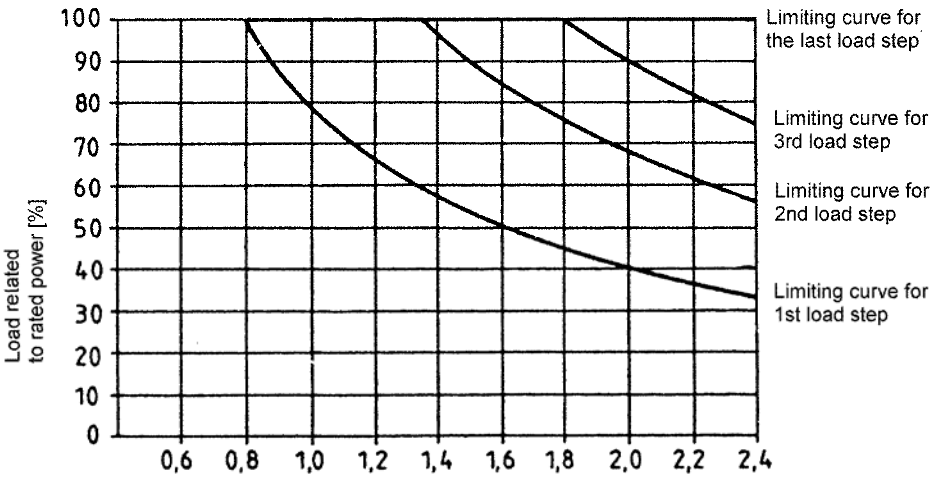


Fig. 2.10.6.4.
Limiting curves for loading 4-stroke engines step by step from no load to rated power as the function of brake mean effective pressure P_e, [MPa]

.5 Emergency generator sets shall fulfil the requirements specified in .1 and .2 even when:

a) their total consumer load is applied suddenly, or

b) their total consumer load is applied in steps, subject to:

– the total load is supplied within 45 seconds since power failure on the main switchboard,

– the maximum step load is declared and demonstrated,

– the power distribution system is designed such that the declared maximum step loading is not exceeded,
– time delay and loading sequence, specified above, are demonstrated at ship board trials.

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

2.10.7 Power generating sets intended for the parallel operation shall fulfil the requirements specified in sub-chapter 3.2.2, *Part VIII – Electrical Installations and Control Systems.*

2.10.8 Electronic governors of rotational speed shall also fulfil the relevant requirements specified in *Part VIII – Electrical Installations and Control Systems.*

2.10.9 The control devices of the main IC engine and control posts shall also fulfil the requirements specified in 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 oil 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 damper installed at the free end of crankshaft shall be such as to enable the fitting of torsional vibration measuring device.

2.12 Control and Protection Systems of Dual Fuel Diesel Engines (DFD)

2.12.1 Application

In addition to the requirements for oil firing diesel engines specified by the PRS Rules, and the requirements contained in chapters 5 and 16 of the *IGC Code*, as far as found applicable, the following requirements shall be applied to dual-fuel diesel engines utilizing high pressure methane gas (NG):

2.12.2 Operation Mode

2.12.2.1 DFD engines shall be of a dual-fuel type employing pilot fuel ignition and shall be capable of immediate change-over to oil fuel only.

2.12.2.2 Only oil fuel shall be used when starting the engine.

2.12.2.3 Only oil fuel shall, in principle, be used when the operation of an engine is unstable and/or during maneuvering and port operations.

2.12.2.4 In case of shut-off of the gas fuel supply, the engines shall be capable of continuous operation by oil fuel only.

2.12.3 Protection of Crankcase

2.12.3.1 Crankcase relief valves shall be fitted in way of each crankthrow. The construction and operating pressure of the relief valves shall be determined taking account of explosions due to gas leaks.

2.12.3.2 If a trunk piston type engine is used as DFD engine, the crankcase shall be protected by the following measures:
– ventilation shall be provided to prevent the accumulation of leaked gas, the outlet for which shall be led to a safe location in the open space through flame arrester;
– gas detecting or equivalent equipment. It is recommended that means for automatic injection of inert gas be provided;
– oil mist detector.

2.12.3.3 If a cross-head type engine is used as DFD, the crankcase shall be protected by oil mist detector or bearing temperature detector.

2.12.4 Protection for Piston Underside Space of Cross-Head Type Engine

2.12.4.1 Gas detecting or equivalent equipment shall be provided for piston underside space of cross-head type engine.

2.12.5 Engine Exhaust System

2.12.5.1 Explosion relief valves or other appropriate protection system against explosion shall be provided in the exhaust, scavenge and air inlet manifolds.

2.12.5.2 The exhaust gas pipes from DFD engines shall not be connected to the exhaust pipes of other engines or systems.

2.12.6 Starting Air Line

2.12.6.1 Starting air branch pipes to each cylinder shall be provided with effective flame arresters.

2.12.7 Combustion Control

2.12.7.1 A failure mode and effect analysis (FMEA) examining all possible faults affecting the combustion process shall be submitted.

Details of required monitoring will be determined based on the outcome of the analysis. However, the following table may serve as guidance.

Table 2.12.7.1

Faulty condition	Alarm	Aut. shut-off of the interlocked valves*
Function of gas fuel injection valves and pilot oil fuel injection valves	X	X
Exhaust gas temperature at each cylinder outlet and deviation from average	X	X
Cylinder pressure or ignition failure of each cylinder	X	X

* It is recommended that the gas master valve is also closed.

2.12.8 Gas Fuel Supply

2.12.8.1 Flame arresters shall be provided at the inlet to the gas supply manifold for the engine.

2.12.8.2 Arrangements shall be made so that the gas supply to the engine can be shut-off manually from starting platform or any other control position.

2.12.8.3 The arrangement and installation of the gas piping shall provide the necessary flexibility for the gas supply piping to accommodate the oscillating movements of DFD engine, without risk of fatigue failure.

2.12.8.4 The connecting of gas line and protection pipes or ducts regulated in 2.12.9.1 to the gas fuel injection valves shall provide complete coverage by the protection pipe or ducts.

2.12.9 Gas Fuel Supply Piping System

2.12.9.1 Gas fuel piping may pass through or extend into machinery spaces or gas-safe spaces other than accommodation spaces, service spaces and control stations, provided that they fulfil one of the following :

.1 The system complies with 16.3.1.1 of the *IGC Code*, and in addition, with a), b) and c) given below.

a) The pressure in the space between concentric pipes is monitored continuously. Automatic valves specified in 16.3.6 of the *IGC Code* (hereinafter referred to as “interlocked gas valves”) and the master gas fuel valves specified in 16.3.7 of the *IGC Code* (hereinafter referred to as “master gas valves”) shall be closed before the pressure drop to below the inner pipe pressure (however, an interlocked gas valve connected to vent outlet shall be opened) and an alarm shall activate.

b) Construction and strength of the outer pipes shall fulfill the requirements of 5.2 of the *IGC Code*.

c) It shall be so arranged that the inside of the gas fuel supply piping system between the master gas valve and the DFD engine shall be automatically purged with inert gas, when the master gas valve is closed; or

.2 The system complies with 16.3.1.2 of the *IGC Code*, and in addition, with a) through d) given below.

a) Materials, construction of protection pipes or ducts and mechanical ventilation systems shall have sufficient strength in case of bursting and rapid expansion of high pressure gas in the event of gas pipe burst.

b) The capacity of mechanical ventilating system shall be determined taking into account the flow rate of gas fuel and construction and arrangement of protective pipes or ducts, as deemed appropriate by the PRS.

c) The air intakes of mechanical ventilating systems shall be provided with non-return devices effective for gas fuel leaks. However, if a gas detector is fitted at the air intakes, these requirements may be dispensed with.

d) The number of flange joints of protective pipes or ducts shall be minimized; or

.3 Alternative arrangements to those given in paragraph .1 and .2 will be specially considered based upon an equivalent level of safety.

2.12.9.2 Sufficient constructive strength of high pressure gas piping system shall be ensured by carrying out stress analysis taking into account the stresses due to the weight of the piping system including acceleration load when significant, internal pressure and loads induced by hog and sag of the ships.

2.12.9.3 All valves and expansion joints used in high pressure gas fuel supply lines shall be of approved type.

2.12.9.4 Joints on entire length of the gas fuel supply lines shall be butt-welded joints with full penetration and to be 100% radiographed, except situations where are separately considered by the PRS.

2.12.9.5 Pipe joints other than welded joints are subject each time to separate PRS approval.

2.12.9.6 For all butt-welded joints of high pressure gas fuel supply lines, post-weld heat treatment shall be performed depending on the kind of material.

2.12.10 Gas Fuel Supply Shut-off

2.12.10.1 In addition to the cases specified in 16.3.6 of the *IGC Code*, supply of gas fuel to DFD engines shall be shut off by the interlocked gas valves in the following cases;

– conditions specified in 2.12.7.1 occur,
– DFD engine stops for any reason,
– conditions specified in 2.12.9.1.1 (a) occur.

2.12.10.2 In addition to the cases specified in 16.3.7 of *IGC Code*, the master gas valve shall be closed if any of the following conditions occur:

– oil mist detector or bearing temperature detector specified in 2.12.3.2 and 2.12.3.3 detects failure,
– any kind of gas fuel leakage is detected,
– condition specified in 2.12.9.1.1a) occurs,
– conditions specified in 2.12.11.1 occur.

2.12.11 Emergency Stop of DFD Engines

2.12.11.1 DFD engine shall stop before the gas concentration detected by the gas detectors specified in 16.2.2 of the *IGC Code* reaches 60% of lower flammable limit.

2.12.12 Gas Fuel Make-up Plant and Related Storage Tanks

2.12.12.1 Construction, control and safety system of high pressure gas compressors, pressure vessels and heat exchangers constituting a gas fuel make-up plant shall satisfy the PRS requirements.

2.12.12.2 The possibility for fatigue failure of the high pressure gas piping due to vibration shall be taken into account.

2.12.12.3 The possibility for pulsation of gas fuel supply pressure caused by the high-pressure gas compressor shall be taken into account.

3 TURBINES

3.1 Application

The requirements specified in this Chapter apply to the turbines for main propulsion and to those driving electric generators and auxiliaries.

3.2 Steam Turbines

3.2.1 General Requirements

3.2.1.1 The main geared turbine set shall be so designed as to enable the reversing from full speed ahead at the rated power to astern speed, and the reversing in the opposite direction using the backsteam.

3.2.1.2 Turbines intended to be installed onboard ships as the main turbines shall also fulfil the requirements specified in sub-chapter 1.8 of *Part VI – Machinery Installations and Refrigerating Plants*. In multi-screw ships with fixed pitch propellers an astern turbine shall be provided to each shaft.

3.2.1.3 Turbines driving auxiliary machinery shall be of such design as to be capable of being started without preheating.

3.2.1.4 In single screw vessels with multi-case turbines, provision shall be made for safe operation when the steam inflow to any casing is shut down. For this purpose, the steam may be supplied directly to the low-pressure turbine and from the high or medium-pressure turbine the steam may be led directly to the condenser.

Suitable means and control systems shall be provided for operation in such conditions such as the steam pressure and temperature not to exceed the values that are safe for the turbine and condenser.

The pipes and valves for these means must be readily available and properly marked. A fit up test of all combinations of pipes and valves shall be performed prior to the first sea trials.

The permissible power/speeds when operating without one of the turbines (all combinations of pipes and valves) shall be specified and information provided on board.

The operation of the turbines under emergency conditions shall be assessed for the potential influence on shaft alignment and gear teeth loading conditions.

3.2.2 Rotor

3.2.2.1 The strength of rotor parts shall be calculated for the maximum power, as well as for other possible loads at which the stress may rise to the maximum values.

Moreover, a check calculation of the stress shall be made for the rotor and parts thereof running at the speed exceeding the maximum values by 20%.

3.2.2.2 The rotor critical speed shall be in excess of the rated speed corresponding to the rated power by not less than 20%.

The difference between rotor critical speed and rated speed can be reduced, provided the reliability of the turbine has been proved for all operating conditions of load.

3.2.2.3 Each new design of blading requires a calculation of vibration with subsequent experimental verification of the vibration characteristics.

3.2.2.4 The constructions of blade tenon with detachable part of the disk side and other similar constructions, which may cause considerable local weakening of the rim, are not allowed.

3.2.2.5 Completely assembled turbine rotors shall be dynamically balanced in a machine of sensitivity adequate to the size and weight of the rotor.

3.2.3 Casing

3.2.3.1 In cast steel turbine casings it is permitted for some cast elements and branches for connecting receivers, tubes and fittings to be joined by welding.

3.2.3.2 The connection of astern turbine inlet branch with the outer turbine casing shall ensure free thermal deformation of these parts.

3.2.3.3 Gaskets shall not be used between the flanges of horizontal and vertical joints of turbines. The joint planes are allowed to be coated with graphite paste for the purpose of packing.

3.2.3.4 The diaphragms fixed in the turbine casing shall have a possibility of radial thermal expansion within permissible misalignment.

3.2.3.5 The diaphragms shall be designed for a load corresponding to the maximum pressure drop in the stage. The actual deflection of the diaphragms shall be less than that which may cause contact of diafragm with a disk or seizure of the diaphragm packing.

3.2.3.6 The low-pressure turbine casing shall be provided with openings for the inspection of blades, their fastenings and shroud in the last stages. The turbines integrated with condensers shall be provided with openings for the inspection of the upper rows of condenser tubes and, where possible, for providing access inside the condenser.

3.2.3.7 The turbine shall be so designed as to allow lifting of bearing caps without dismantling the turbine casing, ends of sealing arrangements and pipelines.

3.2.4 Bearings

3.2.4.1 Slide bearings shall be used in the main turbines. For turbines designed for quick start up when in cold condition, it is recommended to use bearings with self-aligning shells.

3.2.4.2 Thrust bearings of the main turbines are, as a rule, to be of a single-collar type. The use of bearings of other types is subject to PRS acceptance in each particular case.

3.2.4.3 The bearings loaded with specific pressure of more than 2 MPa are recommended to be fitted with devices for automatic equalisation of pressure exerted in the pad.

3.2.4.4 The thickness of antifriction lining of thrust bearing pads shall be less than the minimum axial clearance in the turbine blading, however not less than 1 mm.

3.2.5 Steam Suction, Gland-sealing and Blowing Systems

3.2.5.1 The main turbine sets shall be provided with a steam suction and gland-sealing system with automatic control of sealing steam pressure.

In addition to the automatic control, provision shall be made for manual control of the steam suction and gland-sealing system.

3.2.5.2 Each turbine shall have a blowing system to ensure complete removal of condensate from all stages and spaces of the turbine.

The blowing system shall be so arranged as to prevent the condensate from entering the idle turbines.

3.2.6 Control, Safety Devices and Governors

3.2.6.1 Controls of manoeuvring valves for turbine set 7500 kW and more shall be power-driven; emergency manual control of the valves shall also be provided.

3.2.6.2 The time required for resetting the controls of the turbine set manoeuvring gear from full ahead to full astern or vice versa shall not be more than 15 seconds.

The construction of manoeuvring gear shall be such as to make impossible the simultaneous admission of steam to the ahead turbine and to astern turbine.

3.2.6.3 Main and auxiliary turbines shall be provided with an emergency protective device (quick-closing stop valve) to shut off automatically the admission of steam to the turbine when the rotational speed is exceeded by 15%.

PRS may approve a single emergency overspeed device (quick closing-stop valve) for two or more turbines connected with the same gear.

The quick-closing stop valve shall be actuated by an overspeed protective device connected directly with the turbine shaft.

A hydraulic switch actuated by impeller driven directly by the turbine shaft may be used as the overspeed protective device.

The speed governors of turbine sets intended for driving electric power generators shall fulfil the requirements specified in paragraph 2.10.6.

3.2.6.4 Each turbine shall be fitted with a device to shut off the steam in emergency. This device, closing immediately the quick-acting stop valve, shall be activated manually.

In main turbo-electric propulsion set, this device shall be operated from two positions, one located at one of the turbines and the other in the control station.

In auxiliary turbo-electric sets, this device shall be fitted close to the overspeed protective device.

As such device (so called emergency stop) is considered each manually activated device, irrespective of the means of the activating pulse transmitting, for instance mechanically or with use of external power.

3.2.6.5 Where turbine installation comprises a reverse gear, c.p. propeller or other free-coupling arrangement or is a turbo-electric set, a separate speed governor shall be fitted to be capable of controlling the speed of the unloaded turbine without bringing the overspeed protective device into action.

3.2.6.6 Where exhaust steam from auxiliary systems is led to the main turbine, it shall be cut off at activation of the overspeed protective device.

3.2.6.7 The auxiliary turbines for driving electric generators shall be fitted with:
- speed governor keeping the momentary variations of rotational speed within 10% and the steady state rotational speed under newly established conditions within 5% in case of abrupt drop of full load, and
- safety governor, in addition to the speed governor, preventing the rotational speed to be exceeded by more than 15% (see also 3.2.6.3).

3.2.6.8 The main turbines for ahead drive shall be fitted with quick acting device shutting off the admission of steam in case of dangerous drop of pressure in the bearing lubrication system. This device shall not close the steam flow to the astern turbine.

Where deemed necessary, PRS may require apprioprate means to be provided to protect the turbines in case of:
- abnormal axial rotor displacement,
- excessive condenser pressure,
- too high condensate level.

3.2.6.9 Auxiliary turbines with speed governors other than hydraulic ones using the oil from the turbine lubricating system, shall be fitted with alarm devices and means of cutting-off the steam flow in case of oil pressure drop in the lubricating system of bearings.

3.2.6.10 Main turbines shall be provided with effective emergency supply of lubricating oil actuated automatically in case of the pressure drop below the preset value.

The emergency supply of lubricating oil may be secured from a gravitational tank of sufficient capacity for proper lubrication until the turbine stops or with use of other, equivalent means. If emergency pumps are used for this purpose, their operation can not be affected by the power decay.

A device cooling the bearings after stopping the turbine may be required.

3.2.6.11 To provide a warning to personnel of excessive pressure, sentinel valves or equivalent shall be provided at the exhaust end of all turbines. The valve discharge outlets shall be visible and suitably guarded if necessary.

Where the pressure of inlet steam of auxiliary turbines exceeds the design pressure of the turbine stage to which the steam from the auxiliary turbine outlet is admitted, the casing of this stage and the inlet pipeline shall be provided with devices relieving the pressure if higher than the design value.

3.2.6.12 The steam pipelines shall be fitted with non-return valves or other approved devices preventing the return of steam or condensate to the turbine.

3.2.6.13 Efficient steam strainers shall be provided close to the inlets to ahead and astern high pressure turbines or, alternatively, at the inlets to manoeuvring valves.

3.3 Gas Turbines

3.3.1 Definitions

3.3.1.1 G a s t u r b i n e means in Chapter 3.3, the engine consisting of:
- compressor,
- combustion chamber(s),
- gas generator turbine and heat exchanger (if applicable),
- power turbine,

together with foundation, control devices and all integrated auxiliary systems.

3.3.1.2 In Chapter 3.3, a t u r b i n e means a gas generator turbine as well as power turbine.

3.3.2 Reference Conditions

The model reference atmosphere shall be taken in accordance with standard ISO 2314:
- temperature 15 °C
- relative humidity 60%
- atmospheric pressure 101.3 kPa (760 mm Hg).

3.3.3 Arrangement

3.3.3.1 The air-inlet system shall be so located and designed as to exclude, as far as possible, the entrance of harmful foreign matter, including seawater and exhaust gas. Where considered necessary, the air intake system shall incorporate filtration system and de-icig arrangements. No screwed joints shall be used in the air intake ducting structure. Riveted joints are not recommended.

3.3.3.2 The exhaust outlets shall be so located and arranged as to preclude, as far as possible, reverse suction of combustion gases to the compressor.

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

3.3.3.4 Pipe or duct connections shall be made in such a way as to prevent the transmission of excessive loads or torques to the turbine casing. Pipes and ducts connected to casings as well as platform gratings shall be so arranged that thermal expansion is not restricted.

3.3.3.5 Hand trip gear for shutting off the fuel in an emergency shall be provided locally at the turbine control platform and, where applicable, at other control stations.

3.3.3.6 If the temperature on the gas turbine outer surface exceeds 220 °C and the casing cannot be insulated in a way that excludes leakage of flammable fluid onto that surface, then the gas turbine shall be fitted within an enclosure. The enclosure shall be provided with an appropriate mechanical ventilation, a fire detection system and automatic fire extinguishing system.

3.3.4 Design

3.3.4.1 Insulation of gas turbine shall fulfil the requirements specified in paragraph 1.9.8 of *Part VI – Machinery Installations and Refrigerating Plants.*

3.3.4.2 The design of the gas turbine shall assure that, after possible failure and separation of any rotor blade, no damage is done to the structure outside turbine and compressor casings. Particularly, the possibility of subsequent fire, fuel or any other combustible fluid leak and injury to the personnel shall be avoided.

3.3.4.3 The service life between major overhauls, as set by the manufacturer and confirmed with tests, in general, shall not be less than 5000 hours, for typical operational conditions of the ship.

3.3.5 Starting Arrangements

3.3.5.1 Starting program, if applicable, shall ensure that the starting is aborted if during the starting sequence the appropriate check parameters, such as rotating speed, air pressure after compressor etc., are not met.

3.3.5.2 With use of automatic or interlocked means, clearing all parts of the main gas turbine of the accumulation of liquid fuel or purging gaseous fuel shall be ensured before ignition commences on.

Prior to each ignition, the purge phase shall be of sufficient duration to displace the gas turbine volumes minimum 3 times.

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

3.3.6 Controls and Governors

3.3.6.1 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%.

3.3.6.2 Propulsion turbines 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.

3.3.6.3 Each turbine driving the main or emergency electric generator shall fulfil the requirements specified in paragraphs 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 full load of the emergency generator, the turbine shall be able to accept that sum of the emergency loads.

3.3.6.4 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.

3.3.7 Monitoring Systems on Ships with Continuous Machinery Watch

3.3.7.1 Monitoring systems shall fulfil the relevant requirements for automatic and remote control systems in *Part VIII – Electrical Installations and Control Systems*, Chapters 20.1, 20.2, 20.3, 20.4. Additionally, the monitoring systems shall fulfil the relevant requirements of *Part VIII*, Chapters 20.5 and 20.6, the scope of the compliance is subject to PRS acceptance in each particular case.

3.3.7.2 The gas turbines driving main propulsion shall be fitted with alarm systems and safety system in accordance with Tab. 3.3.7.2-1. Other gas turbines shall be fitted with alarm and safety systems in accordance with Tab. 3.3.7.2-2. For gas turbines having a rated power of less then 100 kW, those requirements may be lowered after an agreement with PRS.

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

3.3.7.4 In addition to the alarms specified in Tab. 3.3.7.2-1, it is recommended that a low level alarm for lubricating oil system tank be provided.

Table 3.3.7.2-1
Monitoring systems for main propulsion turbines

No.	Monitored parameter	Alarm system: monitored value of parameter	Safety system*	Comments
1	2	3	4	5
1	Speed **	Maximum	shutdown	applies to every 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 oil supply pressure	Maximum		–
7	Bearing temperature	Maximum	–	–
8	Fuel oil temperature	Maximum	–	–
9	Cooling fluid temperature	Maximum	–	
10	Flame and ignition	Flame decay and ignition failure	shutdown ***	see also paragraph 3.3.5.2
11	Starting procedure	Starting failure	shutdown	see also paragraph 3.3.5.2
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 fluid pressure of the speed governor and the safety system servomotors
17	Safety system	automatic shutdown	–	applies also to the hand trip

* The shutdown by the safety system means: acting in accordance with the requirements specified in paragraph 3.3.7.3. After the shutdown the turbine may be motored.

** It is recommended that the alarm level be set to 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 3.3.7.2-2
Monitoring systems for auxiliary turbines

No.	Monitored parameter	Alarm system: monitored value of parameter	Safety system	Comments
1	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 that the alarm level be set at 5÷8% above rated speed. The shutdown level shall be set at 15% above the rated speed.

3.3.8 Survey, Testing and Certificates

3.3.8.1 Gas turbines for PRS classed ships shall be PRS type approved.

3.3.8.2 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 specialized national agency.

3.3.8.3 Each gas turbine as described in paragraphs 3.3.8.1 and 3.3.8.2 shall be submitted to PRS supervision during manufacturing and tests, in accordance with paragraphs 3.3.8.4 to 3.3.8.18.

3.3.8.4 PRS survey of the manufacture and shop tests, described in paragraphs 3.3.8.4 to 3.3.8.14, comprises:

.1 Checking the applied materials and technologies for conformity with approved technical documentation,

.2 Checking the configuration for conformity with approved technical documentation,

.3 Certification testing according to the approved certification tests program, including:
- pressure test of casings, piping and fittings,
- testing the components,
- turbine shop trial.

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

3.3.8.5 The materials manufactured under the survey in accordance with paragraph 1.4.3.12, as well as welding, heat treatment and other procedures accepted during the documentation approval shall be checked.

3.3.8.6 Any 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. The certification testing may be started no sooner than those deviations are approved.

3.3.8.7 All casings shall be tested to a hydraulic pressure as required in paragraph 1.5.2.1, where the value of *p*, used in calculations, is the highest pressure in the casing during normal operation, or the pressure during starting, whichever is the 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 in accordance with Table 1.5.3.1.

3.3.8.8 In 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 five per cent above the nominal setting of the overspeed protective device, or 15 per cent above the maximum design speed, whichever is the higher.

3.3.8.9 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, partially or fully, the shop trial aboard the ship.

Shop trial scope:

.1 Starts and stopping tests;

.2 Checking the turbine smooth running at no load;

.3 The test to demonstrate the gas turbine's 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 gas turbine is intended for main propulsion;

.6 the performance test of the gas turbine, carried out according to the international or national standards accepted by PRS. The test shall be performed out under ambient conditions being as close as possible to the standard reference conditions as set in sub-chapter 3.3.2. 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.

3.3.8.10 For turbines driving electric generators it is recommended that the capability of delivering 110% of the rated power for a period of 5 minutes, as well as fulfilment of the requirmeents specified in paragraph 3.3.6.3 be checked.

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

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

3.3.8.13 Certification tests are positively accepted when the test results are complying with the design data and for every test the acceptance criteria from the PRS approved test schedule are fulfilled.

3.3.8.14 The 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 sea trials.

3.3.8.15 The sea trials for gas turbine shall be performed in accordance with an approved test program. The compliance of the turbine and its installation with the approved documents shall be demonstrated, as well as the capability to provide the main propulsion or other power delivery in all real variants of running at sea and of manoeuvres. In course of testing the main propulsion and essential electric generators driving turbines shall be performed, 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 levels.

3.3.8.16 During the test of monitoring systems, the compliance with the requirements specified in paragraph 3.3.7.1 shall be demonstrated.

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

3.3.8.18 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 require also submitting sea trial test reports for gas turbines intended for other purposes.

4 GEARS, DISENGAGING AND FLEXIBLE COUPLINGS

4.1 General Requirements

4.1.1 The construction of a gear shall ensure normal operation in the conditions specified in paragraph 1.6.1, *Part VI – Machinery Installations and Refrigerating Plants*. Flexible and disengaging couplings fitted in propulsion lines at ships with ice reinforcements shall fulfil the requirements specified in paragraph 22.2.10 of *Part VI – Machinery Installations and Refrigerating Plants.*

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

.1 Static balancing shall be applied to parts rotating with the following 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 a tangential velocity:
$v \geq 50$ m/s.

4.2 Reduction Gears

4.2.1 General Requirements

4.2.1.1 The requirements of this section apply to the propulsion gears and auxiliary gears with cylindrical wheels of external and internal mesh having spur or helical teeth of involute profile.

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

4.2.1.2 The technical documentation of reduction gears (see sub-chapter 1.3.4) shall contain all data necessary for design calculation, carried out in accordance with the requirements specified in sub-chapter 4.2.3. The calculation applies to gear wheels and shafts transmitting the power from the engine output to gear output.

4.2.2 Input Data for Stress Calculation in Gear Wheel Teeth

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

4.2.2.2 In order to make the requirement provisions more simple, the following nomenclature has been assumed:

pinion – this gear wheel of the pair that has less number of teeth (all the symbols concerning this wheel are marked with subscript character 1), wheel – the gear wheel of the pair with the greater number of teeth (all the symbols concerning this wheel are marked with subscript character 2).

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

a – centre distance, [mm];
b – face width, [mm];
b_1 – toothed rim width – pinion,, [mm];
b_2 – toothed rim width – wheel, [mm];
d – pitch cylinder diameter (reference diameter), [mm];
d_1 – pitch cylinder diameter – pinion, [mm];
d_2 – pitch cylinder diameter – wheel, [mm];
d_{a1} – tip circle diameter – pinion, [mm];
d_{a2} – tip circle diameter – wheel, [mm];
d_{b1} – base circle diameter – pinion, [mm];
d_{b2} – base circle diameter – wheel, [mm];
d_{f1} – root circle diameter – pinion, [mm];
d_{f2} – root circle diameter – wheel, [mm];
d_{w1} – working circle diameter – pinion, [mm];
d_{w2} – working circle diameter – wheel, [mm];
F_t – rated tangential force at working cylinder, [N];
F_b – rated tangential force at transverse section of base cylinder, [N];
h – tooth depth, [mm];
m_n – normal module, [mm];
m_t – transverse module, [mm];
n_1 – rotational speed – pinion, [rpm];
n_2 – rotational speed – wheel, [rpm];
P – maximum power transmitted by the gear (in the case of main gears intended for ships with ice class, the requirements specified in sub-chapter 22.2.10, *Part VI – Machinery Installations and Refrigerating Plants*, shall be taken into account), [kW];
T_1 – torque transmitted by pinion, [Nm];
T_2 – torque transmitted by wheel, [Nm];
u – gear ratio;
v – tangential velocity at generating cylinder, [m/s];
x_1 – correction coefficient of basic rack tooth profile – pinion;
x_2 – correction coefficient of basic rack tooth profile – wheel;
z_1 – number of teeth – pinion;
z_2 – number of teeth – wheel;
z_n – virtual number of teeth;
α_n – profile angle at normal section of pitch cylinder, [°];
α_t – profile angle at transverse section of pitch cylinder, [°];
α_{tw} – profile angle at transverse section of working cylinder, [°];
β – base helix angle at pitch cylinder, [°];

β_b – base helix angle at base cylinder, [°];
ε_α – transverse contact ratio [–];
ε_β – pitch contact ratio, [–];
ε_γ – total contact ratio, [–];
$inv\ \alpha$ – tooth profile involute angle associated with considered profile angle α, [rad];
α – profile angle (for definition of involute angle), [°].

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

4.2.2.3 Selected Formulae for Gearing

Gearing ratio is defined as follows:

$$u = \frac{z_2}{z_1} = \frac{d_{w2}}{d_{w1}} = \frac{d_2}{d_1} \tag{4.2.2.3}$$

where u takes the following signs:
– plus – for external mesh,
– minus – for internal mesh.

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

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

$$d = \frac{zm_n}{\cos\beta},$$

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

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

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

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

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

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

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

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

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

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

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

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

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

4.2.2.4 Rated Tangential Force F_t

Rated tangential force F_t, tangent to working cylinder and positioned in the plane perpendicular to the rotation axis is calculated from the maximum continuous power transmitted by the gear, taking into account the requirements specified in sub-chapter 22.2.10 of *Part VI – Machinery Installations and Refrigerating Plants*, with the use of the following formulae:

$$T_1 = 9549 \cdot \frac{P}{n_1}\,, \qquad T_2 = 9549 \cdot \frac{P}{n_2} \tag{4.2.2.4-1}$$

$$F_t = 2000 \cdot \frac{T_1}{d_1} = 2000 \cdot \frac{T_2}{d_2}\,, \quad \text{[N]}. \tag{4.2.2.4-2}$$

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

This sub-chapter defines the coefficients applied in the formulae checking gear wheel teeth strength for the contact stress (in accordance with sub-chapter 4.2.4) and for bending stress (in accordance with sub-chapter 4.2.5). Other coefficients specific for the strength formulae are presented in sub-chapters 4.2.4 and 4.2.5.

All the coefficients shall be calculated from the respective formulae or following particular instructions.

4.2.3.1 Application Factor K_A

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

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

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

The requirements specified in sub-chapter 22.2.10 of *Part VI – Machinery Installations and Refrigerating Plants* shall also be taken into account, if applicable.

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

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

K_A factor shall be determined by measurements or using an analytical method approved by PRS. Where the factor is impossible to be determined that way, its value may be taken in accordance with Table 4.2.3.1.

Table 4.2.3.1
Values of K_A for different applications

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

4.2.3.2 Load Sharing Factor K_γ

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

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

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

– for planetary gears:

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

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 Dynamic Factor K_v

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

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

This factor depends mainly on:
– mesh errors (depending on pitch and profile errors),
– pinion's and wheel's weights,
– changes in mesh rigidity during the wheel loading cycle,
– tangential velocity at working cylinder,
– dynamical unbalance of wheels and shaft,
– rigidity of shaft and bearings,
– gear damping characteristics.

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

Factor K_v may be calculated as follows:
.1 for spur gears:
K_v – in accordance with Fig. 4.2.3.3-2;

.2 for helical gears:

– if $\varepsilon_\beta > 1$

K_v – in accordance with Fig. 4.2.3.3-1,

– if $\varepsilon_\beta < 1$

K_v is obtained by linear interpolation using the following formula:

$$K_v = K_{v2} - \varepsilon_\beta \cdot (K_{v2} - K_{v1}),$$

where:

K_{v1} – value of K_v for helical gears, see Fig. 4.2.3.3-1,

K_{v2} – value of K_v for spur gears, see Fig. 4.2.3.3-2;

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

$$K_v = 1 + K_1 \cdot \frac{v \cdot z_1}{100}, \tag{4.2.3.3.3}$$

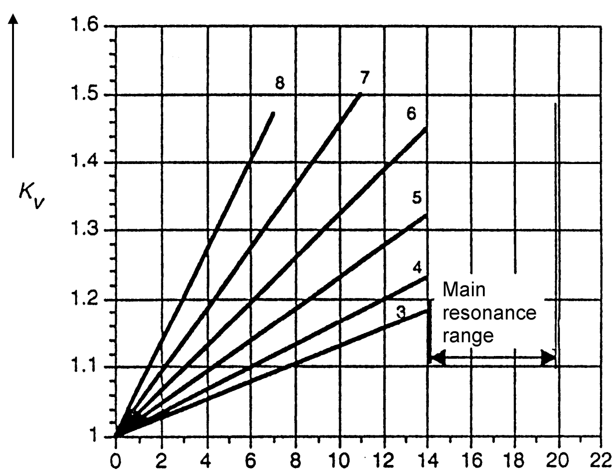
where:

K_1 – in accordance with Table 4.2.3.3.

Table 4.2.3.3
Values of K_1 for calculation of K_v

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

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



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

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

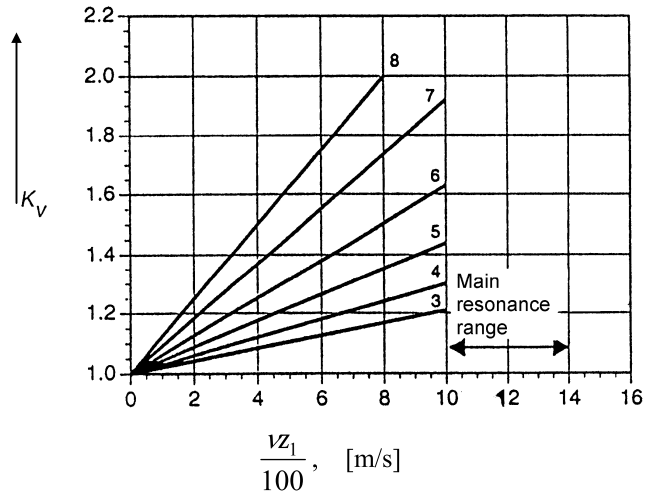


Fig. 4.2.3.3-2. Dynamic factor for spur gears. Accuracy classes 3 – 8 acc. to ISO 1328

For other gears than specified above, factor K_v shall be calculated in accordance with the requirements of standard ISO 6336 – 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 tooth root bending stress, take into account the effects of uneven load distribution throughout the tooth face width.

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

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

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

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

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

Factors $K_{H\beta}$ and $K_{F\beta}$ depend mainly on:
– teeth machining accuracy;
– assembly errors due to hole boring errors;
– bearings' clearances;
– misalignment of pinion and wheel axes;
– deformations due to insufficient rigidity of gear parts, shafts, bearings, casing and foundation;
– thermal elongations and other deformations at working temperature;

– compensating construction of parts (barrel shape, tooth tips' relief etc.).

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

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

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

where:

$$N = \frac{\left(\frac{b}{h}\right)^2}{1 + \frac{b}{h} + \left(\frac{b}{h}\right)^2} \qquad \frac{b}{h} = \min\left(\frac{b_1}{h_1}, \frac{b_2}{h_2}\right)$$

Note:

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

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

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

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

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

Transverse load distribution factors such as:

$K_{H\alpha}$ – for contact stress,

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

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

Factors $K_{H\alpha}$ and $K_{F\alpha}$ depend mainly on:

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

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

4.2.3.6 Factor selection methods other than those specified in sub-chapter 4.2.3 may be used subject to PRS approval in each particular case.

4.2.4 Contact Stress in Gear Wheel Teeth

4.2.4.1 The strength criterion for the contact stress is specified using Hertzian formulae for calculation of interface pressure at the active mesh point (or at the internal mesh point) of a single pair of teeth. The contact stress σ_H shall not exceed the permissible contact stress σ_{HP}.

4.2.4.2 The basic formula of contact stress σ_H is as follows:

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

where:

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

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

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

where:

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

4.2.4.3 Calculation of Allowable Contact Stress σ_{HP}

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

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

where:

σ_{Hlim} – fatigue strength of tooth material for contact stress, [N/mm²] (see paragraph 4.2.4.9);
S_H – safety factor for contact stresses (see paragraph 4.2.4.14);
Z_N – life factor for contact stress (see paragraph 4.2.4.10);

Z_L – factor of lubrication (see paragraph 4.2.4.11);
Z_v – velocity factor (see paragraph 4.2.4.11);
Z_R – roughness factor (see paragraph 4.2.4.11);
Z_W – hardness ratio factor (see paragraph 4.2.4.12);
Z_X – size factor (see paragraph 4.2.4.13).

4.2.4.4 Single Tooth Pair Contact Factors Z_B and Z_D

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

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

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

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

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

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

where:

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

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

– for helical gearing where,

if $\varepsilon_\beta \geq 1$

$$Z_B = Z_D = 1$$

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

Therefore:

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

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

4.2.4.5 Zone Factor Z_H

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

Zone factor Z_H shall be calculated using the following formula:

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

4.2.4.6 Material 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 Z_E shall be calculated using the following formula:

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

where:

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

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

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

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

4.2.4.7 Contact Ratio Factor Z_ε

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

Contact ratio factor Z_ε shall be calculated as follows:

– for spur gears using the following formula:

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}}; \tag{4.2.4.7-1}$$

– for helical gears using an appropriate alternative formula:
if $\varepsilon_\beta < 1$

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

if $\varepsilon_\beta \geq 1$

$$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}} \tag{4.2.4.7-3}$$

4.2.4.8 Helix Angle Factor Z_β

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

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

$$Z_\beta = \sqrt{\cos\beta} \qquad (4.2.4.8)$$

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

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

For this purpose the pitting may be determined:

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

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

The endurance limit for Hertzian contact stress depends mainly on:

– material composition, homogeneity and defects;
– mechanical properties;
– residual stress;
– hardening process, hardened layer depth, hardening gradient;
– material structure (forged, rolled, cast).

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

4.2.4.10 Life Factor for Contact Stress Z_N

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

The factor depends mainly on:

– material and hardening method;
– number of load cycles;
– Z_R, Z_v, Z_L, Z_W, Z_X factors.

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

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

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

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

These factors depend mainly on:

- the lubricating oil viscosity in the teeth contact area;
- the sum of momentary velocities on the teeth surfaces;
- the load;
- the relative radius of curvature at pitch point;
- roughness of tooth surface;
- hardness of pinion and wheel.

These factors shall be determined as follows:

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

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

where:

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

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

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

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

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

where:

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

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

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

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

where:

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

Note:
If $\sigma_{H\,lim} <$ 850 MPa, then C_{ZR} = 0.150.
If $\sigma_{H\,lim} >$ 1200 MPa, then C_{ZR} = 0.080.

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

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

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

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

where:
if the roughness is given as mean value – R_a

$$R_{Z1} = 6 \cdot R_{a1},$$

$$R_{Z2} = 6 \cdot R_{a2},$$

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

Note:
Roughness shall be measured at sides of several teeth.

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

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

where:

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

Note:
d_{b2} is negative for inner gearing.

4.2.4.12 Hardness Ratio Factor Z_W

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

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

Factor Z_W shall be calculated using the following formula:

$$Z_W = 1.2 - \frac{HB - 130}{1700} \tag{4.2.4.12}$$

where:

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

4.2.4.13 Size Factor Z_X

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

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

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

4.2.4.14 Contact Stress Safety Factor S_H

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

The safety factor shall be selected from Table 4.2.4.14.

Table 4.2.4.14

Gear type	S_H	
	Multiple set	Single set
Main propulsion gear	1.2	1.4
Auxiliary gear	1.15	1.2

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

4.2.5 Bending Stress in Gear Wheel Tooth Root

4.2.5.1 A criterion for bending stress in tooth root determines the permissible level of local tensile stress in the tooth root. The root bending stress σ_F and the permissible root bending stress σ_{FP} shall be calculated separately for the pinion and wheel. The value of σ_F shall not exceed that of σ_{FP}. The following formulae apply to gears with toothed rim thickness greater than 3.5 m_n for $\alpha_n \leq 25°$ and $\beta \leq 30°$. For greater values of α_n and β, the calculation results shall be confirmed experimentally or verified in accordance with the requirements specified in standard ISO 6336 – Method A.

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

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

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

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

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

where:
σ_{FE} – endurance limit for bending stress, [N/mm^2] (see 4.2.5.7);
S_F – safety factor for root bending stress (see 4.2.5.13);
Y_d – design factor (see 4.2.5.8);
Y_N – life factor for tooth root (see 4.2.5.9);
$Y_{\delta relT}$ – relative notch sensitivity factor (see 4.2.5.10);
Y_{RrelT} – relative surface finish factor (see 4.2.5.11);
Y_X – size factor (see 4.2.5.12).

4.2.5.4 Tooth Profile Factor Y_F

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

Tooth profile factor Y_F shall be determined using the formula below:

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

where:

h_F – bending moment arm for root stress caused by the force applied in the single tooth pair external contact, [mm];

s_{Fn} – tooth root chord in critical section, [mm];

α_{Fen} – pressure angle in the single tooth pair external contact at normal section, [°].

Note:

The quantities used to determine Y_F are shown in Fig. 4.2.5.5.

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

4.2.5.5 Stress Concentration Factor Y_S

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

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

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

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

where:

q_s – notch parameter determined using the formula below,

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

where:

ρ_F – tooth root fillet radius, [mm].

L – tooth bending factor determined using the formula below:

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

h_F , s_{Fn} see paragraph 4.2.5.4

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

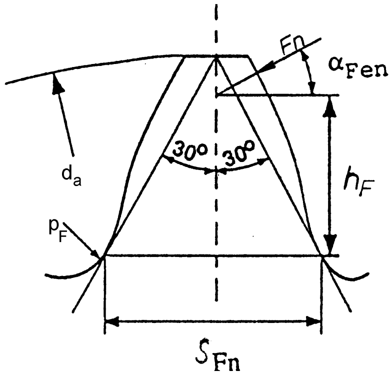


Fig. 4.2.5.5

4.2.5.6 Helix Angle Factor Y_β

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

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

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

It shall be taken that:
$\varepsilon_\beta = 1$ where $\varepsilon_\beta > 1$, and
$\beta = 30°$ where $\beta > 30°$.

4.2.5.7 Endurance Limit For Bending Stress σ_{FE}

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

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

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

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

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

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

4.2.5.8 Design Factor Y_d

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

Design factor Y_d for the load while the vessel is going astern shall be determined in accordance with Table 4.2.5.8.

Table 4.2.5.8

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

4.2.5.9 Life Factor for Tooth Root Y_N

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

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

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

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

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

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

The factor shall be taken as follows:
- for notch parameters (see paragraph 4.2.5.5) within $1.5 \leq q_s < 4$, $Y_{\delta relT} = 1$;
- for notch parameters beyond that interval, in accordance with the requirements specified in standard ISO 6336.

4.2.5.11 Relative Surface Finish Factor Y_{RrelT}

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

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

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

$$R_Z = 6R_a$$

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

4.2.5.12 Size Factor Y_X

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

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

Size factor Y_X shall be determined in accordance with Table 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 steels and through hardened steels
$Y_X = 0.85$	for $m_n \ge 30$	
$Y_X = 1.05 - 0.010\ m_n$	for $5 < m_n < 25$	skin hardened steels
$Y_X = 0.80$	for $m_n \ge 25$	

4.2.5.13 Safety Factor for Tooth Root Bending Stress S_F

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

Safety factor for tooth root bending stress S_F shall be determined in accordance with Table 4.2.5.13.

Table 4.2.5.13
Factor S_F

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

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

4.2.6 Shafts

Shafts which are not subjected to variable bending loads shall fulfil, to the applicable extent, the requirements specified in sub-chapters 2.2, 2.3, 2.4, 2.6 of *Part VI – Machinery Installations and Refrigerating Plants* (where applicable).

Main propulsion gears provided for ice-strengthened ships shall also fulfil the requirements specified in paragraphs 22.2.4 and 22.2.10 of *Part VI – Machinery Installations and Refrigerating Plants.*

4.2.7 Gear Wheels' Manufacture – General Notes

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

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

Friction factors for the calculation of shrink fit shall be taken in accordance with Table 4.2.7.2.

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

Table 4.2.7.2
Friction factors for shrink-fit calculation

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 oil penetration	0.18	0.14

4.2.8 Bearing System

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

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

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

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

4.2.9 Gearcases

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

Inspection openings shall be provided in gearcases to enable the teeth of pinions and of wheels to be readily examined.

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

4.2.10 Lubrication

4.2.10.1 Lubrication system shall ensure proper supply of oil to the bearings, teeth and other parts which need lubrication. The requirements specified in paragraph 13.1.3 of *Part VI – Machinery Installations and Refrigerating Plants* shall be fulfilled.

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

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

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

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

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

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

4.3 Disengaging and Flexible Couplings

4.3.1 General Requirements

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

4.3.1.2 Documentation concerning flexible couplings (see paragraph 1.3.3.9) shall include the following characteristics:

T_{KN} – rated torque for continuous operation;
$T_{K\max}$ – maximum torque for operation in transient conditions;
T_{KW} – allowable dynamic torque for the full range of torques from 0 to T_{KN} ;
$C_{T\,DYN}$ – dynamic stiffness for the full range of torques T_{KN} and T_{KW};
– rotational speed limit;
– allowable torque transmitted by the angular displacement limiter (where provided).

Additionally – for information – the following data shall be provided:

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

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

4.3.1.4 Flange connections and connecting bolts shall fulfil the requirements specified in sub-chapter 2.6, *Part VI – Machinery Installations and Refrigerating Plants* and, in the case of keyless connections – also to requirements specified in sub-chapter 2.8 of this *Part*.

4.3.2 Flexible Couplings

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

4.3.2.2 If the requirement specified in paragraph 4.3.2.1 is not fulfilled, the static torque breaking elements made from rubber or other synthetic materials shall not be less than eight times the value of the coupling rated torque.

4.3.2.3 The static torque breaking flexible elements in generating sets shall not be less than the torque resulting from the short-circuit current. Where relevant data are unavailable, the breaking torque shall not be less than 4.5 times as much as the coupling rated torque.

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

4.3.3 Disengaging Couplings

4.3.3.1 Disengaging couplings of main engines shall be controlled from the main engine control stand, and shall be fitted also with the arrangements for local control.

The control devices shall ensure so smooth engagement of the coupling that the momentary dynamic load does not exceed the maximum torque specified by the manufacturer or double rated engine torque.

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

4.3.4 Emergency Means

Where the propeller shaft is driven through:

– a hydraulic or electromagnetic transmission,
– a hydraulic or electromagnetic clutch,

provision shall be made for maintaining the ship motion with a speed necessary for its steerability in case of failure of the above-mentioned couplings.

5 AUXILIARY MACHINERY

5.1 Power-driven Air Compressors

5.1.1 General Requirements

5.1.1.1 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 preclude any possibility of its adjustment or disconnection after being fitted on the compressor.

5.1.1.3 Compressor crankcases of more than 0.5 m^3 in volume shall be fitted with safety valves which fulfil the requirements specified in paragraph 2.2.5.

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

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

5.1.2 Crankshaft

5.1.2.1 The method of verifying calculations specified in paragraphs 5.1.2.3 and 5.1.2.4 applies to the steel crankshafts of naval 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 Crankshafts shall be made of steel having tensile strength R_m ranging from 410 to 780 MPa.

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

Crankshafts may be made of nodular cast iron with a tensile strength $500 \le R_m \le 700$ MPa, in accordance with the requirements specified in Chapter 15, *Part IX – Materials and Welding*. Crankshafts with other dimensions than those determined by the formulae given below may be applied subject to PRS acceptance in each particular case, provided that complete strength calculations are submitted.

5.1.2.3 Crank pin diameter (d_k) of the compressor shall not be less than that determined in accordance with the formula below:

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

where:

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 the design pressure at high-pressure side in accordance with the requirements specified in paragrahs 21.2.2 and 21.2.3, *Part VI – Machinery Installations and Refrigerating Plants*;

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

- $L = L'$, where one crank is arranged between two main bearings (L' – actual distance between centres of main bearings),
- $L = 1.1L'$, where two cranks with 180° angle are arranged between two main bearings;

S – piston stroke, [mm];

K, f, φ – coefficients determined 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 coefficient *K*

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

Table 5.1.2.3-2
Values of coefficient *f*

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

Table 5.1.2.3-3
Values of coefficient φ

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 in accordance with the formula below:

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

where:

d_k – see formula 5.1.2.3-1;
d_0 – co-axial hole diameter, [mm];
d_a – actual 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 in accordance with the formula below:

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

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

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

where:
$a = 0.9$ for shafts with entire surface nitrided or submitted to other kind of heat treatment accepted by 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 relevant provisions of paragraph 5.1.2.3;
C_1 – distance from the centre of the main bearing to the mid-plane of the crank web, [mm]; where two cranks are arranged between two main bearings, the distance to the midplane of the web located further from the support under consideration shall be taken;
b – breadth of crank web, [mm];
f_1 – coefficient taken in accordance with Table 5.1.2.4-3;
R_m – tensile strength, [MPa].

Table 5.1.2.4-1
Values of coefficient ψ_1

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

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, coefficient ψ_1 shall be taken as for $\varepsilon/h_k = 0$.

Table 5.1.2.4-2
Values of coefficient ψ_2

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

For d_k – see formula 5.1.2.3-1.

Intermediate values of the coefficients specified in Tables 5.1.2.4-1 and 5.1.2.4-2 shall be determined by linear interpolation.

Table 5.1.2.4-3
Values of coefficient f_1

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

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

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

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

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

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

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. Application of overflow valves for this purpose is not recommended.

5.2.1.6 Strength calculation

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

5.2.1.7 Self-priming pumps

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

5.2.2 Additional Requirements for Flammable Liquid Pumps

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

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

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

5.3 Fans, Air Blowers and Turbochargers

5.3.1 General Requirements

5.3.1.1 The requirements specified in this sub-chapter apply to fans intended for systems covered by requirements specified in *Part VI – Machinery Installations and Refrigerating Plants*, as well as to internal combustion engine turbo-blowers and boiler air-blowers.

5.3.1.2 Impellers of fans and air blowers, including couplings, as well as the assembled rotors of turbochargers shall be dynamically balanced in accordance with the requirements specified in paragraph 4.1.2.

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

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

5.3.1.5 Strength calculation

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

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

5.3.2 Additional Requirements for Pump Room Fans

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

5.3.2.2 Terminals of ventilation ducts shall be protected against the entry of foreign matter 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 not sparking if in normal conditions as well as in abnormal conditions there is no risk of sparks generation. Casing and rotating parts of fan shall be made of such materials, which do not cause electric charge accumulation, and the fans installed shall be properly earthed to the hull of ship in accordance with the requirements of *Part VIII – Electrical Installations and Control Systems*.

5.3.2.4 Except the cases specified in paragraph 5.3.2.5, rotors and fan casings in way of rotor shall be made of such materials which do not generate sparks, as confirmed by appropriate tests.

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

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

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

.1 rotor made of an aluminium or magnesium alloy and steel casing, irrespective of the radial clearance value,

.2 casing made of an aluminium or magnesium alloy and steel rotor, irrespective of the radial clearance value,

.3 any combination of rotor and casing made of steel with the design radial clearance less than 13 mm.

5.4 Oil and Fuel Separators

5.4.1 General Requirements

5.4.1.1 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 assembly.

5.4.1.2 The arrangement of case and drum set shall be such that the resonant rotational speed of both empty drum and drum filled with liquid exceeds the rated number of revolutions.

The resonant speed lower than the rated one may be accepted, provided that long-time reliable operation of the separator has been confirmed.

5.4.1.3 The construction of clutches shall preclude sparking and their heating in all operating conditions and shall ensure effective heat transmission from the working surfaces.

5.4.2 Strength Calculations and Equipment of Separators

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

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

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

6 DECK MACHINERY

6.1 General Requirements

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

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

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

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

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

6.1.5 Measurement and control 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 shall also fulfil the requirements specified in Chapter 7.

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

6.2 Steering Gears and Their Installation on Board Ship

6.2.1 General Requirements

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

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

[*)] For definition of main steering gear – see 1.2 of *Part III – Hull Equipment*.

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

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

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

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

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 Where the main steering gear* comprises two or more identical power units, an auxiliary steering gear* need not be fitted, provided that:

- **.1** in a passenger ship, the main steering gear* is capable of operating the rudder as required in paragraph 6.2.1.1 while any one of the power units* is out of operation;
- **.2** in a cargo ship, the main steering gear* is capable of operating the rudder as required in paragraph 6.2.1.1 while operating with all power units*; and
- **.3** the main steering gear* is so arranged that after a single failure in its piping system or in one of the power units the defect can be isolated so that steering capability can be maintained or speedily regained.

6.2.1.6 Hydraulic steering gear with mechanical drive shall be provided with:

- **.1** device for keeping the hydraulic oil clean adequate to type and design of the hydraulic system;
- **.2** low level alarm of hydraulic oil in each circulating tank (visual and audible alarm signals shall be received in the wheelhouse and engine room).

A spare tank of hydraulic oil shall be provided, the capacity of which shall be sufficient for filling at least one of the power units, including the circulation tank. The spare oil tank shall be equipped with a gauge for measuring the tank content and permanently connected to hydraulic pipe in a way which allows for easy filling of the hydraulic system from a post in steering gear compartment.

6.2.1.7 Each part of hydraulic power system, that can be separated from the system and subjected to load from the drive source or by external forces (caused by water pressure exerted on the rudder blade) shall be fitted with relief valves set to

---

*) For definition of auxiliary steering gear and power unit – see 1.2 *of Part III – Hull Equipment*.

a pressure not exceeding the design pressure, but not less than 1.25 times the rated pressure of the system. The minimum output of relief valve(s) shall not be less than 1.1 of total capacity of pumps connected to it. In no case the pressure rise shall exceed 1.1 times the setting of the relief valves, the change of oil viscosity in extreme ambient conditions shall be taken into account. Means of sealing with lead shall be provided for the relief valves.

PRS recommends to carry out the following tests of the relief valves:
– output (throughput),
– resistance to water (hydraulic) hammer.

6.2.1.8 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 Rudder Position Indicators

A rudder gear part rigidly coupled with the rudder stock (tiller, quadrant, etc.) shall be fitted with a dial, calibrated for accuracy not less than 1°, to indicate the position of the rudder related to ship's centre line.

6.2.1.10 Limit Switches

Each steering gear shall be provided with an arrangement for stopping its operation before the rudder reaches its limit switches, permanently fixed to the ship hull; the steering gear capability to move the rudder immediately in the opposite direction shall be maintained.

6.2.1.11 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.

6.2.1.12 Requirements concerning the electric drive and signalling are specified in sub-chapter 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 sub-chapter 2.6 of *Part III – Hull Equipment.*

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

Where applicable the given below standard warning label shall be fitted at steering post in the wheelhouse or the statement ought to be included into ship "Procedures manual".

Warning label in Polish:

UWAGA:

Gdy oba zespoły energetyczne maszyny sterowej pracują jednocześnie, w pewnych warunkach ster może nie reagować na zadane polecenie. Należy wówczas wyłączać kolejno pompy aż kontrola nad sterem zostanie przywrócona.

In case of warning label in English:

CAUTION:

In some circumstances when 2 power units are running simultaneously the rudder may not respond to helm. If this happens stop each pump in turn until control is regained.

The label concerns the steering gears equipped with two identical power generators prepared for simultaneous operation and generally equipped with own, separate control system or two separate control circuits able to work in the same time.

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 shall not exceed 650 MPa. Upon special agreement with PRS, grey cast iron may be used for doubled, slightly loaded parts.

6.2.2.2 Pipes of the hydraulic steering systems shall fulfil the requirements relevant to class I piping and flexible joints specified in sub-chapter 1.16.2 of *Part VI – Machinery Installations and Refrigerating Plants.*

6.2.2.3 Piping shall be so made as to enable easy switching on and off individual cylinders and units and shall additionally fulfil the requirements specified in Chapter 7.

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

6.2.2.4 Hydraulic steering gear pumps shall be provided with protective means to prevent reverse rotation of an inoperative pump or with automatic arrangements to shut off the flow of liquid through the inoperative pump.

6.2.2.5 Where simultaneous operation of more than one steering gear or power unit is provided, then the risk of hydraulic lock shall be taken into account in the case of failure of singular power unit or control system.

If such risk can not be eliminated, a visual and audible alarm shall be provided in the wheelhouse for warning against the loss of steering capability and for identification of damaged system. Appropriate instructions for switching off the damaged system shall also be displayed in the wheelhouse.

Such an alarm shall be activated (for instance) if:

– setting of variable capacity pump is different than the value set by the control system,
– three-way, full flow valve or a similar device of fixed capacity pump is in wrong position.

6.2.3 Construction and Strength Calculation

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

Welded parts and welding procedure are liable to PRS approval. All welded joints within hydraulic cylinders or interconnected parts situated in lines of the force flux, shall, as a rule, be 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 sub-chapter 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 shall not be less than the greater of the below values:

– 1.25 of rated pressure (i.e. this corresponding to M_{ZN} torque), or
– assumed safety valve setting.

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

6.2.3.4 The stresses in considered part shall not exceed the following values, whichever less:

$$R_m/A \quad \text{or} \quad R_e/B$$

where:

R_m – tensile strength, [MPa];
R_e – physical yield point or proof stress ($R_{0.2}$), [MPa].

The values of safety factors A and B are specified in Table 6.2.3.4.

Table 6.2.3.4
Values of safety factors *A* and *B*

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 paragraph 2.6.2.2 of *Part III – Hull Equipment*) shall have strength not lower than that of the rudder stock.

6.2.4 Connection to Rudder Stock

6.2.4.1 Connection of the steering gear to the elements rigidly fixed to the rudder stock shall be such as to preclude the steering gear damage due to axial displacement of the rudder stock.

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 sub-chapter 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 stand-by steering gear shall also be of self-locking type or may be fitted with device locking the gear in desired position, provided that there is a possibility to change this position.

6.2.5.2 The main hand-operated steering gear shall fulfil the requirements specified in paragraph 6.2.1.1 – when handled by one man with a force not exceeding 120 N applied to the steering wheel handles and with the number of revolutions not greater than $9/R$ when putting the rudder from hard over to hard over (R – radius of steering wheel handle measured from the wheel axis of rotation to the mid-length of the handle, [m]).

6.2.5.3 The stand-by hand-operated main steering gear shall fulfil the requirements specified in paragraph 6.2.1.2 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 in paragraph 6.2.1.7.

For the stand-by hand-operated steering gear, the requirements specified in paragraph 6.2.1.7 need not be fulfilled.

6.2.6 Pump Type Test

The pumps of hydraulic power units shall be subjected to type test. The test duration shall be 100 hours at least. The test stand shall be arranged for idle running of the pump, as well as for the pump operation with maximum capacity at maximum working pressure. The idle running test periods shall be performed alternately with the periods of full load operation. The transition from one operating condition into the other one shall be carried out at least as quickly as during the operation on board the ship. No abnormal heating, vibration or other irregularities of pump operation can occur during the whole time of test duration. Upon completion of the test the pump shall be dismantled and its parts subjected to inspection.

This test may be omitted for the power units, for which the reliability has been confirmed by operational trials during service of the ship.

6.2.7 Tests on Board Ship

6.2.7.1 The steering gear shall be subjected to tightness and operating tests after its installation on board the ship.

6.2.7.2 The scope of at sea trials in the presence of PRS surveyor shall include:

.1 checking compliance with the requirements specified in paragraphs 6.2.1.1 and 6.2.1.2 regarding the rudder deflection, by main and standby steering gear. In the case of controlled pitch propeller, the pitch shall be set to the maximum value for nominal engine rotation speed full ahead.

If the vessel cannot be tested at the deepest draught, steering gear trials shall be conducted at a displacement as close as reasonably possible to full-load displacement as required by Section 6.1.2 of ISO 19019:2005 on the conditions that either the rudder is fully submerged (zero speed waterline) and the vessel is in an acceptable trim condition, or the rudder load and torque at the specified trial loading condition have been predicted and extrapolated to the full load condition.

In any case for the main steering gear trial, the speed of ship corresponding to the number of maximum continuous revolution of main engine and maximum design pitch applies.

.2 power units of the steering gear testing and their switching on/off;

.3 switching off and cutting off 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 sub-chapter 5.5 of *Part VIII – Electrical Installations and Control Systems*;

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

.7 checking communication means between wheelhouse, machinery room and steering gear compartment;

.8 alarm system and indicators operation in accordance with the requirements specified in paragraph 6.2.2.5 as well as in sub-chapters 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) and signalling system check.

6.3 Windlasses

6.3.1 Drive

6.3.1.1 Power of the windlass driving motor shall ensure continuous heaving up a chain cable with an anchor of normal holding force for at least 30 minutes with a speed at least 9 m/min (0.15 m/s) and chain cable pull P_1 or P_2 on the cable lifter not less than that determined in accordance with the formulae below:

– for all types of ships, except service ships:

$$P_1 = 9.81ad^2, \quad [\mathrm{N}] \qquad (6.3.1.1\text{-}1)$$

where:

a – coefficient taking the following values:
3.75 for steel grade 1 chain cables,
4.25 for steel grade 2 chain cables,
4.75 for steel grade 3 chain cables,
(for chain cable steel grades – see 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 subject to PRS acceptance in each particular case;

– for service ships assigned an additional mark **SUPPLY VESSEL** in their symbol of class:

$$P_2 = 11.1(qh+G), \quad [\mathrm{N}] \qquad (6.3.1.1\text{-}2)$$

where:

q – mass of 1 m of chain, [kg/m];

G – the anchor mass, [kg];

h – design anchoring depth, [m], however not less than:
200 – for ships with the equipment number not exceeding 720,
250 – for ships with the equipment factor exceeding 720 (for the equipment factor – see sub-chapter 1.7 of *Part III – Hull Equipment*).

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

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 extract the anchor from the bottom, the windlass power unit shall produce, in a rated working cycle, a continuous pull of one cable lifter equal at least $1.5P_1$ for a period not less than 2 minutes. However, the requirement specified in paragraph 6.3.1.1 concerning the heave-up speed need not be fulfilled.

6.3.2 Brakes and Clutches

6.3.2.1 Windlasses shall be fitted with disengageable clutches between the cable lifter and the drive shaft.

Windlass with a gear mechanism which is not of self-locking type shall be fitted with automatic cable lifter brakes to prevent paying out of the chain in case of the power failure or power unit failure.

The automatic cable lifter brake shall be capable of maintaining the cable lifter pull not less than $1.3P_1$ or $1.3P_2$.

6.3.2.2 Cable lifters shall be fitted with brakes which are capable to stop safely paying out of the chain. This brake shall ensure holding the chain cable without slip on the brake when the cable lifter is declutched and the chain cable loaded with a force:

.1 equal to 0.45 times the breaking load of the cable – for anchor gear with a stopper for holding the chain cable of a ship lying at anchor;

.2 equal to 0.8 times the breaking load of the cable – for anchor gear without the stopper mentioned in .1.

The force applied to the brake drive handle shall not exceed 740 N.

6.3.3 Cable Lifters

6.3.3.1 Cable lifters shall have not less than five cams. For horizontal axis cable lifters, the wrapping angle shall not be less than 115°, whereas for vertical axis cable lifters – not less than 150°.

6.3.3.2 Cable lifters shall be so designed that the detachable links (Kenter links) can pass both in horizontal and vertical position.

6.3.4 Overload Protection

Where the maximum torque of the windlass motor may cause the (equivalent) stress in the windlass components exceeding 0.95 the yield point of the material used, or a rise to the force on the sprocket exceeding 0.5 the test load, a safety coupling shall be installed between the motor and the windlass to prevent overload.

6.3.5 Strength Calculation

Stress of the windlass parts being in flux of the strain lines shall not exceed:

$0.4\ R_e$ – when loaded with rated power of driving motor,

$0.95\ R_e$ – when loaded with the maximum torque of driving motor, 0.95 R_e – when subjected to maximum load caused by anchor cable held by brake – in accordance with paragraph 6.3.2.2; this requirement applies to those parts of windlass which are subjected to the above mentioned load;

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

When designing windlasses, special attention shall be paid to:

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

6.3.6 Additional Requirements for Windlasses with Remote Control

6.3.6.1 Windlasses with remote control shall be fitted with an automatic brake so that the speed of chain cable release, with the chain sprocket disengaged from the drive, does not exceed 3 m/s and is not less than 1.33 m/s, except the initial run.

In ships with the equipment number 400 and less, the automatic brake is not required.

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.

6.3.7 Strength Requirements to Resist Green Sea Forces

6.3.7.1 The requirements specified in this sub-chapter apply to the securing of windlasses located on the exposed deck over the forward $0.25L$, in ships of length 80 m or more, where the height of the exposed deck in way of the windlass location is less than $0.1L$ or 22 m above the summer load waterline, whichever is lesser.

6.3.7.2 Where mooring winches are integral with the anchor windlass, they shall be considered as part of the windlass.

6.3.7.3 The following pressures and associated areas shall be applied (Fig. 6.3.7.3):

– 200 kPa normal to the shaft axis and away from the forward perpendicular, over the projected area in this direction,
– 150 kPa parallel to the shaft axis and acting both inboard and outboard separately, over the multiple of f times the projected area in this direction,

where:

$$f = 1 + \frac{B}{H}\text{, however not greater than 2.5}$$

B – width of windlass measured parallel to the shaft axis, [m]
H – overall height of windlass, [m].

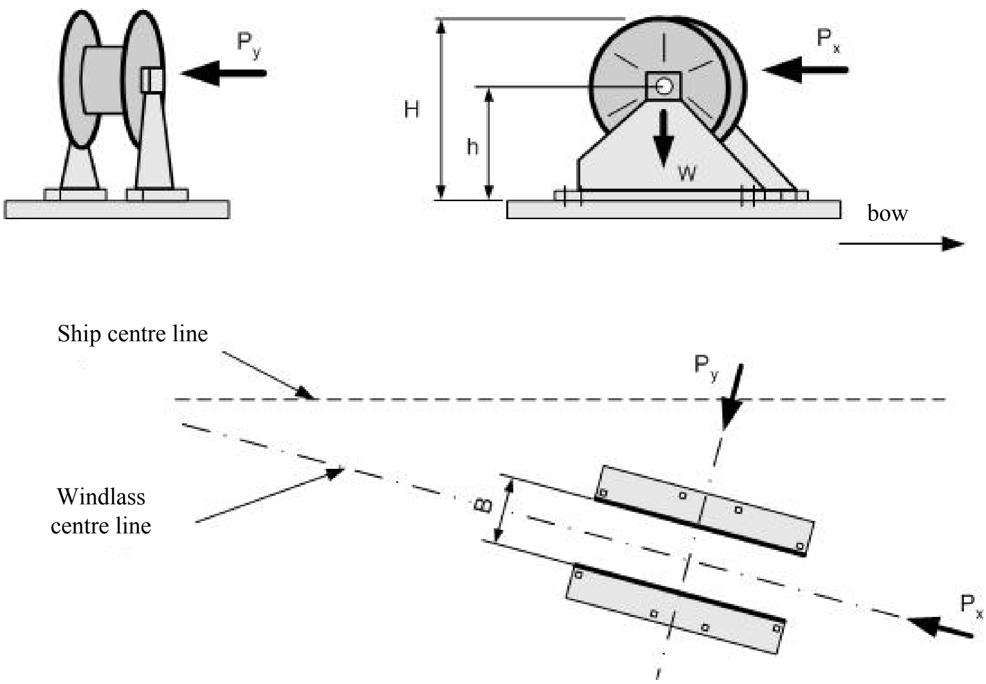


Fig. 6.3.7.3. Direction of forces and weight

Note:
Force P_y shall be examined from both inboard and outboard direction separately (see 6.3.7.5).

6.3.7.4 Forces in the bolts, chocks and stoppers securing the windlass to the deck shall be calculated.

The windlass supported by N bolt groups, each containing one or more bolts is shown in Fig. 6.3.7.4.

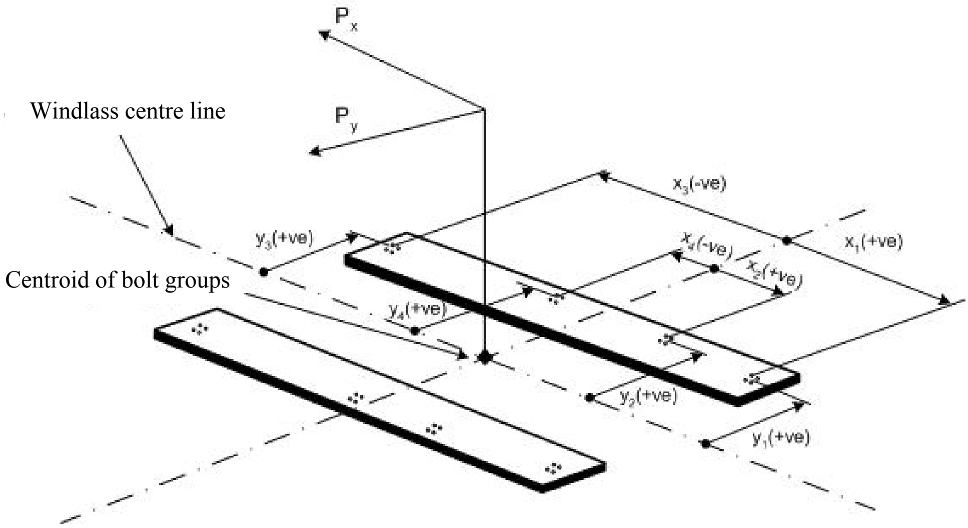


Fig. 6.3.7.4. Sign Convention

6.3.7.5 The axial force R_i in bolt group i, positive in tension, shall be calculated in accordance with the formula below:

$$R_{xi} = \frac{P_x \cdot h_{xi} \cdot A_i}{I_x}, \quad \text{[kN]} \tag{6.3.7.5-1}$$

$$R_{yi} = \frac{P_y \cdot h_{yi} \cdot A_i}{I_y}, \quad \text{[kN]} \tag{6.3.7.5-2}$$

$$R_i = R_{xi} + R_{yi} - R_{si}, \quad \text{[kN]} \tag{6.3.7.5-3}$$

where:

P_x – force acting normal to the shaft axis, [kN],

P_y – force acting parallel to the shaft axis, either inboard or outboard, whichever gives the greater force in bolt group i, [kN],

h – shaft height above the windlass mounting, [cm],

x_i, y_i – x and y coordinates of bolt group i from the centroid of all N bolt groups, positive in the direction opposite to that of the applied force, [cm],

A_i – cross-sectional area of bolt groups in group i, [cm^2],

I_x – $\Sigma A_i x_i^2$ for N bolt groups, [cm^4],

I_y – $\Sigma A_i y_i^2$ for N bolt groups, [cm^4],

R_{si} – static reaction at bolt group i, due to weight of windlass, [kN].

6.3.7.6 Shear forces F_{xi}, F_{yi} applied to the bolt group i, and the resultant combined force F_i shall be calculated in accordance with the formula below:

$$F_{xi} = \frac{P_x - \alpha g M}{N}, \quad [\text{kN}] \tag{6.3.7.6-1}$$

$$F_{yi} = \frac{P_y - \alpha g M}{N}, \quad [\text{kN}] \tag{6.3.7.6-2}$$

$$F_i = \sqrt{F_{xi}^{\ 2} + F_{yi}^{\ 2}}, \quad [\text{kN}] \tag{6.3.7.6-3}$$

where:
α – friction factor, to be taken as 0.5,
M – mass of windlass, [t],
g – gravity acceleration, [m/s^2],
N – number of bolt groups.

6.3.7.7 The requirements specified in sub-chapter 6.3.7 do not apply to bulk carriers which are subject to the requirements of *Publication No. 84/P – Requirements Concerning the Construction and Strength of the Hull and Hull Equipment of Sea-going Bulk Carriers of 90 m in Length and Above – 2009.*

6.4 Mooring Winches

6.4.1 Drives

6.4.1.1 Mooring winch motor shall ensure uninterrupted heaving-in of a mooring line at a rated pull for a period of not less than 30 minutes.

The heaving-in speed of the mooring line when reeling the first layer on the drum, with the rated pull, shall be at least:
up to 80 kN – 0.25 m/s,
from 81 to 160 kN – 0.20 m/s,
from 161 to 250 kN – 0.16 m/s,
above 250 kN – 0.13 m/s.

The speed of heaving-in the mooring line by mooring head shall not exceed 0.3 m/s at the rated load.

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

6.4.1.2 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, shall not exceed 0.8 of the line breaking force.

6.4.1.3 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 sub-chapter 6.4.3.

6.4.2 Brakes

6.4.2.1 Mooring winch shall be provided with an automatic braking device holding the mooring line under tension of not less than 1.5 times the rated pull at the power loss or drive failure.

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

The force applied to the brake handle to exert such torque shall not 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 Stresses in the elements securing the mooring winch to the foundation and in load-bearing parts of the winch under mooring line breaking load exerted in the drum and in the mooring head at its mid-length shall not exceed 0.95 times the yield strength of their material.

Stresses in the winch components shall be determined taking account of all 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 Pull Force Automatic Control

6.4.4.1 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 shall not exceed 1.5 times nor be less than 1.05 times the pre-set hauling tension.

Mooring winches with remote control shall be provided with alarm devices giving the signal in the remote control station when the permissible pull has been exceeded. The alarm shall be given irrespective of the length of released line.

6.5 Towing Winches

6.5.1 Where automatic devices are used for governing the tension of the towline, provision shall be made for the continuous control of the tension. The tension indicators shall be fitted at the towing winch and in the wheelhouse.

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 fulfil the requirements specified in paragraph 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 Brakes of towing winches shall fulfil the following requirements:

.1 The towing winch shall be provided with an automatic braking device to stop the winch when the pull is at least 1.25 times the rated pull in case of power decay or failure in the driving system.

.2 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 also be provided with a manual control system. The brake design shall provide for quick release of the brake to ensure free heaving-in the towline.

6.5.6 The towline shall be so fixed to the winch drum that in the case of full release of the towline, the towline 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 equivalent stresses occurring in the components which may be subjected to acting forces caused by the above-mentioned loads shall not exceed 0.95 of this component material yield point.

6.5.8 The strength characteristics of the towline intended for working with the towing gear shall be marked on the towing gear.

7 HYDRAULIC DRIVES

7.1 Application

7.1.1 The requirements specified in this Chapter apply to all hydraulic appliances and systems aboard the ship except for those mentioned in paragraph 7.1.2.

7.1.2 Independent appliances – cased in individual housings – fulfilling recognised standards which are not associated with the vessel propulsion, steering and manoeuvring need not fulfil the requirements specified in this Chapter.

7.2 General Requirements

7.2.1 The arrangements for the storage, distribution and utilisation of hydraulic fluids intended to be used at high pressure in hydraulic systems shall be such as to ensure the safety of the ship and persons on board. In locations where sources of ignition are present, the hydraulic oil system shall fulfil the requirements specified in paragraphs 2.5.3 and 2.5.4 of *Part VII* as well as sub-chapter 1.16.2, paragraphs 1.10.4; 1.16.6.3; 1.16.6.4; 9.4.2; 12.2.2; 12.2.3 and sub-chapter 12.11 contained in *Part VI – Machinery Installations and Refrigerating Plants*.

7.2.2 Hydraulic oil shall not be a source of corrosion in the hydraulic system. Its ignition temperature shall not be less than 150 ºC. Hydraulic oil shall be suitable for working within the range of operating temperatures of the hydraulic arrangement or system. In particular, this regards the range of viscosity change.

7.2.3 Hydraulic arrangements shall be protected with relief valves. Unless provided otherwise in other parts of the *Rules*, the opening pressure of the relief valve shall not exceed 1.1 of the maximum working pressure.

The nominal flow rate of the relief valves shall be so selected that the generated hydraulic oil pressure does not exceed 1.1 of the pre-set pressure of valve opening at the maximum pump output.

7.2.4 In the case of hydraulic systems and appliances working continuously such as hydraulic main propulsion, steering gears, hydrodynamic couplings, the possibility of cleaning oil filters without stopping the system shall be provided.

7.2.5 A failure of the hydraulic system shall not cause damage to the associated piece of machinery or equipment.

7.2.6 Hydraulic systems of steering gears, as well as hydraulic systems actuating variable pitch propellers shall not have any connection with other hydraulic systems.

7.2.7 Where a feed pipe of hydraulic-powered windlasses has a connection to other hydraulic systems, it shall be supplied by two independent pump systems, each of which shall ensure continuous operation of the windlass having the requirement specified in sub-chapter 6.3.1 fulfilled.

7.3 Flammable Hydraulic Oil Tanks

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

.1 in the case of tanks not adjacent to vessel shell plating which are situated outside the machinery compartments of Category A, in compartments situated above the load waterline where there are not sources of ignition such as internal combustion engines or boilers, the application of cylindrical level indicator glasses is permitted.

.2 in the case of tanks with the capacity less than 100 dm^3, situated in machinery compartments, PRS may consider acceptance of cylindrical level indicator glasses.

7.4 Pipe Connections

Pipe connections shall fulfil the requirements specified in sub-chapter 1.16 of *Part VI – Machinery Installations and Refrigerating Plants* and additionally:

.1 pipes installed on board the vessel shall have the inside surface as clean as it is required for hydraulic components;

.2 in pipelines with a nominal diameter less than 50 mm, threaded sleeve joints of the type approved by PRS shall be applied; however, the joints with the rubber washer may only be applied for connection of hydraulic components but not for connection of pipe segments;

.3 pipe joints without PRS approval may only be applied, subject to PRS acceptance in each particular case, where they fulfil the requirements specified in the relevant national standard and are provided with an appropriate inspection certificate;

.4 pipelines shall not have soldered joints;

.5 flexible hoses with connection fittings shall fulfil the requirements specified in paragraph 1.16.2.9 of *Part VI – Machinery Installations and Refrigerating Plants* and shall be type approved by PRS. Subject to PRS acceptance in each particular case, fireproof hoses without PRS approval may be applied, except in the installations of steering gears and hydraulic control systems of watertight doors, ports and ramps in the vessel shell, provided they fulfil the relevant national standard and have an appropriate inspection certificate.

7.5 Hydraulic Components

7.5.1 Hydraulic accumulators shall fulfil the strength requirements for pressure vessels of the particular class. Each accumulator, which may be cut off the hydraulic system shall be provided with an individual relief valve. A safety valve or other protecting device shall be installed on the gas side to prevent overpressure.

7.5.2 Hydraulic cylinders shall fulfil the strength requirements for pressure vessels of the particular class.

7.5.3 Hydraulic cylinders shall be type-approved by PRS.

7.5.4 Subject to PRS acceptance in each particular case, hydraulic cylinders which are not type-approved by PRS may be applied if they fulfil the requirements specified in the relevant national standard and are provided with an appropriate inspection certificate.

7.5.5 Valves, pumps, hydraulic motors and high pressure filters shall be type-approved by PRS

7.5.6 Hydraulic cylinders which do not fulfil the requirements specified in paragraphs 7.5.3 and 7.5.4 as well as other hydraulic components which do not fulfil the requirement specified in paragraph 7.5.5 may be applied if they have been manufactured under PRS survey in accordance with the approved documentation and have been approved by PRS surveyor on the manufacturer's premises in accordance with the approved testing programme.

7.5.6.1 Hydraulic system components in the power actuating or hydraulic servo systems controlling the power systems of the steering gear (e.g. solenoid valves, magnetic valves) shall be considered as part of the steering gear control system and shall be duplicated and separated.

Hydraulic system components in the steering gear control system that are part of a power unit may be regarded as being duplicated and separated when there are two or more separate power units provided and the piping to each power unit can be isolated.

7.6 Testing

7.6.1 Tests shall be performed in accordance with the testing programme approved by PRS.

7.6.2 Testing programme shall determine the type and scope of tests, acceptance criteria, test site and – if necessary – testing procedure.

7.6.3 Tests shall include:

.1 pressure tests of piping in accordance with the requirements specified in sub-chapter 1.5.4 of *Part VI – Machinery Installations and Refrigerating Plants*;

.2 post-rinsing check of piping cleanness;

.3 operating tests;

.4 hydraulic oil check for impurities before and after operating 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 divided into classes as indicated in Table 8.1.

Table 8.1

Kind of equipment	Class I	Class II	Class III
Steam boilers, including exhaust gas heated economiser water boilers for water temperature over 115°C, steam superheaters and steam reservoirs, thermal oil heaters	$p > 0.35$	$p \leq 0.35$	–
Steam-heated steam generators	$p > 1.6$	$p \leq 1.6$	–
Pressure vessels and heat exchangers	$p > 4.0$ or $t > 350$ or $s > 35$	$1.6 < p \leq 4.0$ or $120 < t \leq 350$ or $16 < s \leq 35$	$p \leq 1.6$ and $t \leq 120$ and $s \leq 16$
Pressure vessels and heat exchangers containing toxic, inflammable or explosive media	irrespective of parameters	–	–

p – design pressure*), [MPa];
t – design wall temperature, [°C];
s – wall thickness, [mm].

8.2 Strength Calculations

8.2.1 General Requirements

8.2.1.1 Wall thicknesses determined by calculation are the lowest permissible values under normal operating conditions. The formulae and strength calculation methods do not take into account the manufacturer's tolerances for thickness and these shall be added as special allowances to the design thickness values.

Additional stresses due to external loads (axial forces, bending moments, torques) imposed on the calculated parts (particularly loads due to dead mass or the mass of attached parts) shall be taken into account on PRS' request.

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 this *Part* of the *Rules* shall be determined on the basis of experimental data and recognized theoretical calculations, and are subject to special consideration by PRS in each particular case.

---

*) For definition of design pressure – see 1.2 of *Part VI – Machinery Installations and Refrigerating Plants.*

8.2.2 Design Pressure

8.2.2.1 Where hydrostatic pressure is greater than 0.05 MPa, the design pressure shall be increased by that value.

8.2.2.2 For uniflow and forced-circulation boilers, the design pressure shall be determined taking account of the hydrodynamic resistance in boiler components at the rated capacity.

8.2.2.3 For flat walls subjected to pressure from both sides, the design pressure shall be taken as the greatest of the acting pressures. Walls in the form of curved surfaces which are subjected to pressure from both sides shall be calculated for the greatest outer and inner pressures. If the pressure on one side of the flat wall or the wall in the form of curved surface is lower than the atmospheric pressure, then the maximum pressure on the other side of the wall increased by 0.1 MPa shall be taken as the design pressure.

8.2.2.4 The design pressure for economizers shall be taken equal to the total sum of the working pressure in the steam manifold and the hydrodynamic resistance in the economizer, piping as well as valves 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 not be taken lower than indicated in Table 8.2.3.1

Table 8.2.3.1

Item	Components of boilers, pressure vessels and heat exchangers and operating conditions thereof	Design wall temperature
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	Not heated components[2)]	T_m

Notes:
1) – see paragraph 8.2.3.4;
2) – see paragraph 8.2.3.3;
T_m – maximum temperature of heated medium, [°C];
T_v – maximum temperature of heating medium, [°C].

8.2.3.2 Design temperature for steam superheater tubes at steam temperatures over 400 °C, as well as for tubes and manifolds of superheaters exposed to radiant heat shall be determined by calculation and is subject to PRS acceptance in each particular case.

8.2.3.3 A wall is considered to be non-heated if one of the following conditions is fulfilled:
– the wall is separated from the furnace or uptake by fire-resisting insulation and the distance between the wall and insulation is 300 mm or more;
– the walls is covered with fire-resisting insulation not exposed to radiant heat.

8.2.3.4 A wall is considered to be protected from radiant heat effect if one of the following conditions is fulfilled:
– the wall is covered with fire-resistant insulation;
– the wall is shielded by a closely spaced row of tubes (with a maximum clearance between the tubes in the row not exceeding 3 mm);
– the wall is shielded 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 not exceeding 2.5 times the outside tube diameter.

8.2.3.5 The design temperature of heated boiler walls and non-heated steam space walls of boilers shall be taken 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 running higher than 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 paragraph 9.1.9.

8.2.3.7 Design temperature for tank walls and pressure vessel walls operating under refrigerant pressure shall be taken equal to 20°C, if higher temperatures are not likely to occur.

8.2.4 Strength Characteristics of Materials and Allowable Stresses

8.2.4.1 For steels with $(R_e/R_m) \le 0.6$, the strength characteristics shall be taken equal to physical yield point or proof stress R_e^t or $R_{0.2}^t$, as well as average creep strength $R_{z/100\,000/t}$ after 10^5 h, at design temperature t.

For steels with $(R_e/R_m) > 0.6$, R_m^t, tensile strength at design temperature t shall also be taken into account.

For steel loaded in the creep conditions (temperature exceeding 450 °C), irrespective of (R_e/R_m) ratio, 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.

The minimum values of R_e^t, $R_{0.2}^t$ and R_m^t and average values of $R_{1/100\,000/t}$ and $R_{z/100\,000/t}$ shall be taken for calculations.

8.2.4.2 For materials whose stress-strain curve does not show a specific yield stress, the tensile strength at the design temperature shall be taken for calculations.

8.2.4.3 For cast iron and non-ferrous alloys, the minimum value of ultimate tensile strength at normal temperature shall be taken for calculations.

8.2.4.4 When using non-ferrous materials and their alloys, it shall be taken into account that the heating during processing and welding reduces the strengthening effect achieved by cold processing. Therefore the strength characteristics to be used for strength calculations of the components and assemblies made of such materials shall be those applicable to their annealed condition.

8.2.4.5 Allowable stresses σ assumed for strength calculations shall be determined as the minimum out of the following three values:

$$\sigma = \frac{R_m^t}{\eta_m}, \quad \sigma = \frac{R_e^t}{\eta_e} \quad \text{or} \quad \sigma = \frac{R_{0.2}^t}{\eta_e}$$

$$\sigma = \frac{R_{z/100\,000/t}}{\eta_z}, \quad \sigma = \frac{R_{1/100\,000/t}}{\eta_p}$$

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 i $R_{0.2}^t$;
η_p – safety factor for creep point, $R_{1/100\,000/t}$.

For values of factors – see sub-chapter 8.2.5.

8.2.5 Safety Factors

8.2.5.1 For components made of steel forgings or rolled steel, subjected to internal pressure, the safety factors shall not be less than:

$$\eta_e = \eta_z = 1.6; \quad \eta_m = 2.7 \text{ and } \eta_p = 1.0.$$

For components subjected to external pressure, safety factors η_e, η_z and η_m shall be increased by 20%.

8.2.5.2 For components of boilers, heat exchangers and pressure vessels of Class II and Class III, made of steels with $(R_e/R_m) \le 0.6$, the safety factors may be reduced, however they shall not be less than:

$$\eta_e = \eta_z = 1.5; \quad \eta_m = 2.6.$$

8.2.5.3 For components of boilers, heat exchangers and pressure vessels made of cast steel and subjected to internal pressure, the safety factors shall not be less than:

$$\eta_e = \eta_z = 2.2; \quad \eta_m = 3.0 \text{ and } \eta_p = 1.0.$$

For components exposed to outer pressure, the safety factors η_e and η_m shall be increased by 20% (η_z remains unchanged).

8.2.5.4 Safety factors η_e and η_z for thermal loaded important 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 pipes subjected to pressure and other similar gas heated walls.

8.2.5.5 Safety factors η_m for components made of cast iron shall be taken not less than 4.8 – for internal and external pressure.

This factor for non-ferrous metals – shall not be less than 4.6 for internal pressure and 5.5 for external pressure. For conical walls, in the latter case, η_m shall not be taken less than 6.0.

8.2.6 Strength Factors

8.2.6.1 Strength factors of welded joints φ shall be determined in accordance with Table 8.2.6.1-1 depending on the joint type and welding process. For particular classes of boilers, pressure vessels and heat exchangers (see Table 8.1), strength factor φ shall not be less than that specified in Table 8.2.6.1-2.

Table 8.2.6.1-1

Welding process	Joint type	Weld type	φ
Automatic	Butt joints	Double-sided Single-sided with backing Single-sided without backing	1.0 0.9 0.8
	Overlap joint	Double-sided Single-sided	0.8 0.7
Semi-automatic and manual	Butt joints	Double-sided Single-sided with backing Single-sided without backing	0.9 0.8 0.7
	Overlap joint	Double-sided Single-sided	0.7 0.6

Notes to Table 8.2.6.1-1:
1. Full penetration shall be achieved in each case.
2. For welded joints made in electroslag process, $\varphi = 1$ shall be taken.

Table 8.2.6.1-2

Kind of 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 Strength factor of cylindrical walls weakened by holes with identical diameter shall be taken equal to the least of the following three values:

.1 strength factor of cylindrical walls weakened by a longitudinal row or a field of equally spaced holes (Fig. 8.2.6.2-1), as determined using the formula below:

$$\varphi = \frac{a - d}{a} \tag{8.2.6.2.1}$$

.2 strength factor, reduced to the longitudinal direction, of cylindrical walls weakened by a transverse row or a field of equally spaced holes (Fig. 8.2.6.2-1), as determined using the formula below:

$$\varphi = 2\frac{a_1 - d}{a_1} \tag{8.2.6.2.2}$$

.3 strength factor, reduced to the longitudinal direction, of cylindrical walls weakened by a field of equally spaced staggered holes (Fig. 8.2.6.2-2 and Fig. 8.2.6.2-3), as determined using the formula below:

$$\varphi = k\frac{a_2 - d}{a_2}, \tag{8.2.6.2.3-1}$$

where:

φ – strength factor of walls weakened by holes;

d – diameter of the hole for expanded tubes or inner diameter of welded-on tubes and extruded branch pieces, [mm];

a – spacing between axes of two adjacent holes arranged along the wall, [mm];

a_1 – spacing between axes of two adjacent holes in the transverse (circumferential) direction, taken as the mean circumference arc length, [mm];

a_2 – spacing between axes of two adjacent holes in staggered rows [mm], as determined using the formula below:

$$a_2 = \sqrt{l^2 + l_1^2}, \quad \text{[mm]} \tag{8.2.6.2.3-2}$$

l – spacing between axes of two adjacent holes in the longitudinal direction (see Fig. 8.2.6.2-2 and Fig. 8.2.6.2-3), [mm];

l_1 – spacing between axes of two adjacent holes in the transverse or circumferential direction (see Fig. 8.2.6.2-2 and Fig. 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

l_1 / l	5.0	4.5	4.0	3.5	3.0	2.5	2.0	1.5	1.0	0.5
k	1.76	1.73	1.70	1.65	1.60	1.51	1.41	1.27	1.13	1.00

Note:
Intermediate values of k shall be determined by linear interpolation.

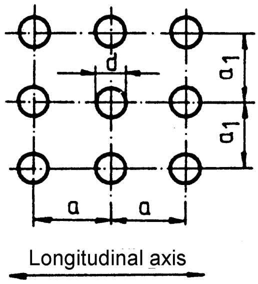


Fig. 8.2.6.2-1

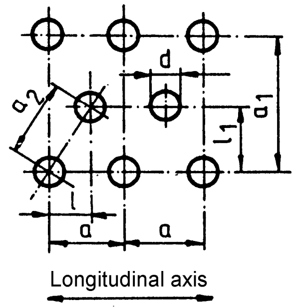


Fig. 8.2.6.2-2

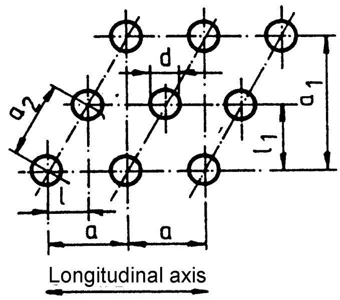


Fig. 8.2.6.2-3

8.2.6.3 Where rows or fields of equally spaced holes contain holes of different diameters, value d in the formulae for strength 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 taken as the value equal to the arithmetic mean of the two largest adjacent holes. In the case of uneven spacing between the holes of equal diameters, the lowest values of a, a_1 or a_2 , respectively, shall be applied in the formulae for strength factor determination.

8.2.6.4 In the case of weld seams with holes, the strength factor shall be taken as the product of the seam strength factor and the strength factor of the wall weakened by the holes.

8.2.6.5 For seamless cylindrical walls not weakened by a weld seam or row/field of holes, strength factor φ shall be taken as equal to 1.0. In no case factor φ shall be taken greater than 1.0.

8.2.6.6 Strength factor of walls weakened by holes for expanded tubes, as determined in accordance with formulae 8.2.6.2.1, 8.2.6.2.2, 8.2.6.2.3, shall not be taken less than 0.3. Calculations with the lesser value of the strength factor are subject to PRS acceptance in each particular case.

8.2.6.7 For walls of cylindrical components made of sheets with different thickness, joined by longitudinal weld seam, the thickness calculation shall be done separately for each sheet, taking account of the actual weakenings.

8.2.6.8 For tubes with longitudinal weld seam, the strength factor is subject to PRS acceptance in each particular case.

8.2.6.9 Strength factors for walls weakened by openings requiring full or partial strengthening shall be determined in acordance with sub-chapter 8.2.19.

8.2.6.10 Strength factors for flat flue sheets shall be determined in accordance with formula 8.2.6.2.1 for tangential and radial spacings respectively. The lesser obtained strength factor shall be taken for calculation of the flat flue sheet thickness.

8.2.7 Design Thickness Allowances

8.2.7.1 In every case where the design wall thickness allowance c is not expressly specified, it shall be taken at least 1 mm. For steel walls with more than 30 mm in thickness, as well as for walls of corrosion-resistant non-ferrous metals or high alloy materials, and for materials adequately protected against corrosion, e.g. by cladding or coating with a protective compound, the design thickness allowance may be waived subject to PRS acceptance in each particular case.

8.2.7.2 For pressure vessels and heat exchangers inaccessible for internal examination and for those whose are subjected to heavy corrosion or wear, PRS may require an increased allowance c to the design thickness.

8.2.8 Cylindrical and Spherical Elements and Tubes Subjected to Internal Pressure

8.2.8.1 The requirements specified in this sub-chapter apply where the following conditions are fulfilled:

$\frac{D_a}{D} \le 1.6$ – for cylindrical elements;

$\frac{D_a}{D} \le 1.7$ – for tubes;

$\frac{D_a}{D} \le 1.2$ – for spherical elements.

Cylindrical elements with a diameter $D_a \le 200$ mm shall be considered as tubes. For D_a , D – see paragraph 8.2.8.2.

8.2.8.2 Thickness of cylindrical walls and tubes shall not be less than that calculated in accordance with the formulae below:

$$s = \frac{D_a\ p}{2\sigma\varphi + p} + c, \quad \text{[mm]} \tag{8.2.8.2-1}$$

or

$$s = \frac{D\,p}{2\sigma\varphi - p} + c, \quad \text{[mm]} \tag{8.2.8.2-2}$$

s – wall thickness, [mm];
p – design pressure, [MPa];
D_a – outside diameter, [mm];
D – inside diameter, [mm];
φ – strength efficiency factor (see see sub-chapter 8.2.6);
σ – allowable stress (see paragraph 8.2.4.5), [MPa];
c – design thickness allowance (see see sub-chapter 8.2.7), [mm].

8.2.8.3 Spherical wall thickness shall not be less than those obtained from the formula:

$$s = \frac{D_a\,p}{4\sigma\varphi + p} + c, \quad \text{[mm]} \tag{8.2.8.3-1}$$

or

$$s = \frac{D\,p}{4\sigma\varphi - p} + c, \quad \text{[mm]} \tag{8.2.8.3-2}$$

For symbols – see paragraph 8.2.8.2.

8.2.8.4 Irrespective of the values obtained in accordance with formulae 8.2.8.2-1, 8.2.8.2-2, 8.2.8.3-1 and 8.2.8.3-2, the thickness of spherical and cylindrical walls and tubes shall not be less than:

.1 5 mm – for seamless and welded elements;
.2 12 mm – for tube plates with radial hole arrangement for expanded tubes;
.3 6 mm – for tube plates with welded-on and soldered-on tubes;
.4 specified in Table 8.2.8.4 – for tubes.

Thickness of tube walls heated by gas with temperature exceeding 800°C shall not be less than 6 mm.

Table 8.2.8.4

Tube outside diameter, [mm]	≤20	>20 ≤30	>30 ≤38	>38 ≤51	>51 ≤70	>70 ≤95	>95 ≤102	>102 ≤121	>121 ≤152	>152 ≤191	>191
Minimum wall thickness, [mm]	1.75	2.0	2.2	2.4	2.6	3.0	3.25	3.5	4.0	5.0	5.4

Note:
The decrease in wall thickness due to expanding or bending shall be compensated by allowances.

8.2.8.5 The minimum wall thickness of pipes made of non-ferrous alloys and stainless steel may be less than those specified in paragraph 8.2.8.4, however not less than those determined in accordance with 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 in this sub-chapter apply to cylindrical walls with:

$$\frac{D_a}{D} \le 1.2$$

Wall thickness of pipes with $D_a \le 200$ mm in diameter shall be determined in accordance with paragraph 8.2.8.2.

8.2.9.2 Plain wall thickness of cylindrical elements, with or without stiffeners including plain furnaces of boilers shall not be less than that determined in accordance with the formula below:

$$s = \frac{50\left(B + \sqrt{B^2 + 0.04AC}\right)}{A} + c, \quad \text{[mm]} \qquad (8.2.9.2\text{-}1)$$

where:

$$A = 200 \cdot \frac{\sigma}{D_m}\left(1 + \frac{D_m}{10 \cdot l}\right)\left(1 + \frac{5D_m}{l}\right), \qquad (8.2.9.2\text{-}2)$$

$$B = p\left(1 + \frac{5D_m}{l}\right), \qquad (8.2.9.2\text{-}3)$$

$$C = 0.045 \cdot p \cdot D_m \qquad (8.2.9.2\text{-}4)$$

s – wall thickness, [mm];
p – design pressure (see sub-chapter 8.2.2), [MPa];
D_m – mean diameter, [mm];
σ – allowable stress (see paragraph 8.2.4.5), [MPa];
c – design thickness allowance (see sub-chapter 8.2.7), [mm];
l – design length of cylindrical portion between stiffeners, [mm].

End plates, furnace connections to end plates and combustion chamber as well as stiffening rings (Fig. 8.2.9.2) or similar structures may be considered as stiffeners.

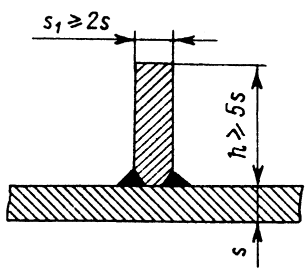


Fig. 8.2.9.2

8.2.9.3 Corrugated furnaces shall have a thickness not less than that determined in accordance with the formula below:

$$s = \frac{p \cdot D}{2 \cdot \sigma} + c \qquad (8.2.9.3)$$

s – wall thickness, [mm];
D – minimum inner diameter of the corrugated portion of furnace, [mm];
p – design pressure (see sub-chapter 8.2.2), [MPa];
σ – allowable stress (see paragraph 8.2.4.5), [MPa];
c – design thickness allowance (see sub-chapter 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 not be less than that calculated in accordance with formula 8.2.9.2-1.

8.2.9.5 Thickness of plain furnaces shall not be less than 7 mm however not more than 20 mm. The thickness of corrugated furnaces shall not be less than 10 mm however not more than 20 mm.

8.2.9.6 Plain furnaces up to 1400 mm in length need not be fitted with stiffening rings. Where a boiler has two or more furnaces, the stiffening rings of adjacent furnaces shall be arranged in alternate planes.

8.2.9.7 Holes and openings in cylindrical and spherical walls shall be compensated for in accordance with the requirements specified in sub-chapter 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), shall not be less than that determined in accordance with the formula below:

$$s_1 = \frac{3.7}{\sigma}\sqrt{pD_1(D_1 - D_0)} + 1, \quad \text{[mm]} \qquad (8.2.9.8)$$

p– design pressure, [MPa].

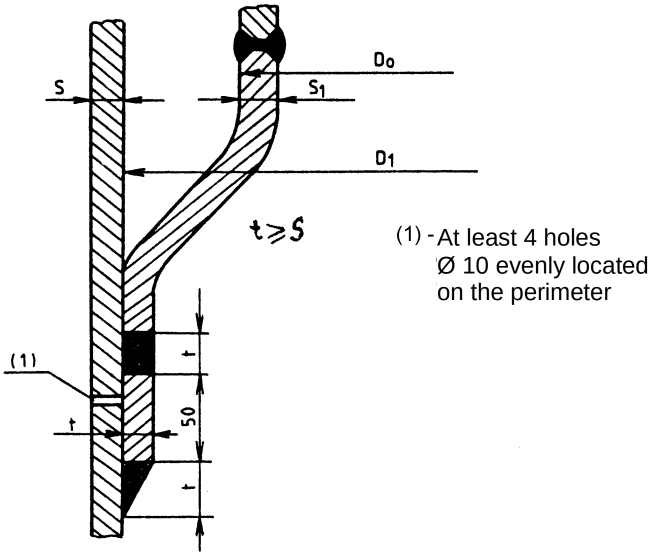


Fig. 8.2.9.8

8.2.10 Conical Elements

8.2.10.1 Wall thickness of conical elements subjected to internal pressure shall not be less than:

.1 at α ≤ 70° – the greater value out of those determined in accordance with the formulae below:

$$s = \frac{D_a py}{4\sigma\varphi} + c\,, \quad [\text{mm}] \qquad (8.2.10.1.1\text{-}1)$$

and

$$s = \frac{D_a py}{(4\sigma\varphi - p)\cos\alpha} + c\,, \quad [\text{mm}] \qquad (8.2.10.1.1\text{-}2)$$

.2 at α > 70° – the value determined in accordance with the formula below:

$$s = 0.3\left[D_a - (r+s)\right]\sqrt{\frac{p}{\sigma\varphi}} \cdot \frac{\alpha}{90^\circ} + c\,, \quad [\text{mm}] \qquad (8.2.10.1.2)$$

s – wall thickness, [mm];
D_c – design diameter (Figures 8.2.10.1.2-1 to 8.2.10.1.2-4), [mm];
D_a – outside diameter (Figures 8.2.10.1.2-1 to 8.2.10.1.2-4), [mm];
p – design pressure (see sub-chapter 8.2.2), [MPa];
y – shape factor (see Table 8.2.10.1);
α, α_1, α_2, α_3 – angles (Figures 8.2.10.1.2-1 ÷ 8.2.10.1.2-4), [°];
σ – allowable stress (see paragraph 8.2.4.5), [MPa];

φ – strength factor (see sub-chapter 8.2.6). In formulae 8.2.10.1.1-1 and 8.2.10.1.2, the strength factor for circumferential weld seam shall be applied, whereas in formula 8.2.10.1.1-2 – for longitudinal weld seam. For seamless conical shell segments, and also where circumferential seam is at the distance from the edge exceeding:

$$0.5\sqrt{\frac{D_a \cdot s}{\cos\alpha}}$$

strength factor $\varphi = 1$ shall be taken;

r – edge radius (Figures 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 sub-chapter 8.2.7), [mm];

Table 8.2.10.1

α, [degs]	Shape factor y as function of r / D_a ratio											
	0.01	0.02	0.03	0.04	0.06	0.08	0.10	0.15	0.20	0.30	0.40	0.50
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 welded joints (see Fig. 8.2.10.1.2-3), shape factor y shall be determined for $r / D_a = 0.01$.

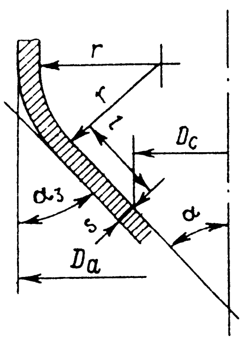


Fig. 8.2.10.1.2-1

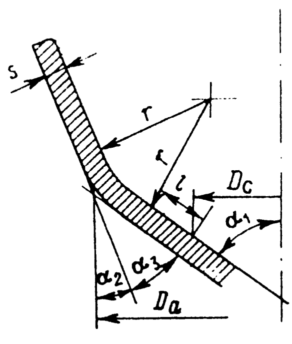


Fig. 8.2.10.1.2-2

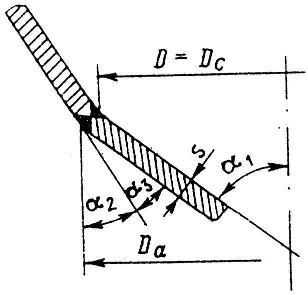


Fig. 8.2.10.1.2-3

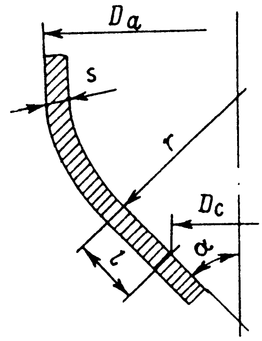


Fig. 8.2.10.1.2-4

l – distance from the edge of the wide end of conical shell, along the generatrix, taken as tenfold wall thickness, however not greater than half the length of the conical shell generatrix segment (Figures 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 in accordance with paragraph 8.2.10.1, provided the following conditions are fulfilled:

.1 strength factor of welded joint $\varphi = 1$ shall be taken;

.2 allowance c shall be taken equal to 2 mm;

.3 design diameter D_c shall be determined in accordance with the formula below:

$$D_c = \frac{d_1 + d_2}{2\cos\alpha}, \quad \text{[mm]} \qquad (8.2.10.2.3)$$

d_1 , d_2 – the largest and the smallest diameter of the cone, respectively, [mm];

.4 for $\alpha < 45°$ it shall be demostrated that the walls are not subject to plastic strain. Pressure p_1, at which plastic strain occurs, shall be determined in accordance with the formula below:

$$p_1 = 26E10^{-6}\frac{D_c}{l_1}\left[\frac{100(s-c)}{D_c}\right]^2\sqrt{\frac{100(s-c)}{D_c}}, \quad \text{[MPa]} \qquad (8.2.10.2.4)$$

E – modulus of elasticity, [MPa];

l_1 – the maximum length of the cone or distance between its supports, [mm].

Fulfilment of inequality $p_1 > p$ (p – design pressure, [MPa]) is the condition of absence of plastic strain of the cone walls.

8.2.10.3 Welded joints (see Fig. 8.2.10.1.2-3) are permitted only with the values of angle $\alpha_3 \le 30°$ and wall thickness $s \le 20$ mm. The joints shall be double-side welded. In conical shell segments with $\alpha \geq 70°$, welded joints may be made without edge bevelling provided that the requirements specified in paragraph 8.2.10.2 are fulfilled.

Such joints are not recommended in boilers.

8.2.10.4 In way of holes and openings in conical walls, adequate strengthening shall be provided in accordance with the requirements specified in sub-chapter 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 (Figures 8.2.11.1-1 ÷ 8.2.11.1-8 and Fig. 1.2 in the Annex) shall not be less than that determined in accordance with the formula below:

$$s = KD_c\sqrt{\frac{p}{\sigma}} + c\,, \quad \text{[mm]} \qquad (8.2.11.1\text{-}1)$$

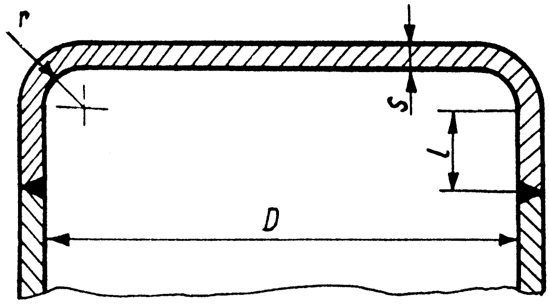
s – wall thickness, [mm];
K – design factor for the design patterns shown in Figures 8.2.11.1-1 to 8.2.11.1-8 and items 1.1 to 1.6 in the Annex), [mm];
D_c – design diameter (Figures 8.2.11.1-2 to 8.2.11.1-7 and item 1.2 in the Annex), [mm]. For such end plates as shown in Fig. 8.2.11.1-1 and Fig. 1.1 in the Annex, the design diameter shall be:

$$D_c = D - r\,, \ \text{[mm]} \qquad (8.2.11.1\text{-}2)$$

For rectangular or oval covers, the design diameter shall be determined in accordance with the formula below:

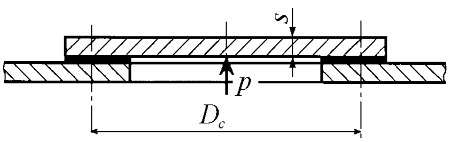
$$D_c = m\sqrt{\frac{2}{1+\left(\frac{m}{n}\right)^2}}\,, \quad \text{[mm]} \qquad (8.2.11.1\text{-}3)$$

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 opening respectively, measured to the axis of the packing arrangement (Fig. 8.2.11.1-8), [mm];
r – inner curvature radius of the dished end plate, [mm];
p – design pressure (see sub-chapter 8.2.2), [MPa];
σ – allowable stress (see paragraph 8.2.4.5), [MPa];
c – design thickness allowance (see sub-chapter 8.2.7), [mm];
l – length of cylindrical portion of end plate (Fig. 8.2.11.1-1 and Fig. 1.1 in the Annex), [mm].



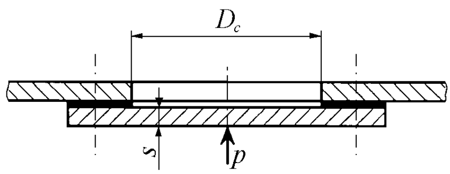
K = 0.30

Fig. 8.2.11.1-1



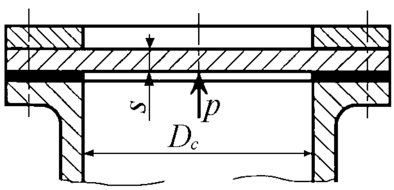
K = 0.41

Fig. 8.2.11.1-2



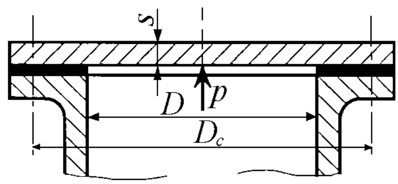
K = 0.45

Fig. 8.2.11.1-3



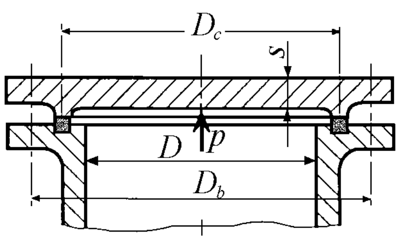
K = 0.41

Fig. 8.2.11.1-4



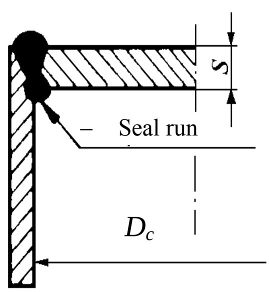
K = 0.35

Rys. 8.2.11.1-5



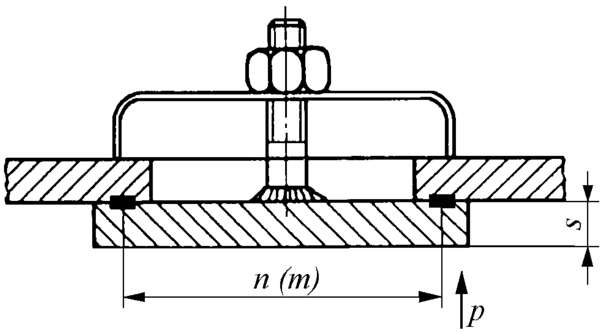
D_b/D	K
1.25	0.6
1.50	0.7
1.75	0.8

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 Thickness of the plates shown in item 1.2 of the Annex shall not be less than that determined in accordance with formula 8.2.11.1-1. Additionally, the following conditions shall be fulfilled:

.1 For circular end plates

$$0.77s_1 \geq s_2 \geq \frac{1.3p}{\sigma}\left(\frac{D_c}{2} - r\right); \qquad (8.2.11.2.1)$$

.2 For rectangular end plates

$$0.55s_1 \geq s_2 \geq \frac{1.3p}{\sigma}\frac{nm}{(n+m)}; \qquad (8.2.11.2.2)$$

s – end plate thickness, [mm];
s_1 – shell thickness, [mm];
s_2 – end plate thickness within the relieving groove, [mm].
For explanation of other symbols – see sub-chapter 8.2.11.
Thickness s_2 shall never 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 are subject to PRS acceptance in each particular case.

8.2.12 Flat Walls Strengthened by Stays

8.2.12.1 Flat walls (Figures 8.2.12.1-2 and 8.2.12.1-3) strengthened by long and short stays, corner stays, stay tubes or other similar structures shall have a thickness not less than that determined in accordance with the formula below:

$$s = KD_c\sqrt{\frac{p}{\sigma}} + c \qquad (8.2.12.1\text{-}1)$$

K – design factor (see Figures 8.2.12.1-1 ÷ 8.2.12.1-3 and 5.1 ÷ 5.3 in the Annex); if the part of the wall area in question is reinforced by stays having variable values of K factor, the formula shall be used with K value equal to the arithmetic mean of these factors.

D_c – calculation diameter (Figs. 8.2.12.1-2 and 8.2.12.1-3), [mm];
With even arrangement of stays:

$$D_c = \sqrt{a_1^2 + a_2^2} \qquad (8.2.12.1\text{-}2)$$

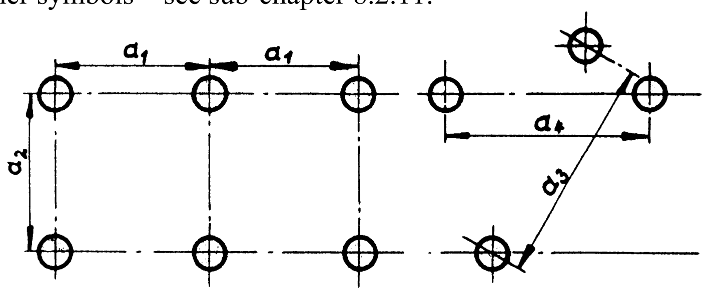
with uneven arrangement of stays:

$$D_c = \frac{a_3 + a_4}{2} \qquad (8.2.12.1\text{-}3)$$

In all other instances, 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 specified in sub-chapter 8.2.13; in this case, the flanging shall be regarded as a point of support. A manhole flanging shall not be regarded as a point of support;

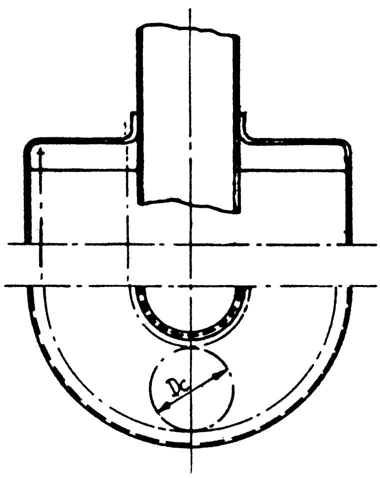
a_1, a_2, a_3, a_4 – pitch or stay-to-stay distance (Fig. 8.2.12.1-1), [mm].

For other symbols – see sub-chapter 8.2.11.



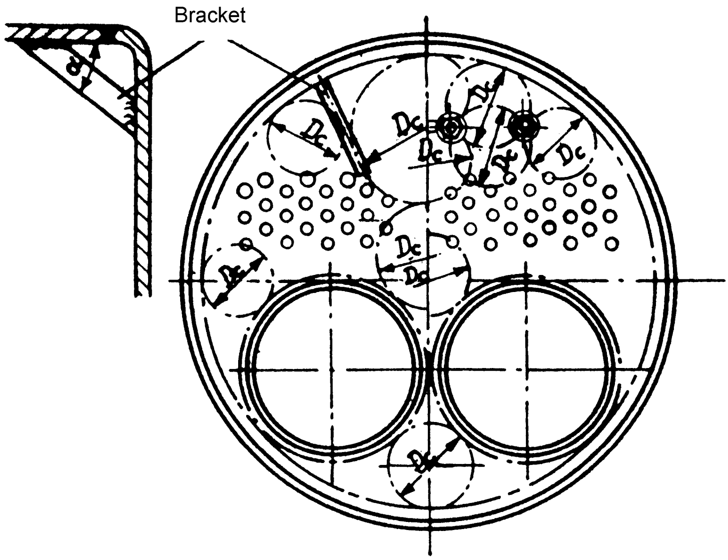
$K = 0.45$

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 Flanging Flat Walls

8.2.13.1 In flat wall and end plate calculations, the flanging can be taken into account when the inner flanging radius is not less than that specified in Table 8.2.13.1.

Table 8.2.13.1

End plate outer diameter [mm]	Flanging radius [mm]
up to 350	25
from 350 to 500	30
from 500 to 950	35
from 950 to 1400	40
from 1400 to 1900	45
over 1900	50

The inner flanging radius shall not be less than 1.3 times the wall thickness.

8.2.13.2 The length of cylindrical portion of a flanged flat end plate shall not be less than determined in accordance with the following formula: $l = 0.5\sqrt{Ds}$, (see Fig. 8.2.11.1-1).

8.2.14 Strengthening of Openings in Flat Walls

8.2.14.1 In flat walls, end plates and covers, openings with diameters greater than four times the thickness shall be strengthened by means of welded-on branch pieces or pads, or by increasing the design wall thickness. The openings shall be arranged at a distance not less than 0.125 times the design diameter from the design diameter outline.

8.2.14.2 If the actual wall thickness is greater than that determined in accordance with formulae 8.2.11.1-1 and 8.2.12.1-1, the maximum diameter of a not strengthened opening shall be determined in accordance with the formula below:

$$d = 8s_r\left(1.5\frac{s_r^2}{s^2} - 1\right) \qquad (8.2.14.2)$$

d – diameter of not strengthened opening, [mm];
s_r – actual wall thickness, [mm];
s – determined in accordance with formulae 8.2.11.1-1 and 8.2.12.1-1, [mm].

8.2.14.3 Edge reinforcement shall be provided for openings of larger diameters than those specified in paragraphs 8.2.14.1 and 8.2.14.2.

The dimensions of reinforcing elements of branches shall fulfil the following condition:

$$s_k\left(\frac{h^2}{s_r^2} - 0.65\right) \geq 0.65d - 1.4s_r \qquad (8.2.14.3)$$

s_k – branch piece wall thickness, [mm], (see Fig. 8.2.14.3), [mm];
d – branch piece inside diameter, [mm];
s_r – see paragraph 8.2.14.2, [mm];
$h = h_1 + h_2$, [mm], (see Fig. 8.2.14.3).

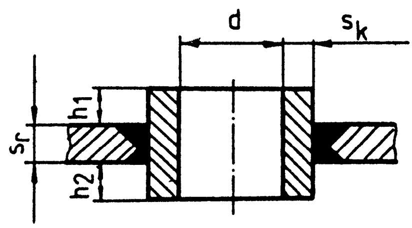


Fig. 8.2.14.3

8.2.15 Tube Plates

8.2.15.1 Thickness s_1 of flat tube plates of heat exchangers shall not be less than that determined in accordance with the formula below:

$$s_1 = 0.9 K D_W \sqrt{\frac{P}{\sigma\varphi}} + c\,, \quad \text{[mm]} \qquad (8.2.15.1)$$

K – factor depending on the ratio of shell wall thickness s to tube plate thickness s_1; for tube plates welded to the shell, K shall be determined in accordance with diagram 8.2.15.1 on the preliminary assumption of s_1 thickness, and the calculation shall be corrected if the difference between assumed value of s_1 and that determined in accordance with formula 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 – shell inner diameter, [mm];

P – design pressure (see sub-chapter 8.2.2), [MPa];

σ – allowable stress (see paragraph 8.2.4.5), [MPa];
for heat exchangers of rigid structure where the thermal elongation factors of shell and pipe materials are different, σ shall be reduced by 10%;

φ – strength factor of tube plate weakened by holes for pipes (see paragraph 8.2.15.2);

c – design thickness allowance (see sub-chapter 8.2.7), [mm].

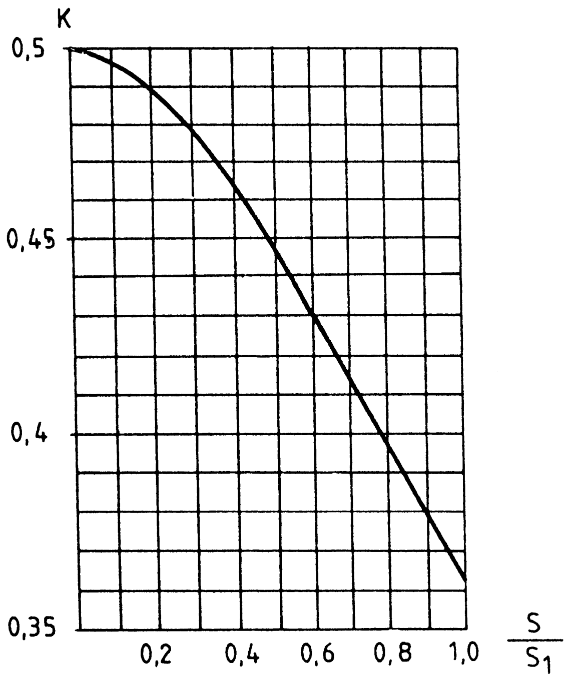


Fig. 8.2.15.1

8.2.15.2 Where $0.75 > d / a > 0.4$ and $D_W / s_1 \geq 40$, the strength factor of a tube plate shall be calculated in accordance with the following formulae:

where holes are arranged in an equilateral triangle pattern:

$$\varphi = 0.935 - 0.65\frac{d}{a} \qquad (8.2.15.2\text{-}1)$$

where holes are arranged in a row or in transposition:

$$\varphi = 0.975 - 0.68\frac{d}{a_2} \qquad (8.2.15.2\text{-}2)$$

d – diameter of tube plate holes, [mm];
a – spacing of hole-axes arranged in triangle pattern, [mm];
a_2 – spacing of hole-axes arranged in a row or in transposition (as well as arranged concentrically), whichever is lesser, [mm].

8.2.15.3 For quotients $d / a = 0.75 \div 0.80$, the tube plate thickness determined in accordance with formula 8.2.15.1 shall fulfil the condition below:

$$f_{\min} \geq 5d$$

$f_{\min}$ – minimum allowable cross sectional area of bridge in tube plate, [mm^2].

For values of $\frac{d}{a}$ and $\frac{D_w}{s_1}$ other than those specified above, as well as for heat exchangers with rigid structure when the difference in mean temperatures exceeds 50 °C, the thickness of tube plates is subject to PRS acceptance in each particular case.

8.2.15.4 In addition to the requirement specified in paragraph 8.2.14.1, the thickness of tube plates with expanded tubes shall fulfil the condition below:

$$s \geq 10 + 0.125\, d \qquad (8.2.15.4)$$

Expanded connections of tubes to tube plates shall also fulfil the requirements specified in paragraphs 8.2.20.6, 8.2.20.7 and 8.2.20.8.

8.2.15.5 If tube plates are stregthened by welded or expanded pipes in accordance with the requirements specified in sub-chapter 8.2.20, then the calculations of such tubes may be performed in accordance with the requirements specified in sub-chapter 8.2.12.

8.2.16 Dished Ends

8.2.16.1 Thickness of dished ends, whether unpierced or pierced, subjected to internal or external pressure (see Fig. 8.2.16.1) shall not be less than that determined in accordance with the formula below:

$$s = \frac{D_a p y}{4\sigma\varphi} + c \qquad (8.2.16.1)$$

s – end wall thickness, [mm];

p – design pressure, [MPa];

D_a – end outer diameter, [mm].
The end shall be flanged within the distance not less than 0.1 D_a measured from the outer edge of the end cylindrical portion (see Fig. 8.2.16.1);

φ – strength factor (see sub-chapter 8.2.6);

σ – allowable stress (see paragraph 8.2.4.5), [MPa];

y – shape factor determined in accordance with Table 8.2.16.1, depending on the ratio of the height to outside diameter of the end and on the value of weakening by holes; for intermediate values of $\frac{h_a}{D}$ and $\frac{d}{\sqrt{D_a s}}$, shape factor y may be determined by linear interpolation.

To determine y in accordance with Table 8.2.16.1, the preliminary value s shall be taken from the standardized thickness series. The final value of s shall not be less than that determined in accordance with 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	Ratio $\frac{h_a}{D_a}$	Shape factor							
		y – for flanged area and un-pierced ends	y_A – for dished part of end with not strengthened holes with respect to $\frac{d}{\sqrt{D_a s}}$						y_c – for dished part of end with strengthened holes
			0.5	1.0	2.0	3.0	4.0	5.0	
Dished elliptical or basket shaped ends with $R_W = D_a$	0.20	2.9	2.9	2.9	3.7	4.6	5.5	6.5	2.4
Dished elliptical or basket shaped ends with $R_W = 0.8\ D_a$	0.25	2.0	2.0	2.3	3.2	4.1	5.0	5.9	1.8
Dished spherical ends with $R_W = 0.5\ D_a$	0.50	1.1	1.2	1.6	2.2	3.0	3.7	4.35	1.1

c – design thickness allowance, to be taken equal to:
2 mm – if subjected to internal pressure;
3 mm – if subjected to external pressure;
for wall thickness exceeding 30 mm, the above values of allowance may be reduced by 1 mm.

d – the largest diameter of not strengthened hole, [mm].

Formula 8.2.16.1 is applicable if the following conditions are fulfilled:

$$\frac{h_a}{D_a} \geq 0.18\ ;\quad \frac{s-c}{D_a} \geq 0.0025;\quad R_W \leq D_a\ ;\quad r \geq 0.1D_a\ ;\quad l \leq 150 \text{ mm},$$

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 dimensions of dished end elements are shown in Fig. 8.2.16.1.

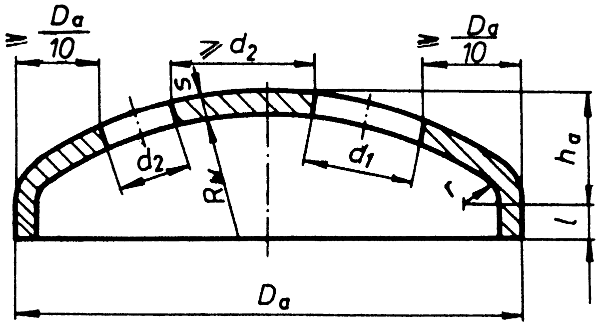


Fig. 8.2.16.1

8.2.16.2 Unpierced ends as well as ends with holes whose diameter is not greater than 4s and not greater than 100 mm arranged at a distance not less than $0.2D_a$ from the outer cylindrical portion of the end are also considered as unpierced ends. Not strengthened holes with the diameter less than the end thickness, however not exceeding 25 mm, are permitted in way of the end curvature.

8.2.16.3 Wall thickness of dished ends in combustion chambers of vertical boilers may also be calculated as for unpierced ends 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 materials the modulus of elasticity value is subject to PRS acceptance in each particular case;

R_W – maximum inner radius of curvature, [mm].

For other symbols – see paragraph 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 made of non-ferrous alloys, the minimum wall thickness may be reduced subject to PRS acceptance in each particular case.

8.2.16.6 Application of dished ends of welded construction is subject to PRS acceptance in each particular case.

8.2.17 Flanged End Plates

Thickness of unpierced flanged end plates (see Fig. 8.2.17) subjected to internal pressure shall not be less than that determined in accordance with the formula below:

$$s = \frac{3Dp}{\sigma} + c \qquad (8.2.17)$$

s – wall thickness, [mm];
p – design pressure (see sub-chapter 8.2.2), [MPa];
D – inside diameter of end plate, taken equal to shell internal diameter, [mm];
σ – allowable stress (see paragraph 8.2.4.5), [MPa];
c – design thickness allowance (see sub-chapter 8.2.7), [mm].

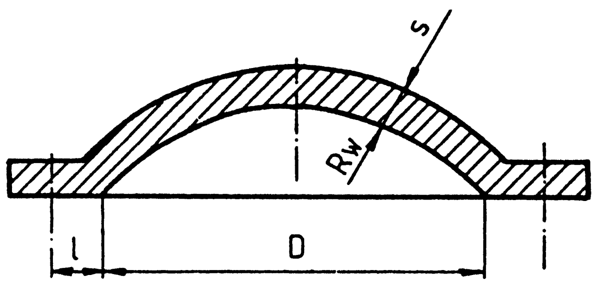


Fig. 8.2.17

Flanged end plates are allowed within a range of diameters D up to 500 mm and for working pressures not higher than 1.5 MPa. The end plate curvature radius R_W shall not be less than 1.2 D, and the distance l shall not exceed $2s$.

8.2.18 Headers of Rectangular Section

8.2.18.1 The wall thickness of rectangular headers (Fig. 8.2.18.1-1) subjected to the internal pressure shall not be less than that determined in accordance with the formula below:

$$s = \frac{pn}{2.52\sigma\varphi_1} + \sqrt{\frac{4.5Kp}{1.26\sigma\varphi_2}} \qquad (8.2.18.1\text{-}1)$$

s – wall thickness, [mm],
p – design pressure (see sub-chapter 8.2.2), [MPa],
n – half of the width of the header side normal to that being calculated, [mm],
m – half of the width of the header side being calculated, [mm],
σ – allowable stress (see paragraph 8.2.4.5), [MPa],
φ_1 and φ_2 – strength factors of headers, weakened by holes, determined as follows:
φ_1 – in accordance with formula 8.2.6.2.1,
φ_2 – in accordance with formula 8.2.6.2.1, if $d < 0.6$ m,

$$\varphi_2 = 1 - \frac{0.6m}{\alpha}, \text{ if } d \geq 0.6 \text{ m} \qquad (8.2.18.1\text{-}2)$$

d – diameter of holes, [mm]. For oval holes, d shall be taken as equal to the size of holes at the longitudinal axis, however, in formulae 8.2.6.2.1 and 8.2.18.1-2 the size at the axis perpendicular to the header centre line shall be taken as d for oval holes.

Where the holes are arranged in staggered pattern, a_2 (see Fig. 8.2.18.1-2) shall be substituted for a in formula 8.2.18.1-2. Where the rectangular headers have longitudinal welds (see Fig. 8.2.18.1-1), strength factors, φ_1 and φ_2 shall be taken as equal, respectively, to the joint factors of weld seams selected as required in sub-chapter 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 locations, the lowest value of strength factor shall be taken for calculations.

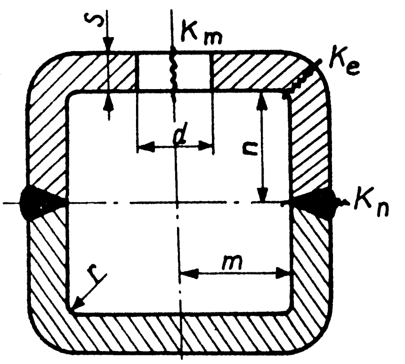


Fig. 8.2.18.1-1

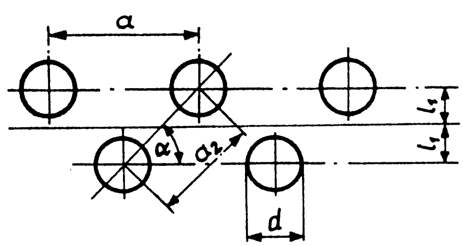


Fig. 8.2.18.1-2

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 formula:
for the centre line of the header 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 welds

$$K_n = \frac{m^3 + n^3}{3(m+n)} - \frac{m^2 - l_1^2}{2}, \quad [\text{mm}^2] \qquad (8.2.18.1\text{-}4)$$

If the above formulae give negative results, then their absolute 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 If fillet welds are accepted by PRS in rectangular headers, then the wall thickness of such headers shall not be less than that determined in accordance with the formula below:

$$s = \frac{p\sqrt{m^2 + n^2}}{2.52\sigma\varphi_1} + \sqrt{\frac{4.5K_e p}{1.26\sigma\varphi_2}} \qquad (8.2.18.2\text{-}1)$$

K_e – design factor for bending moment at the edges, [mm^2], determined in accordance with the formula below:

$$K_e = \frac{m^3 + n^3}{3(m+n)} \qquad (8.2.18.2\text{-}2)$$

For other symbols used – see paragraph 8.2.18.1.

8.2.18.3 Fillet radius of rectangular header side edges shall not be less than 0.33 times the wall thickness, however not less than 8 mm. The minimum thickness of header wall with expanded tubes shall not be less than 14 mm. The width of bridges between the holes shall not be less than 0.25 times the spacing of the hole centres. The wall thickness in way of the fillets shall not be less than that determined in accordance with formulae 8.2.18.1-1 and 8.2.18.2-1.

8.2.19 Openings in Cylindrical, Spherical, Conical Walls and in Dished Ends

8.2.19.1 Strengthening arrangements shall be provided in way of openings. The following strengthening methods are permitted:

.1 wall thickness increased above the design thickness (Figs. 8.2.19.1-1 and 8.2.19.1-2);

.2 disk-shaped strengthening plates welded on the wall being strengthened (Figs. 8.2.19.1-3 and 8.2.19.1-4);

.3 welded-on pipe elements, such as branch pieces, sleeves etc. (Figures 8.2.19.1-5 ÷ 8.2.19.1-7).

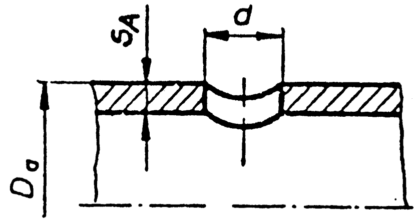


Fig. 8.2.19.1-1

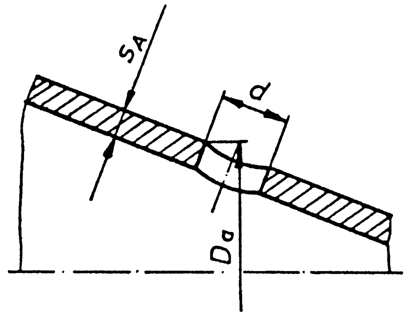


Fig. 8.2.19.1-2

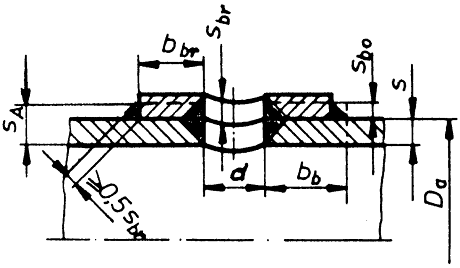


Fig. 8.2.19.1-3

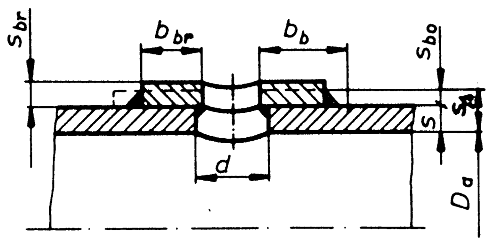


Fig. 8.2.19.1-4

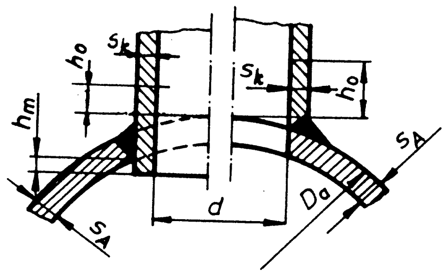


Fig. 8.2.19.1-5

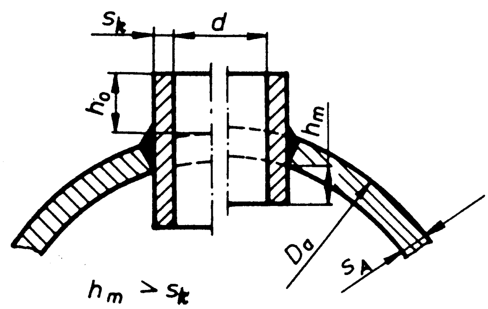


Fig. 8.2.19.1-6

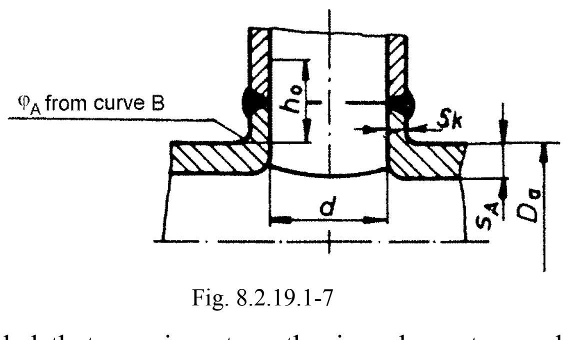


Fig. 8.2.19.1-7

It is recommended that opening strengthening elements, as shown in Figures 8.2.19.1-5 ÷ 8.2.19.1-7, be welded with temporary backing or using other techniques ensuring proper penetration of the welded joint.

8.2.19.2 Thickness of pierced walls shall fulfil the requirements specified in sub-chapters 8.2.8 and 8.2.9 for cylindrical walls, in sub-chapter 8.2.10 – for conical walls and in sub-chapter 8.2.16 – for dished ends.

8.2.19.3 Materials used for the walls being strengthened and for strengthening elements shall have identical strength characteristics, if possible. Where the materials of strengthening elements have worse strength characteristics than the wall material, the cross-sectional area strengthening elements shall be increased accordingly.

Strengthening elements shall be properly connected to the wall being strengthened.

8.2.19.4 Openings in walls shall be located at a distance equal at least triple wall thickness, however not less than 50 mm from the welded joints. The arrangement of openings at the distance less than 50 mm from the welded joints is subject to PRS acceptance in each particular case.

8.2.19.5 Opening diameter (or the largest dimension of an opening other than circular) shall not exceed 500 mm. Application of openings greater than 500 mm and their strengthening methods are subject to PRS acceptance in each particular case.

8.2.19.6 In general, wall thickness of tubular elements (branch pieces, sleeves or nozzles) welded to the walls of boilers, pressure vessels and heat exchangers shall not be less than 5 mm. Application of elements less than 5 mm in thickness is subject to PRS acceptance in each particular case.

8.2.19.7 Opening may be strengthened by increasing design thickness of the wall. In that case, increased wall thickness s_A shall not be less than the value determined in accordance with the following formulae:

for cylindrical shells

$$s_A = \frac{pD_a}{2\sigma\varphi_A + p} + c\,, \qquad (8.2.19.7\text{-}1)$$

for spherical shells

$$s_A = \frac{pD_a}{4\sigma\varphi_A + p} + c \qquad (8.2.19.7\text{-}2)$$

for conical shells

$$s_A = \frac{pD_a}{(2\sigma\varphi_A - p)\cos\alpha} + c \qquad (8.2.19.7\text{-}3)$$

s_A – required wall thickness without compensating elements, [mm];

φ_A – strength factor of wall weakened by opening which is being strengthened, determined for the pattern curve A (see diagram in Fig. 8.2.19.7) depending on dimensionless parameter $\frac{d}{\sqrt{D_a(s_A - c)}}$, and to determine this parameter, the value of s_A obtained in accordance with formulae 8.2.19.7-1 to 8.2.19.7-3 shall be taken;

d – diameter of the opening (inner diameter of a branch piece, sleeve) or the dimension of an oval or elliptical opening along the longitudinal axis, [mm].

For other symbols – see paragraphs 8.2.8.2 and 8.2.10.1

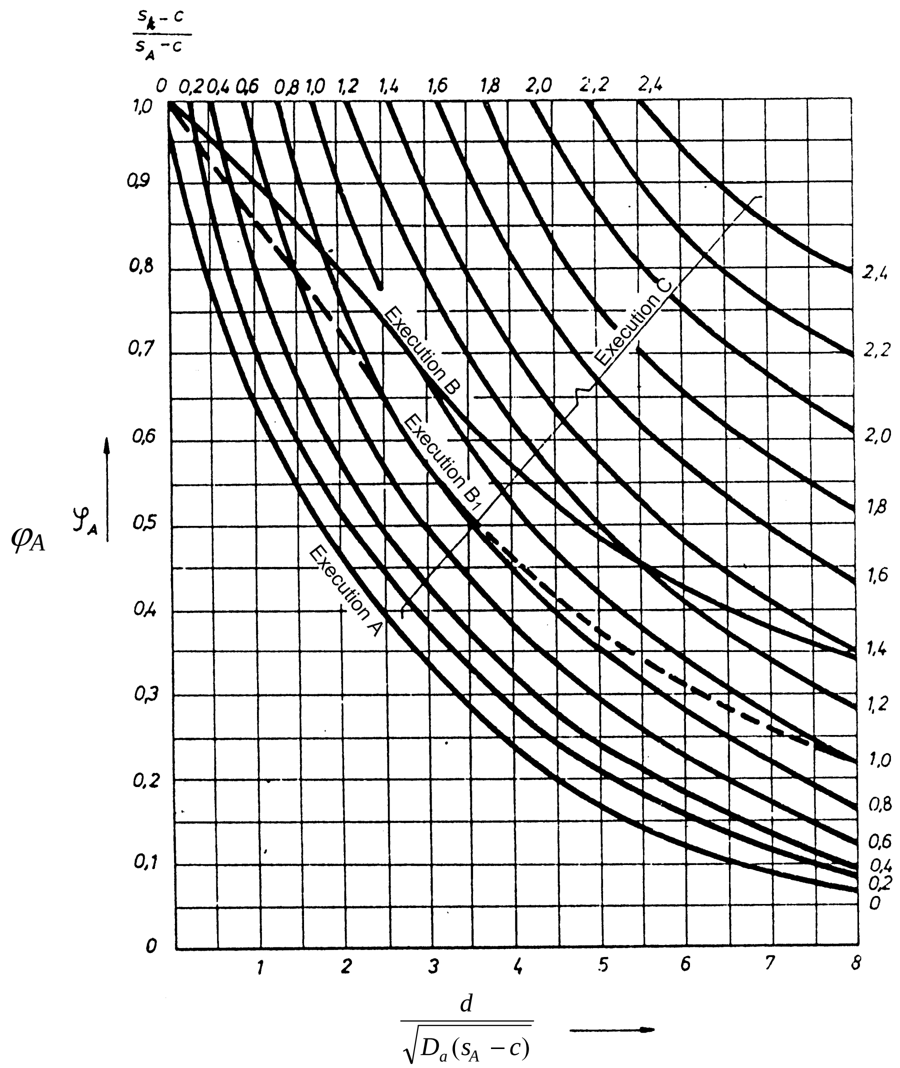


Fig. 8.2.19.7

8.2.19.8 Where disk-shaped plates are used to strengthen openings in cylindrical, spherical or conical walls, the dimensions of the strengthening plates shall be determined in accordance with the following formulae:

$$b_b = \sqrt{D_a(s_A - c)}\,, \tag{8.2.19.8-1}$$

$$s_{b0} \geq s_A - s_r\ , \tag{8.2.19.8-2}$$

b_b – maximum effective width of the plate (see Figures 8.2.19.1-3 and 8.2.19.1-4), [mm];
s_{b0} – plate thickness (see Figures 8.2.19.1-3 and 8.2.19.1-4), [mm];
s_A – total thickness of wall being strengthened and strengthening plate, determined in accordance with the requirements specified in paragraph 8.2.19.7, [mm];
s_r – actual thickness of wall being strengthened, [mm].
For other symbols – see paragraph 8.2.19.7.

Where the actual width of strengthening plate is less than that resulting from formula 8.2.19.8-1, the plate thickness shall be increased respectively, in accordance with the formula below:

$$s_{br} \geq s_{b0}\frac{1+\dfrac{b_b}{b_{br}}}{2}\,, \tag{8.2.19.8-3}$$

s_{br} – actual thickness of plate, [mm];
b_{br} – actual width of the plate, [mm].

Thickness of weld seam connecting the strengthening plate to the wall shall not be less than 0.5 s_{br} (Fig. 8.2.19.1-3).

8.2.19.9 Dimensions of welded tubular elements used to strengthen openings in cylindrical, spherical and conical walls shall not be less than those determined as follows:

.1 Wall thickness s_k of a tubular element (branch piece, sleeve, etc.), [mm], shall be determined as a function of the following dimensionless parameter

$$\frac{d}{\sqrt{D_a(s_A - c)}}$$

and the strength factor φ_A, from curves C shown in Fig. 8.2.19.7. Values φ_r and s_r shall be substituted for φ_A and s_A shown in Fig. 8.2.19.7, where.

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.

Ratio $\dfrac{s_k - c}{s_A - c}$, determined from the diagram in Fig. 8.2.19.7 shall be used to determine the minimum thickness s_k , [mm] of a branch piece or sleeve. In this ratio, actual thickness s_r shall be substituted for s_A.

.2 The minimum design height h_0 [mm] of a tubular strengthening element shall be determined in accordance with the formula below:

$$h_0 = \sqrt{d(s_k - c)}\,, \qquad (8.2.19.9.2\text{-}1)$$

If the actual thickness, h_r, of a tubular strengthening element is less than that determined in accordance with formula 8.2.19.9.2-1, thickness s_k shall be increased respectively as follows:

$$s_{kr} = s_k \frac{h_0}{h_r}\,, \qquad (8.2.19.9.2\text{-}2)$$

8.2.19.10 Openings in dished ends shall be compensated for as follows:

.1 For openings stregthened by increasing the dished end wall thickness, factor y_A obtained from Table 8.2.16.1 shall be substituted for factor y in formula 8.2.16.1.

.2 For openings stregthened by means of disk-shaped stregthening plates, the plate dimensions shall be determined as required in paragraph 8.2.19.8, and the total thickness of the stregthened end wall, s_A, shall be determined in accordance with the formula below:

$$s_A = \frac{p(R_W + s)y_0}{2\sigma\varphi_A} + c \qquad (8.2.19.10.2)$$

R_W – inner radius of curvature in the way of the opening, [mm];
y_0 – shape factor determined in accordance with Table 8.2.16.1;
For other symbols – see paragraphs 8.2.16.1 and 8.2.19.7.

.3 The dimensions of tubular elements strengthening openings shall be determined in accordance with paragraph 8.2.19.9, except that the expression $2(0.5D_a + s)$ shall be substituted for D_a in the following dimensionless parameter

$$\frac{d}{\sqrt{D_a\,(s-c)}}$$

and the actual strength factor φ for the dished end wall thickness, s, shall be determined in accordance with formula 8.2.16.1, assuming $\varphi = \varphi_A$, $y = y_0$ and $s = s_A$ (see paragraph 8.2.16.1).

8.2.19.11 For through tubular strengthening elements with the inward projecting portion $h_m \geq s_r$ (Figures 8.2.19.1-5 and 8.2.19.1-6), thickness of the tubular element may be reduced by 20%, however its thickness shall not be less than that required for the design pressure.

8.2.19.12 The ratio of a tubular strengthening element thickness, s_k, to the thickness of wall being strengthened, s, shall not be greater than 2.4. If this ratio is taken as more than 2.4, for construction reasons, then tubular strengthening element thickness s_k shall be assumed not greater than 2.4 times the thickness of the wall being strengthened in the calculation.

8.2.19.13 Disk-shaped strengthening plates and tubular strengthening elements may also be used in combination (Fig. 8.2.19.13). In that case, the dimensions of strengthening elements shall be determined taking account of the requirements for both the disk-shaped and tubular strengthening elements.

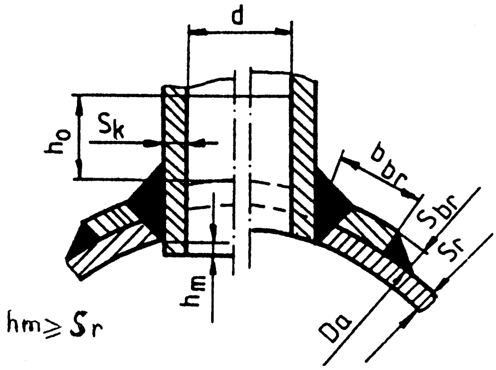


Fig. 8.2.19.13

8.2.19.14 For branch pieces drawn from the wall being strengthened (Fig. 8.2.19.1-7), wall thickness s_A shall not be less than that determined in accordance with formulae 8.2.19.7-1 to 8.2.19.10-2.

Strength factor φ_A for the wall weakened due to a drawn branch piece shall be obtained from diagram 8.2.19.7 as follows:

for $\frac{d}{D_a} \le 0.4$ – from curve B,

for $\frac{d}{D_a} = 1.0$ – from curve B_1,

for $0.4 < \frac{d}{D_a} < 1.0$ – by interpolation of curves B and B_1.

For curves B and B1 – see Fig. 8.2.19.7 diagram.

Thickness of a drawn branch shoulder s_k shall not be less than that determined in accordance with the formula below:

$$s_k \ge s_A \frac{d}{D_a}, \quad \text{[mm]} \tag{8.2.19.14}$$

however not less than that required for the design pressure.

8.2.19.15 The effect of adjacent openings may be disregarded provided that:

$$\left(l + s_{kr1} + s_{kr2}\right) \ge 2\sqrt{D_a\left(s_r - c\right)}, \tag{8.2.19.15-1}$$

$(l + s_{kr1} + s_{kr2})$ – distance between two adjacent openings (Figures 8.2.19.15-1 and 8.2.19.15-2), [mm];
D_a – outside diameter of wall being reinforced, [mm];
s_r – actual thickness of wall being reinforced, [mm];
c – design thickness allowance, [mm], (see sub-chapter 8.2.7).

Where $(l + s_{kr1} + s_{kr2}) < \sqrt{D_a(s_r - c)}$, the stress occurring in the section between the openings due to design pressure shall be checked. Both longitudinal and lateral stresses in that section shall not exceed the allowable values determined in accordance with the formula below:

$$\frac{F}{f_c} \le \sigma \qquad (8.2.19.15\text{-}2)$$

σ – allowable stress (see paragraph 8.2.4.5), [MPa];
F – load exerted by the design pressure upon the cross-section between openings (see paragraph 8.2.19.16), [N];
f_c – cross sectional area between openings (see paragraph 8.2.19.17), [mm^2].

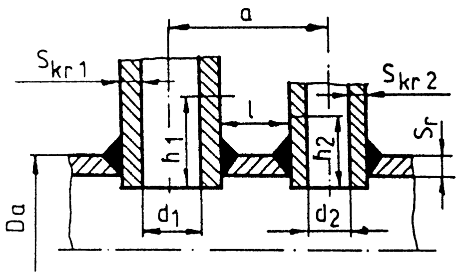


Fig. 8.2.19.15-1

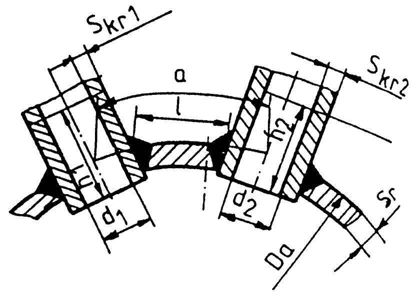


Fig. 8.2.19.15-2

8.2.19.16 Load exerted by the design pressure on the cross-sectional area between two openings shall be determined as follows:

.1 for openings arranged longitudinally along a cylindrical wall:

$$F_a = \frac{Dpa}{2}, \quad [\mathrm{N}] \qquad (8.2.19.16.1)$$

.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 between two adjacent openings, measured at the outside circumference, as shown in Fig. 8.2.19.15-2, [mm];
D – inside diameter (for conical walls measured at the centre of the opening), [mm];
p – design pressure, [MPa];
R_B – inner radius of curvature (see paragraph 8.2.19.10), [mm];
y – shape factor (see 8.2.16.1).

Where openings are arranged in cylindrical walls with a diagonal pitch, the load in question shall be determined in accordance with formula 8.2.19.16.2, and the obtained results shall be multiplied by the following factor:

$$K = 1 + \cos^2 \alpha \ , \qquad (8.2.19.16.3\text{-}2)$$

α – angle between the line of a row of openings and longitudinal axis, [deg].

8.2.19.17 For tubular strengthening elements, cross sectional area, f_c, [mm^2] between two adjacent openings shall be determined in accordance with the formula below:

$$f_c = l(s - c) + 0.5[h_1(s_{kr1} - c) + h_2(s_{kr2} - c)], \quad [\text{mm}^2] \qquad (8.2.19.17\text{-}1)$$

h_1 and h_2 – height of strengthening elements, [mm], determined in accordance with the following formulae:

for blind strengthening elements:

$$h_{1,2} = h_0 + s \ , \qquad (8.2.19.17\text{-}2)$$

for through strengthening elements:

$$h_{1,2} = h_0 + s + h_m \ , \qquad (8.2.19.17\text{-}3)$$

l – width of bridge between two adjacent openings (Figures 8.2.19.15-1 and 8.2.19.15-2), [mm];
s – thickness of wall being reinforced, [mm];
s_{kr1} and s_{kr2} – thicknesses of tubular strengthening elements (Figures 8.2.19.15-1 and 8.2.19.15-2), [mm];
c – design thickness allowance, [mm], (see sub-chapter 8.2.7);
h_0 – design height of the tubular stiffener (see formula 8.2.19.9.2-1), [mm];
h_m – design height of tubular strengthening element projecting inwards (see Figures 8.2.19.1-5, 8.2.19.1-6 and 8.2.19.13), [mm].

For openings to be strengthened by other means (combined or disk-shaped strengthening elements, etc.), the values of f_c shall be determined in accordance with the same procedure.

8.2.19.18 For drawn branch pieces arranged in a row, strength factor φ, determined for this row in accordance with formula 8.2.6.2.1, shall not be less than strength factor φ_A, obtained from curves B and B1 in Fig. 8.2.19.7. For $\varphi < \varphi_A$, the value of φ shall be used to determine the wall thickness in accordance with paragraph 8.2.19.14.

This requirement also applies to welded branch pieces arranged in a row, whose thickness is determined only for the internal pressure effect.

8.2.20 Stays

8.2.20.1 Cross-sectional area of long and short stays, corner stays and stay tubes, subjected to tensile or compressive stresses shall not be less than that determined in accordance with the formula below:

$$f = \frac{pf_s}{\sigma \cos \alpha}, \qquad (8.2.20.1)$$

f – cross-sectional area of single stay, [mm^2];
p – design pressure (see sub-chapter 8.2.2), [MPa];
σ – allowable stress (see sub-chapter 8.2.4.5), [MPa];
α – angle between the corner stay and the wall to which the stay is attached, [deg], (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 lines passing at right angles through the centres of the lines interconnecting the centre of stay with the adjacent points of support (stays). The cross-sectional area of the stays and tubes within this area may be determined according to the surface area per stay.

8.2.20.2 For stays subjected to bending, the allowable bending stress shall be determined with a safety factor not less than 2.25.

8.2.20.3 In the case of end plates with a single strengthening stay (Fig. 8.2.20.3), the stay shall be so designed as to make it capable of bearing at least half the load acting on the end plate. Thickness of such an end plate shall fulfil the requirements specified in paragraph 8.2.12.1.

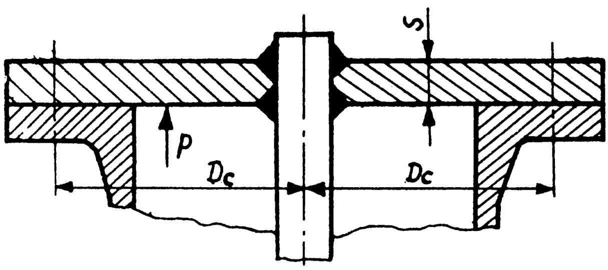


Fig. 8.2.20.3

8.2.20.4 Stay and regular fire tubes shall have thickness not less than the values specified in Table 8.2.20.4.

Thickness of stay tubes with diameter over 70 mm shall not 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 Cross-sectional area of welds connecting stays shall be such as to fulfil the following requirement:

$$\frac{\pi d_a e}{f} \geq 1.25, \qquad (8.2.20.5)$$

d_a – stay diameter or outside diameter for tubes, [mm];
e – weld thickness (Figures 5.1 ÷ 5.3 in the Annex), [mm];
f – cross-sectional area of the stay (see paragraph 8.2.20.1); [mm^2].

8.2.20.6 For flared tubes, the flared belt length in the tube plate shall not be less than 12 mm. The flared joints for working pressures above 1.6 MPa shall be made with sealing grooves.

8.2.20.7 Flared 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 tube flanging;
s – tube wall thickness, [mm];
l – flared belt length, [mm].

For other symbols – see paragraph 8.2.20.1.

The length of flared belt in tubes l shall not be taken greater than 40 mm.

8.2.20.8 The flared length of plain pipes shall be not less than that determined in accordance with the formula below:

$$l = \frac{pf_s K_r}{q}, \quad [\text{mm}] \qquad (8.2.20.8\text{-}1)$$

K_r = 5.0 safety factor of flared joint,
p, f_s – see paragraph 8.2.20.1,
q – strength of pipe joint over 1 mm of flared belt, evaluated experimentally from the formula given below, [N/mm]:

$$q = \frac{F}{l_1} \tag{8.2.20.8-2}$$

F – axial force necessary to extract the flared tube from the tube plate, [N];
l_1 – length of flared belt used for experimental determination the of value of q [mm].

8.2.21 Top Girders

The section modulus of top girders with rectangular cross-section shall not be less than that determined in accordance with the formula below:

$$W = \frac{1000M}{1.3\sigma Z}, \tag{8.2.21-1}$$

W – section modulus for single girder, [mm^3],
σ – allowable stress (see paragraph 8.2.4.5), [MPa],
Z – rigidity factor for the wall being strengthened, for the structure as shown in Fig. 8.2.21, $z = 1.33$,
M – bending moment per one girder, [Nm]; for a rectangular section, the moment shall be determined in accordance with the formula below:

$$M = \frac{pal^2}{8000} \tag{8.2.21-2}$$

l – design length of the girder, [mm], (Fig. 8.2.21),
p – design pressure, [MPa],
a – spacing between axes of adjacent girders, [mm],

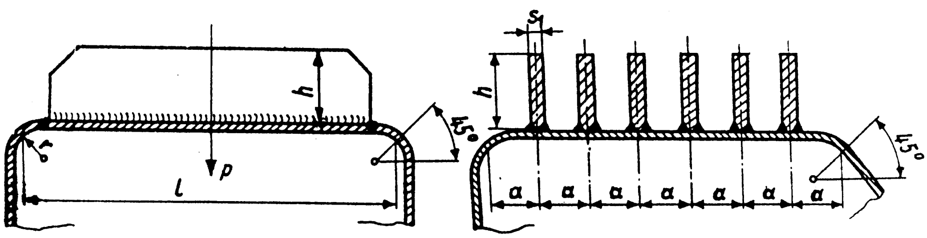


Fig. 8.2.21

s_1 – width of girder, [mm], (Fig. 8.2.21),
h – height of girder which shall not exceed $8s_1$ [mm], (Fig. 8.2.21).

9 BOILERS

9.1 Boiler Design

9.1.1 Boilers shall be designed for the conditions specified in sub-chapter 1.6.1 of *Part VI – Machinery Installations and Refrigerating Plants*.

9.1.2 The thickness of tubes thinned in the process of bending shall not be less than the design value.

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.

Drilled holes shall be provided at short-stay ends, as shown in Fig. 5.3 in the Annex.

9.1.4 For walls reinforced by short stays and exposed to flame and high-temperature gases, the distance between stay centres shall not be larger 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 shall be so done that the load involved is transferred, as far as possible, directly on to the boiler shell or the most rigid of its parts.

9.1.6 The distance between furnaces and boiler shell shall not be less than 100 mm. The distance between any two furnaces shall not be less than 120 mm.

9.1.7 Branch pieces installed to 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 shall not be subjected to excessive bending stresses and shall be reinforced by stiffening fins if so required.

9.1.8 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, preferably from both sides. Branch pieces may also be welded from one side, using removable backing strip or by some other method that ensures penetration throughout the entire thickness of the boiler wall.

9.1.9 Boiler drums and headers of wall thickness greater than 20 mm, as well as superheated 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.

9.1.10 Where use is made of non-metal sealing gaskets for closures of manholes and other openings, the design shall prevent the possibility of gaskets being forced out.

9.1.11 Suitable design provisions shall be made to prevent steam formation in economizers of boilers.

9.1.12 A name-plate including all principal particulars of the boiler shall be provided in a visible place on the boiler.

9.1.13 The fastening elements on boilers, except for the elements not being under load, shall not be welded directly to the boiler shells but shall be attached to the welded pads.

9.1.14 Tubes flue rolled on headers and tube plates shall be of seamless type.

9.1.15 Boilers with finned pipes shall 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 the 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 in a range of bores up to 15 mm.

The construction of welded pads, branches shall fulfil the requirements specified in sub-chapter 8.2.19.

9.2.2 Valve covers shall be secured to valve cases by studs or bolts. Valves with bore diameters of 32 mm and less may have screwed joints provided that there are means preventing them from being loosened.

9.2.3 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.

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 main boiler and each auxiliary boiler for essential services shall be equipped with at least two feed valves. Auxiliary boilers for other services, and also waste-heat boilers may have one feed valve each.

9.3.2 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 specified in Chapter 17 of *Part VI – Machinery Installations and Refrigerating Plants*.

9.4 Water Level Indicators

9.4.1 Every boiler with a free water surface shall be provided with at least two independent water level indicators with reflecting glass (see paragraph 9.4.3).

Subject to PRS acceptance in each particular case, one of water level indicators may be replaced by:

– suitable safety and indication means of lower and upper water level; (safety and indication sensors shall be independent), or
– remote, 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 waste-heat boilers with free water evaporating surface and steam reservoirs (steam separators) may be provided with single water level indicators with reflecting glass.

9.4.2 Forced circulation boilers shall be provided with two independent alarms to signal a shortage of water flow. The second alarm is not required, provided the requirements specified in sub-chapters 4.2 and 4.3 and also in Table 21.3.1-1, *Part VIII – Electrical Installations and Control Systems* are fulfilled. This requirement does not apply to waste-heat boilers.

9.4.3 Flat prismatic reflecting glass shall be used in water level indicators for boilers with a working pressure of less than 3.2 MPa. 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 front of the boiler, at an equal and possibly shortest distance from the vertical centre plane of the 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 gauges and their branch pipes shall not be 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:

32 mm – for bent branches of main boilers;
20 mm – for straight branches of main boilers and 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 for highly reliable remote water level indicators (placed at lower position) or other types of water gauges approved by PRS. This requirement does not apply to waste-heat boilers and their steam receivers.

9.4.11 The indication error of the remote water level indicators shall not 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 shall not exceed 10% of the distance between the lower and the upper level.

9.5 Marking of 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 close to the water level indicators.

The lowest level reference line, as well as the plate shall not be covered with boiler insulation.

9.5.2 The lowest water level in the boiler shall not be less than 150 mm above the highest heating-surface point. This distance shall also be maintained when the ship is listed 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 down 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 waste-heat 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 valves or cocks 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 the design capacity below 750 kg/h and waste-heat 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 with suitable range.

Where remote temperature control is installed, the local thermometers shall also 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 drum, as a rule, on a common branch piece; additionally one valve shall be fitted on the 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 and steam reservoirs (steam separators).

9.7.2 Aggregate cross-sectional area, *f*, of safety valves shall not be less than that determined in accordance with the formula below:

for saturated steam

$$f = K\frac{G}{10.2p_w+1}\ , \quad [\mathrm{mm}^2] \qquad (9.7.2\text{-}1)$$

for superheated steam

$$f = K\frac{G}{10.2p_w+1}\sqrt{\frac{V_H}{V_S}}\ , \quad [\mathrm{mm}^2] \qquad (9.7.2\text{-}2)$$

f – aggregate cross-sectional area of safety valves, [mm^2];
G – design capacity, [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 and temperature, [m^3/kg];
K – factor as per Table 9.7.2.

Table 9.7.2

Valve lift	*K* factor
$\frac{d}{20} \le h \langle \frac{d}{16}$	22
$\frac{d}{16} \le h \langle \frac{d}{12}$	14
$\frac{d}{12} \le h \langle \frac{d}{4}$	10.5
$\frac{d}{4} \le h \langle \frac{d}{3}$	5.25
$\frac{d}{3} \langle h$	3.3

d – minimum diameter of valve, [mm];
h – valve lift, [mm].

Safety valves shall not be less than 32 mm or more than 100 mm in diameter.

If specially approved by PRS, the use of valves with smaller areas than required in accordance with formulae 9.7.2-1 and 9.7.2-2 may be allowed, provided it is 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 non-disconnectable superheater may be included in the aggregate area of the valves to be determined in accordance with formulae 9.7.2-1 and 9.7.2-2. This area shall not 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 main and auxiliary boilers of essential services, after being lifted shall stop the steam escape at the pressure not less than 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 shall not be less than 1.1 times the aggregate cross-sectional area of the valves installed.

9.7.7 The cross-sectional area of the outlet steam branch of the safety valve, as well as of the pipe connected thereto, shall not be less than twice the aggregate 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.

Supply pipes leading to the safety valves are not allowed to be installed inside the boiler, nor can any provision be made on the safety valve bodies or their connections for steam extraction or for other purposes.

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 remote control gear for safety valves of steam superheaters, waste-heat boilers and their steam tanks (separators) shall be operated 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 without the knowledge of the personnel.

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 Stop valves of the main and auxiliary steam lines shall be provided with remote control gear for the operation from the upper deck or from other readily accessible position outside the boiler room.

9.8.3 Where a single main boiler or a single auxiliary boiler of essential services is 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 Valves

9.9.1 Boilers shall be fitted with blow-down and scum arrangements and, where necessary, with drain valves.

Blow-down and drain valves shall be fitted directly to the boiler shell. For boilers of working pressure lower than 1.6 MPa, these valves may be installed on welded-on branch pieces.

Steam superheaters, economizers and steam accumulators shall be provided with blow-down valves or drain valves.

9.9.2 The inside diameter of blow-down valves and pipes shall not be less than 20 mm or 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 The scum arrangements in boilers with a free water surface (evaporating surface) shall be such as to ensure 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 internal surfaces. Where the arrangement of manholes is not possible, provision shall be made for sight holes.

9.12.2 Manhole openings shall have dimensions not less than:
300 x 400 mm – for oval openings, or
400 mm – for round openings.

In separate 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 working water 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 Incinerating Boilers

9.13.1 These provisions apply to auxiliary boilers utilized for incinerating garbage and oil wastes of flash point above 60 °C.

9.13.2 Automatic control systems of incinerating boilers for unmanned operation shall fulfil the relevant requirements of Chapter 20 of *Part VIII – Electrical Installations and Control Systems*.

9.13.3 Special furnace chamber shall be provided for incineration of garbage and oil wastes, the chamber shall fulfil 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 shall not 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, however not less than 45 cm^2;

.5 the chamber charging device shall be such as to prevent the simultaneous opening. 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.

9.13.4 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.

9.13.5 Incinerating boilers shall be provided with effective system of soot removal.

9.14 Thermal Oil Heaters

9.14.1 The provisions of this Section refer to heaters for thermal oils. Thermal oil heaters are, in general, to be installed in separate spaces, equipped with exhaust ventilation, capable to perform at least 6 air changes per hour.

9.14.2 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.

9.14.3 The construction of combustion chambers and burners shall secure uniform heat distribution.

Only such non-uniformity of heat distribution may be admitted at which the temperature in thermal oil boundary layer at any place of the heating surface does not exceed the upper allowable temperature limit for the thermal oil used.

The construction of combustion chamber and location of burners shall prevent 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 hermetization devices and a separate smothering system of type approved by PRS.

9.14.4 Each thermal oil heater shall be fitted with:

– shut-off valves at inlet and outlet of thermal oil. Such valves of oil fired and exhaust gas fired thermal oil heaters 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 provided that the requirement specified in paragraph 14.6.2 of *Part VI – Machinery Installations and Refrigerating Plants* is fulfilled;
- pressure gauge;
- at least two spring-loaded safety valves of closed type, of identical construction and dimensions, the throughput of each one being not less than the capacity of circulating pump. The cross-sectional area of safety valves shall not be less than that corresponding to the diameter of 32 mm and not greater than that corresponding to the diameter of 100 mm;
- arrangements for taking samples of thermal oil;
- inspection openings in accordance with sub-chapter 9.12.

9.14.5 Each thermal oil heater shall be equipped with effective means for soot removal.

9.14.6 Thermal oil heater tubes shall be connected to headers and chambers by welding.

9.14.7 Bellows type valves shall be applied to thermal oil boilers. Application of gland type fittings is subject to PRS acceptance in each particular case.

9.14.8 Thermal oil heaters shall be provided with alarm and safety system activated at limit temperatures of thermal oil and exhaust gas, fitted at the outlet of the heaters.

9.14.9 Thermal oil heaters shall be provided with automatic combustion control, audible and visual alarm, interlock device in accordance with the requirements specified in paragraph 11.2.1, as well as protective device as specified in paragraph 11.2.2.

9.15 Water Heating Boilers

Construction and materials of water heating boilers shall fulfil the requirements for steam boilers.

9.16 Additional Requirements for Waste-heat Boilers

9.16.1 Waste-heat boilers shall be fitted with devices closing the supply of hot gas to the boiler in the case of alarm system activation.

9.16.2 The boiler shall be so designed and installed that all tubes can be easily and readily inspected for any signs of corrosion and leakage.

9.16.3 The boiler shall be fitted with temperature sensor(s) and fire detection alarm.

9.16.4 A fixed fire extinguishing and cooling systems shall be fitted. A sprinkler system of sufficient capacity may be accepted.

The exhaust duct below the boiler 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.

9.16.5 Except for the case mentioned in paragraph 9.16.6.1, only one safety valve may be installed on waste-heat boilers.

9.16.6 Waste-heat boilers that may be isolated from the steam plant system in a flooding condition shall fulfil the following requirements:

.1 shell type boilers having a total heating surface of 50 m^2 or more shall be provided with at least two safety valves,

.2 shell type boilers shall be provided with removable lagging at the circumference of the tube end plates to enable ultrasonic examination of the tube plate to shell connection,

.3 the manufacturer shall provide operating instructions for each boiler which shall include reference to:
- feed water treatment and sampling arrangements,
- operating temperatures – exhaust gas and feed water temperatures as well as operating pressure,
- inspection and cleaning procedures,
- records of maintenance and inspection,
- the need to maintain adequate water flow through the boiler under all operating conditions,
- periodical operational checks of the safety devices to be carried out by the operating personnel and to be documented accordingly,
- procedures for using the waste-heat boiler in the dry condition,
- procedures for maintenance and overhaul of safety valves.

10 CONTROL, SAFETY AND ALARM SYSTEMS OF BOILERS

10.1 General Requirements

10.1.1 The requirements specified this Chapter apply to permanently attended boilers.

The requirements for control system, alarm system and safety system of unattended boilers are specified in sub-chapter 20.7 and Chapter 21 of *Part VIII – Electrical Installations and Control Systems*.

10.2 Control Systems

10.2.1 Main water-tube boilers and 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.

10.2.2 The control systems shall be capable of maintaining the water level, steam pressure and other variable parameters within the predominated limits over the entire load range and to ensure quick changes of boiler load.

10.3 Safety Systems

10.3.1 The boilers shall be provided with non-detachable system ensuring the water level in the boiler (see sub-chapter 9.5) not to fall beneath the lowest permissible level.

10.3.2 The boilers with automatic control of combustion shall be provided with a safety system in accordance with the requirements specified in sub-chapter 11.2.

10.4 Alarm Systems

10.4.1 Boilers with automatic control of feed and combustion shall be provided with audible and visual alarm system at the control stand.

10.4.2 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 paragraph 11.2.3),
- salinity of feed water exceeding the permissible level (see paragraph 17.2.4 of *Part VI – Machinery Installations and Refrigerating Plants*).

10.4.3 The lowest water level alarms of the main boilers and auxiliary boilers of essential services shall be activated prior to the activation of the safety system.

10.4.4 Provision shall be made for the audible alarm to be switched off manually after its activation.

11 OIL FUEL INSTALLATIONS OF BOILERS

11.1 General Requirements

11.1.1 All the components of oil fuel installation such as pumps, fans, quick closing valves and electric drives, shall be of type approved by PRS and shall be manufactured and tested under the survey of PRS or other competent technical inspection body.

Electric equipment, control, safety and alarm systems shall fulfil the relevant requirements specified in *Part VIII – Electrical Installations and Control Systems*.

Piping systems and fittings of oil fuel installation shall fulfil the relevant requirements specified in *Part VI – Machinery Installations and Refrigerating Plants*.

11.1.2 The requirements specified in this Chapter apply to the equipment for firing the boilers with fuel oil of flash point not less than 60 °C.

Where crude oil or slops are used as the fuel for tanker boilers, then, according to requirements specified in sub-chapter 22.5.5 of *Part VI – Machinery Installations and Refrigerating Plants*, the furnaces and smoke uptake pipes shall be gas-tight and tested for gas tightness before being taken in use.

11.1.3 Burners shall be so designed as to ensure the possibility for the control of the flame jet size and shape.

11.1.4 In the case of variable-delivery burners, provision shall be made to control the amount of combustion air.

11.1.5 Proper design solutions shall be applied to preclude the possibility of turning and removing the burners from their positions before cutting-off the fuel supply.

11.1.6 Where the fuel is atomized by means of steam or air, the construction of burners shall preclude the possibility of penetration of steam or air to fuel oil and vice versa.

11.1.7 Where fuel preheating is applied, provision shall be made to preclude the possibility of fuel overheating when the boiler capacity has been reduced or the burners have been cut off.

11.1.8 Proper drip trays shall be provided where fuel leaks may be expected.

11.1.9 Proper sight glasses shall be provided to monitor the combustion process in the furnace. Means shall be provided to prevent flame and hot air outburst when the burner is removed.

11.1.10 Proper arrangement shall be provided for the storage and smothering of the manual ignition torch.

It is recommended that the inlets of boiler fans be protected against penetration of moisture and solids.

11.2 Additional Requirements for Permanently Attended Boilers with Automatic Firing Control

11.2.1 Firing installations of boilers shall be provided with an interlock to enable the fuel supply to the furnace only when the following conditions are fulfilled:

.1 the burner is in the operating position,
.2 all electrical equipment is connected to the 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.

In general, the shut-off of fuel supply shall be effected by two self-closing valves connected in series. Where the daily service tank is situated below the furnace, one such valve is sufficient.

11.2.2 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 case of:

.1 low pressure of combustion air or decay of combustion air flow,
.2 burner flame failure,
.3 water level in the boiler reaching its lower limit.

Activation of protective devices shall actuate visual and audible alarms.

11.2.3 Firing installations shall be equipped with a burner flame jet monitor. Such monitor shall respond only to the flame of the burner under control.

11.2.4 Capacity of the pilot burner shall be such that the burner is not capable of maintaining, by itself, 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 in the cases mentioned in paragraph 11.2.2, both burners shall stop their operation at the same time.

11.2.5 Firing installation of the main and auxiliary boilers of essential services shall be capable of being started up and controlled manually. Manual control arrangements shall be located as close to the boiler as possible.

While the firing installation is being manually controlled, all the automatic control arrangements mentioned in paragraphs 11.2.1 and 11.2.2 shall be in operation.

11.2.6 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.

12 PRESSURE VESSELS AND HEAT EXCHANGERS

12.1 Construction of Pressure Vessels and Heat Exchangers

12.1.1 Components of pressure vessels and heat exchangers being in contact with sea water or other possibly corrosive media shall be constructed from corrosion-resistant materials. In the case of other materials, the method of their protection against corrosion is subject to PRS acceptance in each particular case.

12.1.2 Construction of pressure vessels and heat exchangers shall provide their reliable operation in the conditions specified in sub-chapter 1.6.1 of *Part VI – Machinery Installations and Refrigerating Plants*.

12.1.3 Pressure vessels and heat exchangers shall fulfil the requirements specified in paragraphs 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.

12.1.4 Where necessary, construction of pressure vessels and heat exchangers shall take account of possible thermal expansion of the shell and other components.

12.1.5 Shells of heat exchangers and pressure vessels shall be fixed to their seatings by supports. Upper fixing arrangements shall be provided if necessary.

Construction of the fixing arrangements for pressure vessels and heat exchangers to the foundations shall also take account of the requirements specified in sub-chapter 1.11 of *Part VI – Machinery Installations and Refrigerating Plants*.

12.2 Fittings and Gauges

12.2.1 Pressure vessels and heat exchangers or their inseparable sets shall be fitted with non-disconnectable safety valves. In the case of several non-interconnected spaces, safety valves shall be provided for each space. Hydrophore tanks shall be fitted with safety valves located on the waterside.

In justified cases, PRS may waive the above-mentioned requirements.

12.2.2 In general, safety valves shall be of a spring-loaded type. Safety diaphragms of a type approved by PRS are permitted in fuel and oil heaters, provided they are installed on the fuel and oil side.

12.2.3 The discharge capacity of safety valves shall be such that under no conditions the working pressure is exceeded by more than 10 %.

12.2.4 Safety valves shall be so designed as to be capable of being sealed or fitted with an equivalent means to prevent their unauthorised adjustment. Materials used for springs and sealing surfaces of valves shall be resistant to corrosive effect of the medium.

12.2.5 Level indicators and sight glasses may only be installed on pressure vessels and heat exchangers where required by the conditions of control and inspection. Level indicators and sight glasses shall be of reliable construction and protected adequately. For steam, oil and refrigerants, flat glass plates shall be used for level indicators and sight glasses.

12.2.6 Pressure vessels and heat exchangers shall be provided with flanges or flanged branch pieces for installation of fittings and mountings.

In hydrophore tanks, threaded branch pieces may also be applied.

12.2.7 Pressure vessels and heat exchangers shall be provided with adequate blowdown arrangements as well as drain arrangements.

12.2.8 Pressure vessels and heat exchangers shall be provided with manholes for internal examination. Where the manholes are impracticable, adequate sight holes shall be provided. Pressure vessels and heat exchangers with more than 2.5 m in length shall be provided with the inspection holes at both ends.

Where the pressure vessel or heat exchanger is of dismountable construction or where corrosion and contamination of internal surfaces is precluded, manholes or inspection holes are not required.

Manholes or sight holes are not required where the construction of pressure vessel or heat exchanger precludes the possibility of inspection through such holes.

For the dimensions of manholes' openings – see paragraph 9.12.2.

12.2.9 Pressure vessels and heat exchangers, as well as their inseparable units shall be equipped with a pressure gauge or a compound pressure gauge. In heat exchangers divided into several spaces, a pressure gauge or a compound pressure gauge shall provided for each space.

Pressure gauges shall fulfil the requirements specified in paragraphs 9.6.1 and 9.6.5.

12.2.10 Fuel heaters where the fuel temperature may exceed 220 °C shall be fitted – apart from the temperature controller – also with sensor warning about high temperature or stopped flow of fuel.

For electric heaters – see also sub-chapter 15.4 of *Part VIII – Electrical Installations and Control Systems*.

12.3 Requirements for Particular Types of Pressure Vessels and Heat Exchangers

12.3.1 Air Receivers

12.3.1.1 Safety valves of starting air receivers for main and auxiliary engines, as well as of fire protection systems, after being lifted, shall completely stop the air escape at the pressure inside the receiver not less than 0.85 of the working pressure.

12.3.1.2 Where air compressors, reducing valves or pipes from which air is supplied to the receivers are provided with safety valves so adjusted to prevent the receivers from being supplied with air of the pressure higher than the working pressure, safety valves need not be fitted on such receivers. In that case, fusible plugs shall be fitted on the receivers instead of the safety valves.

12.3.1.3 The fusible plugs shall have a fusion temperature within 100 – 130 °C. The fusion temperature shall be permanently marked on the fusible plug. Air receivers having a capacity over 0.7 m^3 shall be fitted with plugs not less than 10 mm in diameter.

12.3.1.4 Air receivers shall be equipped with water-draining arrangements. In air receivers positioned horizontally, the water draining arrangements shall be installed at both ends of the receiver.

12.3.2 Cylinders for Compressed Gases

12.3.2.1 Cylinders for compressed gases are portable pressure vessels designed for the storage of compressed gases, refrigerants or CO_2, which are stored on board the ship for her operational purposes, but are incapable of being filled by means of the ship's equipment.

12.3.2.2 Strength calculations shall be performed in respect of the requirements specified in sub-chapter 8.2.8 and the following:
- design pressure shall not be less than the pressure which may occur at temperature 45 °C, at the predetermined filling level;
- allowable stress σ shall be determined in accordance with sub-chapter 8.2.4, whereas the safety factor in accordance with paragraph 8.2.5.1;
- allowance *c* for cylinders being exposed to corrosion shall not be taken less than 0.5 mm.

Cylinders may be made of steel with the yield stress greater than 750 MPa but not exceeding 850 MPa, subject to PRS acceptance in each particular case.

12.3.2.3 Non-disconnectable safety devices of approved construction shall be provided to prevent a dangerous overpressure in the cylinder in case of temperature increase. Safety valves or burst disks activated at a pressure exceeding 1.1 times the working pressure but not higher than 0.9 times the test pressure are permitted.

12.3.2.4 Cylinders shall be permanently marked to include the following information:

.1 manufacturer's name,
.2 serial number,
.3 year of manufacture,
.4 kind of gas,
.5 capacity,
.6 test pressure,

.7 tare,
.8 maximum load (pressure/weight),
.9 stamp and date of testing.

12.3.2.5 Cylinders shall be hydraulically tested under pressure equal to 1.5 times the working pressure.

12.3.2.6 Cylinders which are designed for the storage of compressed gases, refrigerants or extinguishing agents shall be approved by PRS or shall be manufactured in accordance with the relevant standards under the survey of a competent technical inspection body approved by PRS.

12.3.3 Condensers

12.3.3.1 Construction of condensers and their location on board shall be such as to enable tube replacement.

In general, the main condenser shell shall be of steel welded construction.

Baffles shall be provided inside condensers, at excess pressure steam inlets, to protect the tubes from the direct steam impact.

Tube fixing shall be so designed as to prevent sagging and dangerous vibration of the tubes.

12.3.3.2 Covers of condenser water chambers shall be provided with manholes in a number and position as may be required to ensure access to the tubes for the purposes of flaring, packing replacement or plugging of any tube.

Cathodic protection shall be provided to prevent electrolytic corrosion of the water chambers, tube plates and tubes.

12.3.3.3 The main condenser shall be capable of operating in emergency conditions with any turbine casing detached.

12.3.3.4 Construction of condenser shall enable fixing of monitoring and measuring devices.

12.3.4 Pressure Vessels and Heat Exchangers of Refrigerating Installations

The requirements specified in sub-chapters 12.1, 12.2, 12.3.2 and 12.3, except for paragraphs 12.3.3.3 and 12.3.3.4, apply to pressure vessels and heat exchangers of the refrigerating and fire extinguishing installations, whereas the requirements specified in sub-chapter 12.2.1 may be considered as guidelines.

Pressure vessels and heat exchangers shall also fulfil the relevant requirements specified in *Part V – Fire Protection* and in Chapter 21 of *Part VI – Machinery Installations and Refrigerating Plants*.

12.3.5 Pressure Vessels for Processing Fishery Products

12.3.5.1 Pressure vessel covers opened periodically shall be fitted with devices preventing a partial closing or spontaneous opening of the covers. Provision shall be made to preclude the possibility of opening the cover in the case of excessive pressure or underpressure, as well as to preclude the possibility of pressurizing the receiver when the cover is partially closed.

12.3.5.2 The internal equipment, such as mixers, coils, disks, partitions, etc., hindering the internal inspection of the vessels shall be readily removable.

12.3.5.3 Sight glasses of not more than 150 mm in diameter, may be used to monitor the working spaces of mixers, provided that the working pressure in such spaces does not exceed 0.25 MPa.

12.3.5.4 In pressure vessels operating at a pressure exceeding 0.25 MPa, the covers of loading openings shall be so designed that, in the case of seal rupture, the hot medium escapes in a safe direction without hazard for the personnel.

12.3.5.5 Pressure vessels operating under vacuum conditions, heated by steam or water of a temperature over 115 °C, shall be fitted with safety valves to prevent the pressure in the vacuum space from rising (due to the heating system leakage) higher than 0.85 times the test pressure.

These vessels shall be designed for such an opening pressure of the safety valve that the design stresses will not exceed 0.8 times the yield stress of the material at the design temperature.

12.3.5.6 For mixers heated by steam or water, as well as for the walls of vessels being in contact with the rotating product, the design wall thickness allowance, c , shall not be taken less than 2 mm.

12.4 Filters and Coolers

12.4.1 Filters and coolers of the main and auxiliary engines shall fulfil the requirements for heat exchangers and pressure vessels with respect to the materials and construction.

12.4.2 Oil fuel filters installed in parallel to enable their cleaning without cutting off the fuel oil supply to engines (duplex filters) shall be provided with arrangements protecting the filter under pressure against being opened inadvertently.

12.4.3 Oil fuel filters or filter chambers shall be provided with adequate means for:
– air venting when being put into operation,
– pressure equalisation before being opened.

Valves or cocks with drain pipes leading to a safe location shall be used for this purpose.

13 THRUSTERS

13.1 Application

13.1.1 The requirements specified in Chapter 13 apply to the ship propulsion, steering or manoeuvring devices which in this Chapter are also referred to as ”devices”. In particular, these requirements cover:
– azimuthing thrusters,
– cycloidal propellers,
– retractable and foldable devices,
– devices for dynamic positioning of the ship,
– water-jet propulsion,
– tunnel thrusters.

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

Other thrusters are considered as auxiliary ones.

13.2 General Requirements

13.2.1 If the ship is propelled by thrusters only, at least two separate devices with independent power supply shall be used. This requirement do not apply to water-jet propulsion.

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

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

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

13.2.4 Adequate means to prevent sea water penetration into both the device and ship hull shall be provided.

13.2.5 Dynamic seals preventing seawater penetration into the device or ship hull shall be type-approved by PRS.

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

13.2.7 Thrusters, which are so installed inside the ship hull as to enable their stretching out or turning, shall be located in a separate watertight compartment unless double seals are arranged in accordance with the requirement specified in paragraph 13.2.5. An alarm system warning of water ingress between the seals as well as the possibility of inspection of the seals during dry-docking shall be provided.

13.2.8 Construction of nozzles shall fulfil the relevant requirements specified in Chapter 2 of *Part III – Hull Equipment.*

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

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

13.3 Drive

13.3.1 Internal combustion engines which drive thrusters directly shall fulfil the requirements specified in Chapter 2. Installations serving engines shall fulfil the relevant requirements specified in *Part VI – Machinery Installations and Refrigerating Plants*, except for the requirement for application of stand-by and spare pumps and other similar appliances.

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

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

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

13.3.5 Main thrusters driven by electric motors shall fulfil the relevant requirements specified in Chapter 17 of *Part VIII – Electrical Installations and Control Systems*.

13.4 Gears and Bearings

13.4.1 Gearings applied in main thrusters shall fulfil the relevant requirements specified in Chapter 4.

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

13.4.3 Basic rating life L1O of rolling-element bearings in main thrusters shall be at least 20 000 hours.

13.4.4 Basic rating life L1O of rolling-element bearings in auxiliary devices shall not be less than 5 000 hours.

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

13.5 Propulsion Shafting

13.5.1 Propulsion shafts shall fulfil the relevant requirements specified in Chapter 2 of *Part VI – Machinery Installations and Refrigerating Plants* including the requirements for ice class where applicable.

13.5.2 With respect to torsional vibrations, the requirements specified in Chapter 4 of *Part VI* apply.

13.6 Propellers

13.6.1 Fixed pitch propellers and controllable pitch propellers shall fulfil the relevant requirements specified in Chapter 3 of *Part VI – Machinery Installations and Refrigerating Plants.*

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

13.7 Control Systems

13.7.1 Remote control systems of the thrusters shall fulfil the relevant requirements specified in sub-chapter 20.2 of *Part VIII – Electrical Installations and Control Systems.*

13.7.2 With respect to the main thrusters, the requirements of paragraphs 20.5.1; 2; 3; 8; 10; 11; 12; 13; 14; 15 of *Part VIII* are obligatory. It is recommended that all other requirements specified in Chapter 20 of *Part VIII* be taken into account.

13.8 Monitoring

13.8.1 Indicating system shall fulfil the relevant requirements specified in sub-chapter 20.4.3 of *Part VIII – Electrical Installations and Control Systems*.

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

13.8.3 Alarm system shall fulfil the requirements specified in sub-chapter 20.4.1 of *Part VIII – Electrical Installations and Control Systems* and the requirements specified in Table 13.8.3. The alarm system of auxiliary devices with an installed power below 200 kW is subject to PRS acceptance in each particular case.

Table 13.8.3
Alarm system of thrusters

Item	Component, installation, system	Parameter under control	Alarm system: Parameter value signalled	Notes
1.	Hydraulic drive of: – propeller, – device rotation, – propeller pitch change	level in spare hydraulic oil tank	minimum	–
2.		hydraulic oil pressure	minimum	–
3.		pressure difference in hydraulic oil filter	maximum	
4.		hydraulic oil temperature	maximum	if cooler is applied
5.	Lubricating oil system	oil pressure or oil level in gravity tank	minimum	
6.	Electrical motor drive of: – propeller, – device rotation, – propeller pitch change	Acc. to *Part VIII – Electrical Equipment and Automatic Control*, Table 21.3.1-1, item 2.5		–
7.	Thruster monitoring system	alarm system power supply	minimum	–
8.		control system power supply	minimum	–
9.		emergency stop means acc. to paragraph 13.2.10	emergency stop	–
10.	Thruster compartment	fire detection	fire	–
11.		water level in bilge well*	high level	–

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

13.9 Survey, Testing and Certificates

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

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

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

13.9.4 Thrusters mentioned in paragraphs 13.9.1, 13.9.2 and 13.9.3 are subject to PRS survey both in production and testing in accordance with the requirements specified in paragraphs 13.9.6 and 13.9.7.

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

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

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

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

13.9.6.1 The survey covers materials used in the production – in accordance with paragraph 1.4.3.13, as well as welding, heat treatment and other procedures which are subject to acceptance within the process of the classification documentation approval.

13.9.6.2 All and any changes and exceptions with the approved documentation of type shall be submitted, together with justification, to PRS for approval. Product testing shall start after the changes and exceptions have been approved.

13.9.6.3 Pressure tests of casings shall be performed in accordance with the requirements specified in paragraph 1.5.2.1. In the case of hydrostatic pressure acting inside and/or outside the casing, working pressure p shall be taken for calculation as the highest hydrostatic pressure acting on one side in the lowest point of the casing.

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

Operating tests include:

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

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

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

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

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

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

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

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

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

13.9.7.2 During the trials of the monitoring systems, fulfilment of the requirements specified in sub-chapter 13.8 shall be demonstrated.

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

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

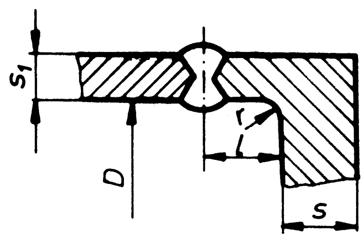
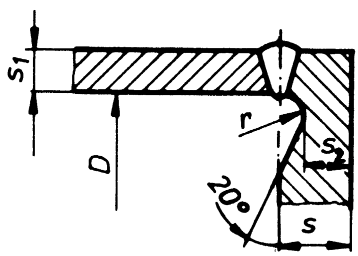
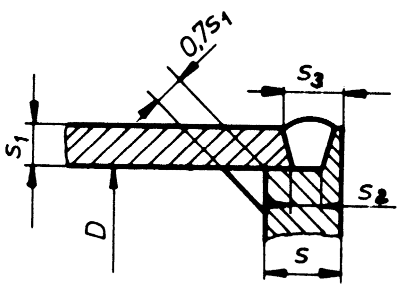
13.9.7.5 PRS may request submission of the sea trials report for consideration.

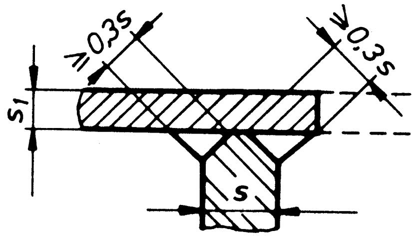
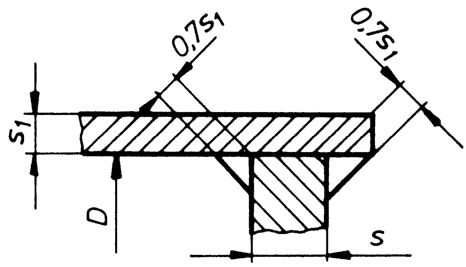
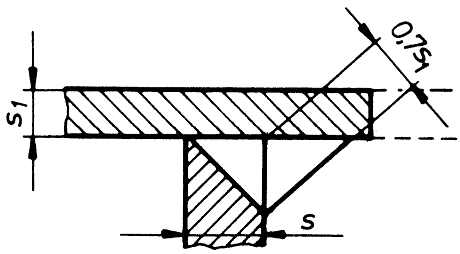
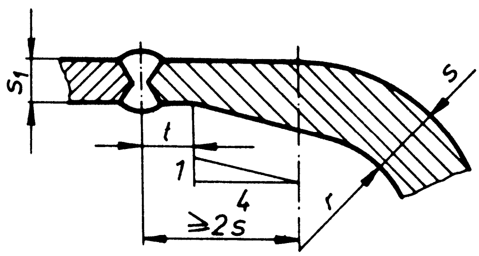
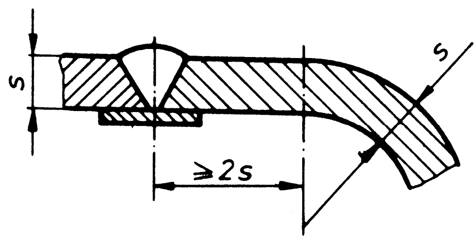
Annex

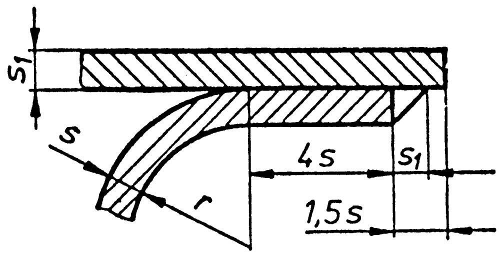
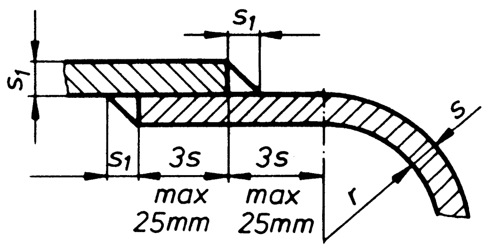
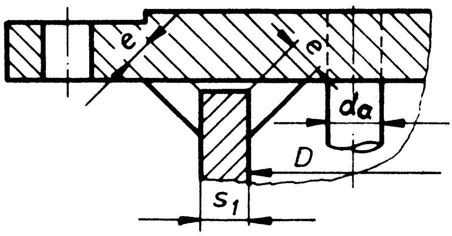
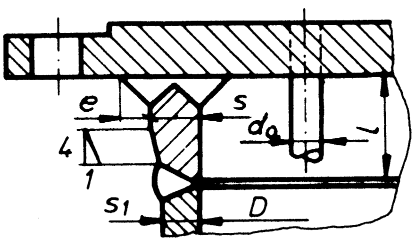
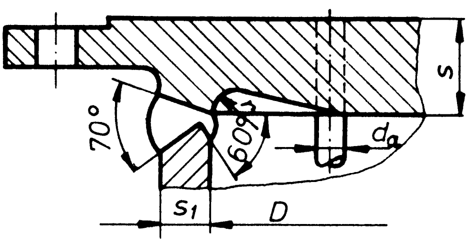
EXEMPLARY WELDED JOINTS USED IN BOILERS, PRESSURE VESSELS AND HEAT EXCHANGERS

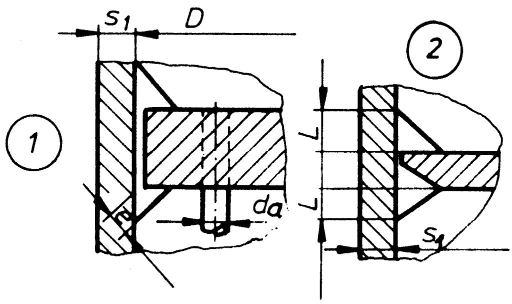
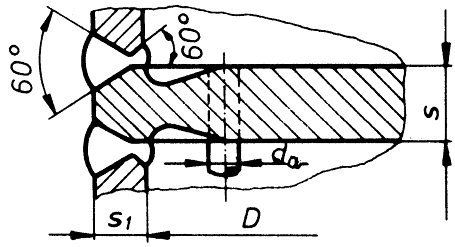
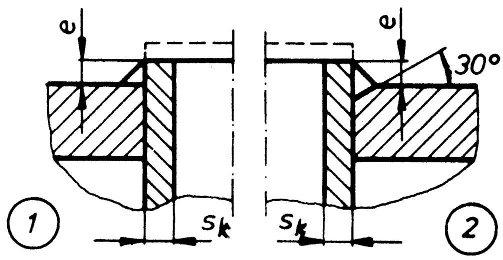
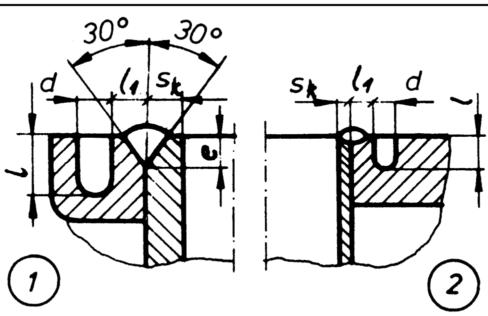
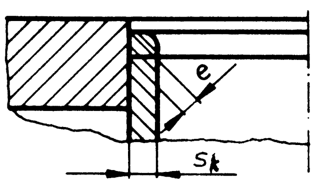
Dimensions of the components prepared for welding shall be determined in accordance with the national standards depending on the welding method. The examples of the most frequently used joints are shown in the tables below.

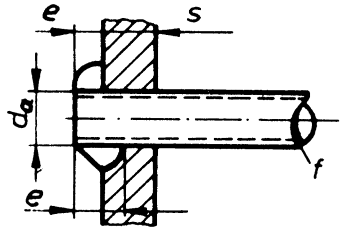
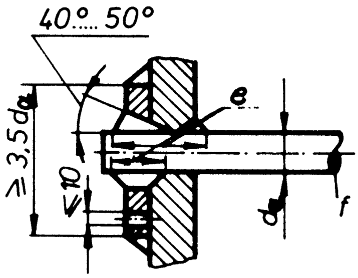
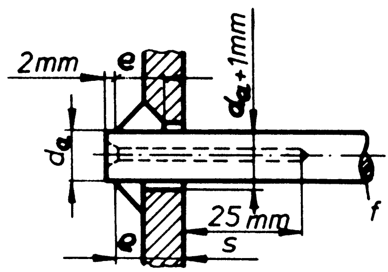
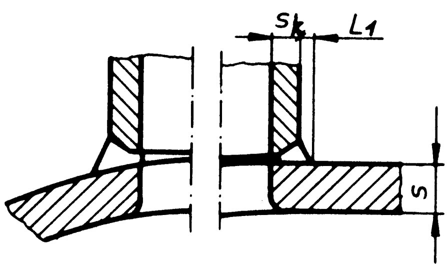
Other welded joints may also be performed having regard to the mechanical properties of parent materials and welding procedure. Such joints as well as necessary modifications to the exemplary joints are subject to PRS acceptance in each particular case.

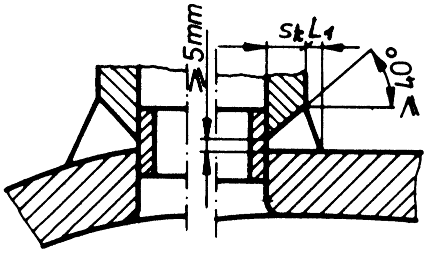
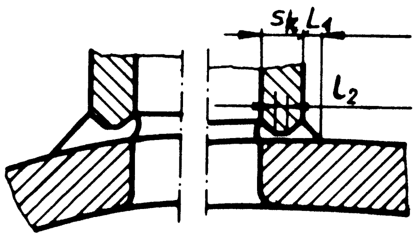
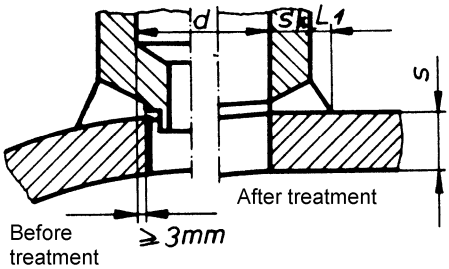
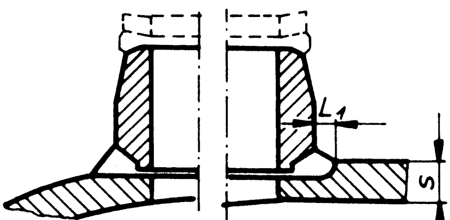
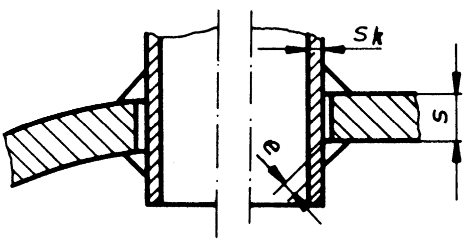
Item	Drawing (example)	Application
1	2	3
1	**Flat end plates and covers**	
1.1		$K = 0.38$ $r \geq \frac{s}{3}$ however not less than 8 mm $l \geq$ s
1.2		$K = 0.45$ $r \geq 0.2\ s$ however not less than 5 mm $s_2 \geq 5$ mm (see Note 1)
1.3		$K = 0.5$ $s_2 \leq s_1$ however not less than 6.5 mm $s_3 \geq 1.24\ s_1$ (see Note 1)

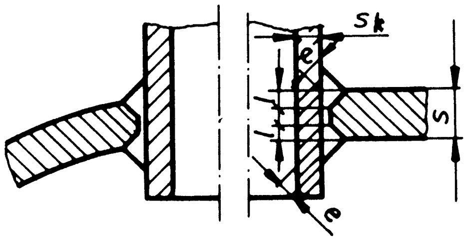
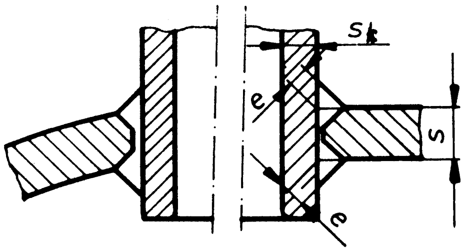
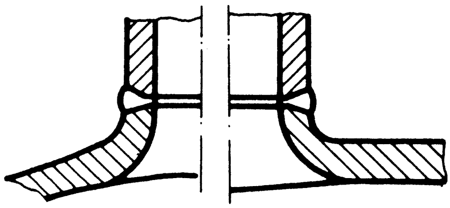
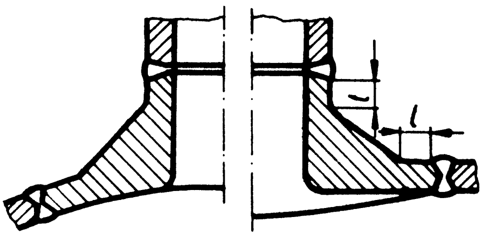
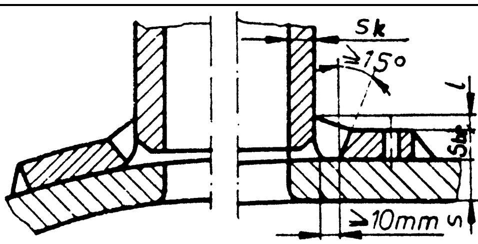
1	2	3
1.4		$K = 0.45$ (see Note 1)
1.5		$K = 0.55$ (see Note 1)
1.6		$K = 0.57$
2	**Dished ends**	
2.1		The joint may be used in boilers and pressure vessels of Class I, II and III (see Notes 2 and 17)
2.2		The joint may be used in boilers and pressure vessels of Class II and III

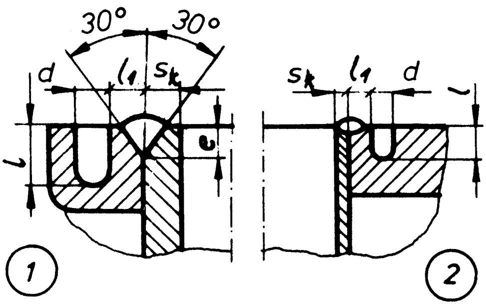
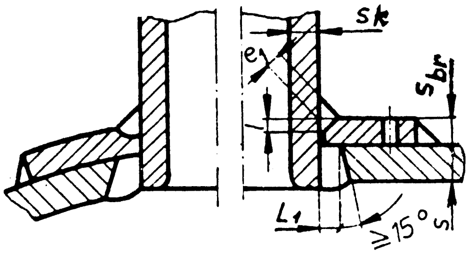
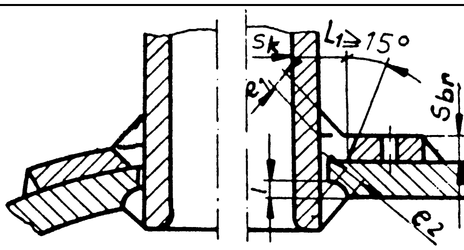
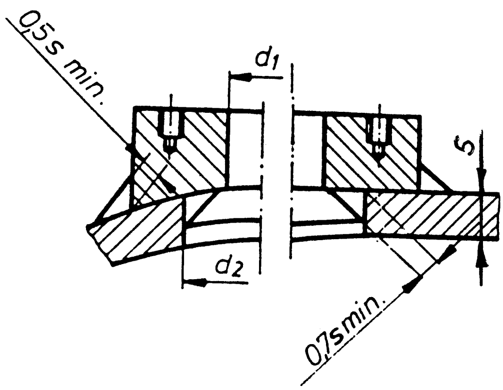
1	2	3
2.3		Not recommendable joint – it may be used only for pressure vessels of Class II not exposed to corrosion $s_1 \leq 16$ mm $D \leq 600$ mm
2.4		The joint may only be used for pressure vessels of Class III $s_1 \ \leq 16$ mm $D \leq 600$ mm
3	**Tube plates**	
3.1		$K = 0.45$ $e \ = 0.7\,s_1$ $s_1 \ \leq 16$ mm (see Notes 3 and 4)
3.2		$K = 0.45$ $e = \frac{1}{3} s_1$ $e > 6$ mm $s_1 \ > 16$ mm (see Notes 5 and 6)
3.3		$K = 0.45$ $r \geq 0.2\,s$ however not less than 5 mm

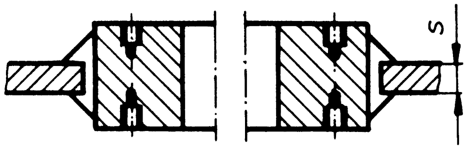
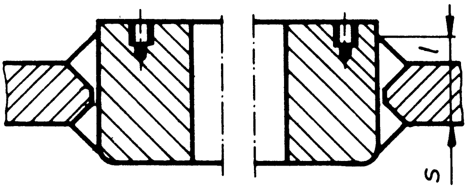
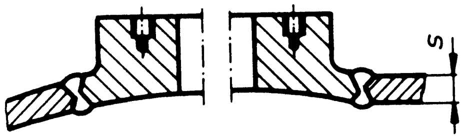
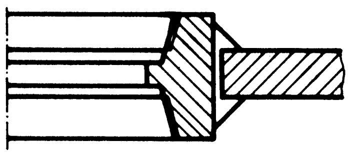
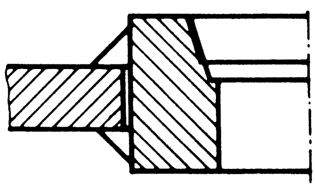
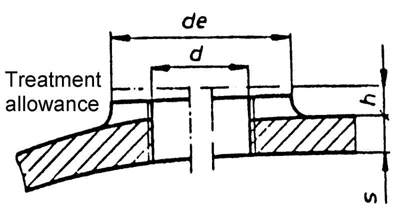
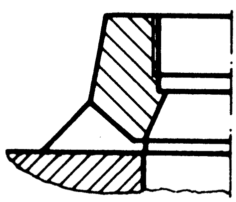
1	2	3
3.4		$K = 0.45$ $e \geq 0.7\, s_1$ if $L > 13$ mm variant 2 , where $L = \frac{1}{3} s_1$ and $L \geq 6$ mm is recommendable (see Note 7)
3.5		$K = 0.45$ $r \geq 0.2\, s$ however not less than 5 mm
4	**Tubes**	
4.1		$e = s_k$ $e \geq 5$ mm $s_k \geq 2.5$ mm (see Notes 8, 9 and 10)
4.2		$d = s_r$; $l_1 = s_r$ $1.5\, s_r < l < 2\, s_r$ 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 joints**	
6.1	**Non-through welded branch pieces**	
6.1.1		$s_k \le 16$ mm $L_1 = \frac{1}{3} s_k$ however not less than 6 mm

1	2	3
6.1.2		$L_1 = \frac{1}{3} s_k$ however not less than 6 mm (see Note 13)
6.1.3		$L_2 = 1.5 \div 2.5$ mm $L_1 \geq \frac{1}{3} s_k$ however not less than 6 mm (see Note 14)
6.1.4		$L_1 \geq \frac{1}{3} s_k$ however not less than 6 mm (see Notes 15 and 16)
6.1.5		$L_1 = 10 \div 13$ mm (see Note 15)
6.2	**Welded penetrating branch pieces**	
6.2.1		Generally, used where $s_k \langle \frac{1}{2} s$ $e = s_k$

1	2	3
6.2.2		Generally, used where $s_k \cong \frac{1}{2}s$ $e = 6 \div 13$ mm $e + l = s_k$
6.2.3		Generally, used where $s_k > \frac{1}{2}s$ $e \geq \frac{1}{10}s$ 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		$l \geq \frac{1}{3}s_k$ however not less than 6 mm

1	2	3
6.4.2		$l \geq \frac{1}{3} s_k$ however not less than 6 mm $L_1 \geq 10$ mm
6.4.3		$e + l = s_k$ or s_{br} (whichever is lesser) $L_1 \geq 10$ mm
6.4.4		$e_2 + l \geq s_k$ $L_1 \geq 10$ mm $2\,s_k \leq (e_2 + l)$ plus $(s_{br} + e_1)$ or L_1 (whichever less of the latest)
6.5	**Pads and branch pieces with threaded holes**	
6.5.1		$d_2 \leq d_1 + 2\,s_{\min}$ (see Note 18)

1	2	3
6.5.2		$s \le 10$ mm (see Notes 19 and 20)
6.5.3		$L \ge 6$ mm $s \le 20$ mm
6.5.4		$s \ge 20$ mm
6.6	**Pads and branch pipes for screw joints**	
6.6.1		
6.6.2		
6.6.3		$d \le s$ $d_e = 2\,d$ $h \le 10$ mm $h \le 0.5\,s$ (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, made of steel, for which $R_m \leq 470$ MPa or $R_e \leq 370$ MPa.
2. The reduction of the thickness of the shell or of the flanged portion of the end plate may be effected in the inside or on the outside.
3. The joint used when welding can be done at either side of the shell.
4. The shells of more than 16 mm in thickness shall have the edges for fillet welds beveled in accordance with drawing 3.2.
5. The joints used when welding is possible at the outside of the shell only.
6. In the shells of no more than 16 mm in thickness the joints may be single-side welded. The breadth of the ring shall not be less than 40 mm.
7. The distance between the internal shell diameter and the external tube plate-diameter shall be as small as practicable.
8. The end of the tube projecting beyond the weld shall be milled or ground.
9. The spacing of the tubes shall not be less than 2.5 s_k and not less than 8 mm.
10. In the case of manual electrical welding, the dimension s_k shall not be less than 2.5 mm.
11. Recommendable when maximum reduction of the tube plate deformation occurring in the process of welding is necessary.
12. The tubes shall be welded manually by electric arc welding.
13. The backing ring shall fit tight and shall be removed after welding.
14. The joint used when welding can be done on the inside of the branch piece.
15. The joint used when the size of branch pieces is exceptionally small in respect to that of the vessel.
16. After welding, the branch piece shall be machined to the final size.
17. The ring shaped portions l shall permit the examination of the joints by X- ray radiography when necessary.
18. The distance between the ring pad and the shell shall not be greater than 3 mm.
19. The distance between the diameter of the opening in the shell and the external diameter of the ring shall be as small as practicable and shall not be greater than 3 mm.
20. The upper and lower bolt holes in the pad shall be shifted in respect to each other.
21. Combined thickness of the ship's shell and deposited welded material shall be sufficient for necessary number of thread turns.

SUPPLEMENT

RETROACTIVE REQUIREMENTS

1 GENERAL

1.1 The requirements specified in this *Supplement* apply to ships, depending on the stage of construction.

1.2 Execution of the applicable retroactive requirements shall be confirmed by PRS Surveyor in the report from the nearest ship survey.

2 REQUIREMENTS

2.1 Existing Passenger Ships of Class B

The requirements specified below shall be fulfilled in due time specified for each ship whose keel was laid or who was at the similar stage of construction:

.1 before 1 January 1940 – by 1 July 2006;
.2 1 January 1940 or later, but before 31 December 1962 – by 1 July 2007;
.3 1 January 1963 or later, but before 31 December 1974 – by 1 July 2008;
.4 1 January 1975 or later, but before 31 December 1984 – by 1 July 2009;
.5 1 January 1985 or later, but before 1 July 1998 – by 1 July 2010.

The ships shall fulfil the relevant requirements of the paragraphs specified in the following Chapters and sub-chapters:

2.2 Engine block: 2.2.4 to 2.2.10;
2.5 Fuel system: 2.5.2 to 2.5.4;
6.2 Steering gears and their installation onboard ship: 6.2.1.1, 6.2.1.3, and 6.2.1.5.

The ships shall fulfil the requirements of paragraphs 2.5.2 to 2.5.4 not later than 1 July, 2003, except that a suitable enclosure of engines having an output of 375 kW, or less, having fuel injection pumps serving more than one injector may be used as an alternative to the jacketed piping system in paragraph 2.5.2.

2.2 All Ships, Including Passenger Ships and Cargo Ships with Mechanical Propulsion of 500 Gross Tonnage and Upwards, Engaged on International Voyages

Application

.1 The requirements of paragraph 2 apply to ships constructed before 1 July 1998.
.2 The requirements of *SOLAS*, specified in paragraph 2 (*SOLAS* – regulations II-2/15.2.9 and 15.2.12) apply to all engines installed in any area on board ship, irrespective of service and location.

.3 The requirements of paragraph 2 apply to engines having single cylinder, multi-cylinder engines fitted with separate fuel pumps and those having multiple fuel injection pump units.

.4 The requirements of paragraph 2 do not apply to lifeboat engines.

Ships shall fulfil the relevant requirements of paragraph 2.5, *Part VII*:

2.5 Fuel system: paragraphs 2.5.2 to 2.5.4.

Ships shall fulfil the requirements of paragraphs 2.5.2 to 2.5.4, as well as with the requirements, specified in paragraphs 1 to 7 below, not later than 1 July 2003, except that a suitable enclosure on engines having an output of 375 kW or less fitted with fuel injection pumps serving more than one injector may be used as an alternative to the jacketed piping system.

.1 For engines of less than 375 kW power output where an enclosure is fitted, the enclosure shall have a similar function to jacketed pipes, i.e. prevent spray from a damaged injector pipe impinging on a hot surface.

.2 The enclosures shall completely surround the injection pipes, except that existing "cold" engine surfaces may be considered as part of the enclosures.

.3 All engine parts within the enclosures shall have a surface temperature not exceeding 220 °C when the engine is running at its maximum rating.

.4 The enclosure shall have sufficient strength and cover area to resist the effects of high pressure spray from a failed fuel pipe in service, prevent hot parts from being sprayed and restrict the area that can be reached by leaked fuel. Where the enclosure is not of metallic construction, it shall be made of non-combustible, non oil-absorbing material.

.5 Screening by the use of reinforced tapes is not acceptable as a suitable enclosure.

.6 Where leaked oil can reach hot surfaces, suitable drainage arrangements shall be fitted to enable rapid passage of leaked oil to a safe location which may be a drain tank. Leaked fuel flow onto "cold" engine surfaces can be accepted, provided that it is prevented from leaking onto hot surfaces by means of screens or other arrangements.

.7 Where the enclosure has penetrations to accommodate high pressure fittings, the penetrations shall be a close fit to prevent leakage.