Contents

RULES FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

Part D MACHINERY INSTALLATIONS

Chapter 1 GENERAL

1.1 General

1.1.1 Scope*

1 The requirements of this Part apply to the main propulsion machinery, power transmission systems, shafting systems, propellers, prime movers other than the main propulsion machinery, boilers and related equipment, incinerators, pressure vessels, auxiliaries, piping systems, and all of their respective control systems (hereinafter all of the above will be referred to as "machinery installations").

2 For machinery installations those which are fitted in either ships with a restricted area of service or ships categorized as "small ships", some of the requirements in this Part may be modified according to the requirements given in **Chapter 25** or other suitable rules deemed appropriate by the Society.

1.1.2 Equivalency

Machinery installations which do not comply with the requirements of this Part may be accepted provided that they are deemed by the Society to be equivalent to those specified in this Part.

1.1.3 Machinery Installations with Novel Design Features*

Machinery installations with novel design features may be accepted, provided the installation complies with any additional requirements on design and test procedures (other than those in this Part) as deemed necessary, with those results satisfactory to the Society.

1.1.4 Modification of Requirements*

For the following machinery installations, piping systems and all their respective control systems, some requirements of this Part may be modified appropriately provided that the Society considers such modifications acceptable:

(1) Small prime movers (including power transmission systems and shafting systems) for either driving generators or auxiliary machinery

(2) Auxiliary machineries for cargo handling and their prime movers (including power transmission systems and shafting systems)

(3) Machinery installations as deemed appropriate by the Society after considering their capacity, purpose and conditions of service

1.1.5 (Deleted)

(Deleted)

1.1.6 Terminology*

1 In this Part auxiliaries are classified into five groups. When auxiliaries have multiple uses and may be classified as belonging to more than one group, they are deemed to belong to the higher class. The five groups are given as follows with group **(1)** being the highest and group **(5)** being the lowest:

(1) Auxiliary machinery essential for main propulsion.
Defined as all auxiliary machinery that is used for the operation of the main propulsion machinery.

(2) Auxiliary machinery for manoeuvring and safety.
Defined as all auxiliary machinery that is used for ensuring safe manoeuvring, the safety of the ship as well as the safety of all persons on board.

(3) Auxiliary machinery for cargo handling.
Defined as all auxiliary machinery that is used for cargo loading and unloading as well as for cargo maintenance.

(4) Auxiliary machinery for specific use.

Defined as all auxiliary machinery that is used for a specific operation while either under way or at anchorage.

(5) Other auxiliary machinery.

Defined as any other auxiliary machinery that is not included in **(1)** to **(4)**.

2 Propulsion Shafting Systems

Defined as the thrust shaft, intermediate shaft, stern tube shaft, propeller shaft, their respective bearings as well as all propellers.

1.1.7 Drawings and Data to be Submitted

All the drawings and data that are to be submitted in connection with machinery installations are to conform to the requirements specified in each Chapter of this Part.

1.2 Materials

1.2.1 Selection of Materials

1 Materials under the requirements of **Part K**

All materials that are used for machinery installations are to be selected, according to the provisions of each Chapter of this Part, from those complying with the requirements specified in **Part K**. Special consideration is to be given with respect to the purpose and conditions of their service.

2 Other Materials

All materials that are used for machinery installations which are not specified in any Chapter of this Part are to conform to the following **(1)** and **(2)**:

(1) All materials used for the main propulsion machinery, power transmission systems, shafting systems, propellers, boilers, pressure vessels and control systems as well as those used for auxiliary machinery essential for main propulsion, manoeuvring and safety and cargo handling are to comply with Japanese Industrial Standards or any other standards deemed appropriate by the Society.

(2) All materials used for auxiliaries, excluding any auxiliary machinery in **(1)**, and their associated power transmission systems, shafting systems, piping systems and control systems are to be selected with consideration given to their purpose and conditions of service. Hereinafter these types of auxiliaries will be referred to as "auxiliary machinery for specific use etc."

1.3 General Requirements for Machinery Installations

1.3.1 General*

1 Machinery installations are to be of a design and construction adequate for the service for which they are intended and are to be so installed and protected as to reduce to a minimum any danger to persons on board, due regard being paid to moving parts, hot surfaces and other hazards.

The design is to have regard to the purpose for which the equipment is intended, the working conditions to which it will be subjected and the environmental conditions on board.

2 Special consideration is to be given to the reliability of any of the single essential machinery and components listed below.

In addition, for ships in which unconventional machinery is used as the main propulsion machinery and propulsion shafting system, additional machinery which enables the ship to proceed at a navigable speed in the event of possible failure of the machinery may be required by the Society.

(1) For ships in which reciprocating internal combustion engines are used as main propulsion machinery (excluding electric propulsion ships):

Reciprocating internal combustion engines, high elastic couplings, reduction gears and propulsion shafting systems

(2) For ships in which steam turbines are used as main propulsion machinery (excluding electric propulsion ships):

Steam turbine engines, main boilers, main condensers, reduction gears and propulsion shafting systems

(3) For ships in which gas turbines are used as main propulsion machinery (excluding electric propulsion ships):

Gas turbine engines, compressors, combustors, reduction gears and propulsion shafting systems

(4) For electric propulsion ships (only those specified in **5.1.1-1, Part H**, hereinafter the same in this Part):

Propulsion motors, reduction gears and propulsion shafting systems

3 For electric propulsion ships, two or more propulsion generators are to be provided.

4 Means are to be provided whereby normal operations of the main propulsion machinery can be sustained or restored even though one of the essential auxiliaries becomes inoperative. Special consideration is to be given to the malfunctioning of any of the following:

(1) A generating set which serves as a main source of electrical power;

(2) The sources of steam supply;

(3) The boiler feed water systems;

(4) The fuel oil supply systems for boilers or engines;

(5) The sources of lubricating oil pressure;

(6) The sources of water pressure;

(7) A condensate pump and the arrangements to maintain vacuum in condensers;

(8) The mechanical air supply for boilers;

(9) An air compressor and a receiver for starting or control purposes;

(10) The hydraulic, pneumatic or electrical means for control in main propulsion machinery including controllable pitch propellers.

However, having regard to overall safety consideration, a partial reduction in propulsion capability from normal operation may be accepted.

5 Means are to be provided to ensure that machinery installations can be brought into operation from the dead ship condition without external aids. In addition, the starting systems in conjunction with other machinery are to be so arranged as to restore propulsion from dead ship condition within 30 *minutes* after blackout.

6 Main propulsion machinery and prime movers for driving generators, and auxiliary machinery (excluding auxiliary machinery for specific use etc.) and their prime movers that are installed in the ships are to be designed to operate under the conditions given in **Table D1.1**. However, deviation from the angles given in **Table D1.1** may be permitted after taking into consideration the type, size and service conditions of the ship.

7 Machinery installations are to be designed to operate smoothly under the temperature conditions given in **Table D1.2**.

8 Provisions are to be made for the facilitation of the cleaning, the inspection and the maintenance of machinery installations.

9 Special consideration is to be given to the design, construction and installation of the machinery installations so that undue stresses caused by vibrations do not occur within normal operating ranges.

10 Waterjet propulsion systems or azimuth thrusters are to comply with the requirements of **Chapter 19** and **Chapter 20**, respectively.

11 The exhaust gas treatment systems specified in the following **(1)** and **(2)** fitted onto machinery installations are to comply with the requirements of **Chapter 21** and **Chapter 22**, respectively.

(1) Selective catalytic reduction (SCR) systems

(2) Exhaust gas cleaning systems (EGCS) (excluding those specified in **2.1.1-5**)

1.3.2 Astern Power

1 Sufficient power for going astern is to be provided to secure proper control of the ship in all normal circumstances.

2 The main propulsion machinery is to be capable of maintaining in free route astern at least at 70% of the ahead revolutions for a period of at least 30 *minutes*. The output astern which may be developed in transient conditions is to be such as to enable the braking of the ship within reasonable time.

3 For the main propulsion systems with reversing gears, controllable pitch propellers or electric propeller drive, running astern is not to lead to the overload of the propulsion machinery.

1.3.3 Limitation in the Use of Fuel Oil

Limitation in the use of fuel oil is to comply with the requirements in **4.2.1, Part R**.

Table D1.1 Angle of Inclination

Type of machinery installation	Athwartships(2)		Fore-and-aft (2)	
	Static inclination (List)	Dynamic inclination (Rolling)	Static inclination (Trim)	Dynamic inclination (Pitching)
Main propulsion machinery Main boilers and essential auxiliary boilers Prime movers driving generators (excluding those for emergency) Auxiliary machinery (excluding auxiliary machinery for specific use, etc.) and their driving units	15°	22.5°	5°(4)	7.5°
Emergency installation (emergency generators, emergency fire pumps and prime movers to drive them) Switchgears(1)(Circuit breakers, etc.) Automatic or remote operated equipment	22.5°(3)	22.5°(3)	10°	10°

Notes:

(1) No undesired switching operations or operational changes are to occur.

(2) Athwartships and fore-and-aft inclinations may occur simultaneously.

(3) In ships intended for the carriage of liquefied gases and of dangerous chemicals the emergency power supply is to also remain operable with the ship flooded to a final athwartships inclination up to maximum of 30°.

(4) Where the length of the ship exceeds 100 m, the fore-and-aft static angle of inclination may be taken as follows:

$\theta = 500/L$

θ : The static angle of inclination (°)

L : Length of the ship specified in **2.1.2, Part A** (m)

Table D1.2 Temperature

	Installed location	Temperature (℃)
Air	In enclosed spaces	0 to 45(Note)
	Machinery components or boilers in spaces subject to temperatures exceeding 45℃, and below 0℃	According to specific conditions
	On the open deck	-25 to 45(Note)
Seawater	-	32(Note)

Note:

Other temperatures deemed appropriate by the Society might be accepted in ships not intended for unrestricted service.

1.3.4 Fire Protections

1 Machinery installations are to be free from leakages of fuel oil, lubricating oil and other flammable oils. For those from which these oils may leak, proper means of leading the leaked oil to a safe location are to be provided.

2 Machinery installations are to be free from the leakage of any gases that may have a harmful effect on the health of the operator as well as any flammable gases. When fear of such gas leaks exists, machinery installations are to be installed in well-ventilated spaces that are capable of purging such gases quickly.

3 In addition to **1.3.4**, fire protections are to comply with the requirements in **4.2** and **5.2**, **Part R**.

1.3.5 Ventilating Systems for Machinery Spaces*

1 Machinery spaces of category *A* are to be adequately ventilated so as to ensure that when any of the machinery or boilers therein are operating at full power, that an adequate supply of air is maintained to the spaces for the safety and comfort of personnel and the operation of the machinery in all weather conditions. Any other machinery spaces other than those classified as category *A* are to be adequately ventilated in a manner that is appropriate for the purpose of that machinery space.

2 In cases where ventilation louvers with means for closure are fitted to emergency generator rooms and closing appliances are fitted to ventilators serving emergency generator rooms, such louvers or closing appliances are to comply with the requirements specified in the following **(1)** to **(4)**:

(1) Louvers and closing appliances may either be hand-operated or power-operated (hydraulic, pneumatic or electric) and are to be operable under fire conditions.

(2) Hand-operated louvers and closing appliances are to be kept open during normal operation of the vessel. In addition, corresponding instruction plates are to be provided at the location where hand-operation is provided.

(3) Power-operated louvers and closing appliances are to be of a fail-to-open type. However, closed power-operated louvers and closing appliances are acceptable during normal operation of the vessel. Power-operated louvers and closing appliances are to open automatically whenever the emergency generator is starting or in operation.

(4) Ventilation openings with means for closure are to be possible to close by a manual operation from a clearly marked safe position outside the space where the closing operation can be easily confirmed. In addition, the louver status (open or closed) is to be indicated at this position, and the closing of louvers and closing appliances is not to be possible from any remote position other than this position.

1.3.6 Machinery Spaces

Machinery spaces are to be sufficiently large enough to ensure the effective operation of any machinery installations installed in that machinery space.

1.3.7 Communication between the Navigating Bridge and Control Stations for the Speed and Direction of Thrust of the Propellers*

Communication between the navigating bridge and control stations for the speed and direction of thrust of the propellers are to comply with following requirements:

(1) At least two independent means are to be provided for communicating orders from the navigating bridge to the position in the machinery space or in the control room from which the speed and the direction of thrust of the propellers are normally controlled. One of these means is to be an engine-room telegraph which provides visual indication of any such orders and responses both on the navigating bridge and in such control stations mentioned above.

(2) Means of communication as deemed appropriate by the Society, are to be provided from the navigating bridge and the engine-room to any position, other than those specified in **(1)** above, from which either the speed or direction of thrust of the propellers may be controlled.

1.3.8 Engineers' Alarm

An engineers' alarm is to be installed at an appropriate location in either the engine control room or manoeuvring platform so that it can be operated properly. Such an alarm is to be clearly audible in the Engineers' accommodation.

1.3.9 Operating and Maintenance Instructions for Ship Machinery and Equipment

Operating and maintenance instructions as well as engineering drawings for all ship machinery and equipment essential to the safe operation of the ship are to be provided and written in a language understandable by her officers and crew members who are required to understand such information in the performance of their duties.

1.3.10 Rating Plate for *A.C.* Generating Sets

A.C. generating sets ("generating set" means a system which is composed of alternators, reciprocating internal combustion engines, couplings, etc.), except for those sets consisting of a propulsion engine which also drives power take-off (PTO) generator(s), are to be installed with rating plates marked with at least the following information:

(1) the generating sets manufacturer's name or mark;

(2) the set serial number;

(3) the set date of manufacture (month/year);

(4) the rated power (both in *kW* and *KVA*) with one of the prefixes *COP*, *PRP* (or, only for emergency generating sets, *LTP*) as defined in *ISO* 8528-1:2018, where the rated power is to be appropriate for the actual use of the generator set;

(5) the rated power factor;

(6) the set rated frequency (*Hz*);

(7) the set rated voltage (*V*);
(8) the set rated current (*A*); and
(9) the mass (*kg*).

1.4 Tests

1.4.1 Shop Tests

1 Before being installed on board, all equipment and components constituting a machinery installation (excluding auxiliary machinery for specific use etc.) are to be tested at facilities (hereinafter referred to as "Shop Tests") that have the proper equipment necessary to conduct such tests in accordance with the relevant requirements of this Part. To implement the tests, in lieu of traditional ordinary surveys where the Surveyor is in attendance, the Society may approve other survey methods which it considers to be appropriate.

2 For equipment and component parts of the machinery installations where shop tests are not specified in the requirements in any Chapter of this Part, and for those of auxiliary machinery for specific use etc., the records of the tests carried out by the manufacturer are to be submitted to the Society upon request.

1.4.2 Mass-production Equipment

For equipment manufactured by a mass-production system deemed appropriate by the Society, a test procedure suited to the production method may be accepted in place of the tests specified in the Rules upon the request of the manufacturer, notwithstanding the requirements of **1.4.1-1** above. Such requests are to be made by submitting the application form (**Form-5-1**).

1.4.3 Omission of Tests

Where machinery installations have test certificates which are deemed appropriate by the Society, a part of or all of the tests for the machinery specified in **1.4.1** may be omitted.

1.4.4 Tests after Installation On Board*

1 Machinery is to be tested after installed on board in accordance with the requirements specified in each Chapter of this Part.

2 Certain auxiliary machinery for specific use etc., as deemed necessary by the Society, are to be tested to the satisfaction of the Society at an appropriate time before being put into service in order to verify that they do not endanger either the ship or the crew during normal operation.

3 The Society may require other tests than those specified in this Part when deemed necessary.

Chapter 2 RECIPROCATING INTERNAL COMBUSTION ENGINES

2.1 General

2.1.1 General*

1 The requirements of this Chapter apply to reciprocating internal combustion engines which are used as the main propulsion machinery or used to drive generators and auxiliaries (hereinafter referred to in this Chapter as all auxiliaries excluding auxiliary machinery for specific use etc.).

2 For reciprocating internal combustion engines which are used for driving emergency generators, in addition to all of the requirements in this Chapter (excluding **2.2.4**, section **2.3**, **2.4.1-4** and the requirement for "devices to stop the operation of the engine" specified in **2.5.5-1**), the requirements of **18.5.2** (if controlled automatically or by remote) as well as those in **3.3** and **3.4, Part H** also apply.

3 For each type of reciprocating internal combustion engines, an approval of use is to be obtained by the engine designer (hereinafter referred to "licensor" in this Chapter) as specified separately by the Society.

4 The requirements of exhaust driven turbochargers specified in this chapter also apply, in principle, to engine driven chargers.

5 Reciprocating internal combustion engines fitted with exhaust gas recirculation (EGR) systems are to be in accordance with requirements specified in **Chapter 23** in addition to those in this Chapter.

6 Gas-fuelled engines to which **Chapter 16, Part N** applies are to be in accordance with **Annex 16.1.1-3, Part N** in addition to this chapter.

7 Gas-fuelled engines to which **Chapter 16, Part N** does not apply (**Part GF** applies instead) are to be in accordance with **Annex 1.1.3-3, Part GF** in addition to this chapter.

2.1.2 Terminology*

1 In this chapter, exhaust driven turbochargers are categorised into the following three groups according to the engine power at maximum continuous rating (*MCR*) supplied by a group of cylinders served by the actual turbocharger (*e.g.*, turbocharger size is to be 50% of total engine power for a V-engine with one turbocharger serving each bank of cylinders).

(1) Category *A* turbochargers

The engine power at *MCR* supplied by a group of cylinders served by the turbocharger is not more than 1000 *kW*.

(2) Category *B* turbochargers

The engine power at *MCR* supplied by a group of cylinders served by the turbocharger is not less than 1000 *kW,* but not more than 2500 *kW*.

(3) Category *C* turbochargers

The engine power at *MCR* supplied by a group of cylinders served by the turbocharger is not less than 2500 *kW*.

2 The terminology used in the application of **2.1.3** and **2.1.4** is as specified in the following **(1)** to **(36)**:

(1) "Acceptance criteria" mean a set of values or criteria which a design, product, service or process is required to conform with, in order to be considered in compliance.

(2) "Appraisal" means evaluation by a competent body.

(3) "Approval" means the granting of permission for a design, product, service or process to be used for a stated purpose under specific conditions based upon a satisfactory appraisal.

(4) "Assembly" means equipment or a system made up of components or parts.

(5) "Assess" means to determine the degree of conformity of a design, product, service, process, system or organization with identified specifications, rules, standards or other normative documents.

(6) "Certificate" means a formal document attesting to the compliance of a design, product, service or process with acceptance criteria.

(7) "Certification" means a procedure whereby a design, product, service or process is approved in accordance with acceptance criteria.

(8) "Competent body" means an organization recognized as having appropriate knowledge and expertise in a specific area.

(9) "Component" means a part, member of equipment or system.
(10) "Conformity" means that a design, product, process or service demonstrates compliance with its specific requirements.
(11) "Contract" means an agreement between two or more parties relating to the scope of service.
(12) "Customer" means a party who purchases or receives goods or services from another.
(13) "Design" means all relevant plans, documents, calculations described in the performance, installation and manufacturing of a product.
(14) "Design appraisal" means evaluation of all relevant plans, calculations and documents related to the design.
(15) "Equipment" means a part of a system assembled from components.
(16) "Equivalent" means an acceptable, no less effective alternative to specified criteria.
(17) "Evaluation" means systematic examination of the extent to which a design, product, service or process satisfies specific criteria.
(18) "Examination" means assessment by a competent person to determine compliance with requirements.
(19) "Inspection" means examination of a design, product service or process by a Surveyor.
(20) "Installation" means the assembling and final placement of components, equipment and subsystems to permit operation of the system.
(21) "Manufacturer" means a party responsible for the manufacturing and quality of the product.
(22) "Manufacturing process" means systematic series of actions directed towards manufacturing a product.
(23) "Material" means goods supplied by one manufacturer to another manufacturer that will require further forming or manufacturing before becoming a new product.
(24) "Modification" means a limited change that does not affect the current approval.
(25) "Product" means a result of the manufacturing process.
(26) "Quality assurance" means all the planned and systematic activities implemented within the quality system, and demonstrated as needed to provide adequate confidence that an entity will fulfil requirements for quality. Refer to *ISO* 9001:2015.
(27) "Regulation" means a rule or order issued by an executive authority or regulatory agency of a government and having the force of law.
(28) "Repair" means to restore to original or near original condition from the results of wear and tear or damages for a product or system in service.
(29) "Requirement" means specified characteristics used for evaluation purposes.
(30) "Information" means additional technical data or details supplementing the drawings requiring approval.
(31) "Specification" means technical data or particulars which are used to establish the suitability of materials, products, components or systems for their intended use.
(32) "Substantive modifications" mean design modifications, which lead to alterations in the stress levels, operational behaviour, fatigue life or an effect on other components or characteristics of importance such as emissions.
(33) "Sub-supplier/subcontractor" means one who contracts to supply material to another supplier.
(34) "Supplier" means one who contracts to furnish materials or design, products, service or components to a customer or user.
(35) "Test" means a technical operation that consists of the determination of one or more characteristics or performance of a given product, material, equipment, organism, physical phenomenon, process or service according to a specified procedure. A technical operation to determine if one or more characteristic(s) or performance of a product, process or service satisfies specific requirements.
(36) "Witness" means an individual physically present at a test and being able to record and give evidence about its outcome.

3 For electronically-controlled engines, the terminology is as specified in the following **(1)** to **(10)**:

(1) "Electronically-controlled engines" are engines whose fuel injection and/or Exhaust valve operation etc. are electronically controlled.
(2) "Accumulators" are small pressure vessels fitted to cylinders which provide hydraulic oil to those actuators attached to fuel injection devices or exhaust valve driving gears.
(3) "Common accumulators" are pressure vessels common to all cylinders for providing hydraulic oil or pressurized fuel oil.
(4) "Control valves" are components to control the delivery of hydraulic oil to drive actuators. The name control valve is generic for on-off-controlled solenoid valves, proportional-controlled valves or variable-controlled valves, etc.
(5) "Fuel oil pressure pumps" are pumps which provide pressurized fuel oil for common accumulators.

(6) "Hydraulic oil pressure pumps" are pumps to provide hydraulic oil for equipment, e.g. fuel injection devices, exhaust valve driving gears or control valves, through common accumulators.
(7) "Functional blocks" are blocks used to classify by function all items making up whole systems into the groups of systems, sub-systems, components, assemblies and parts.
(8) "Reliability block diagrams" are logical figures showing the relationship between functional blocks on an analytic level.
(9) "Normal operation" of main propulsion machinery means those operations at normal out-put conditions, using governors and all safety devices.
(10) "High-pressure" piping means piping in the down-stream of fuel oil pressure pumps or hydraulic oil pressure pumps.

4 For gas-fuelled engines, the terminology is in accordance with **1.4, Annex 1.1.3-3, Part GF**.

2.1.3 Drawings and Data*

1 Drawings and data to be submitted are generally as follows:
(1) Drawings and data for approval
Drawings and data specified in **Table D2.1(a)**
(2) Drawings and data for reference
Drawings and data specified in **Table D2.1(b)**

2 The drawings and data for the inspection and testing specified in **-1** (the items represented by the mark ○ in **Table D2.1(a)** and **Table D2.1(b)**, hereinafter indicated the same way throughout this Chapter) are to be submitted in accordance with **2.1.4-1** by the engine manufacturer producing engines with the drawings and data whose approval of use has been obtained in accordance with **2.1.1-3** (hereinafter referred to as "licensee" in this Chapter). Such drawings and data, however, may be submitted by the licensor in accordance with **2.1.4-2**.

Table D2.1(a) Drawings and Data for Approval

	Items	For inspection and testing
(1)	Engine particulars (in the format designated by the Society)	○
(2)	Material specifications of main parts with information on non-destructive testing and pressure testing as applicable to the material	○
(3)	Bedplate and crankcase of welded design, with welding details and welding instructions(1)	○
(4)	Thrust bearing bedplate of welded design, with welding details and welding instructions(1)	○
(5)	Frame/framebox/gearbox of welded design, with welding details and instructions(1)	○
(6)	Crankshaft, assembly and details	○
(7)	Thrust shaft or intermediate shaft (if integral with engine)	○
(8)	Shaft coupling bolts	○
(9)	Connecting rod bearings (four-stroke design)	—
(10)	Bolts and studs for connecting rods (four-stroke design)	○
(11)	Schematic layout or other equivalent drawings and data on the reciprocating internal combustion engine of the following **(a)** to **(g)** (details of the system so far as supplied by the licensee such as: main dimensions, operating media and maximum working pressures). (a) Starting air system (b) Fuel oil system (c) Lubricating oil system (d) Cooling water system (e) Hydraulic system (f) Hydraulic system (for valve lift) (g) Engine control and safety system	○
(12)	High pressure oil pipes for driving exhaust valves with its shielding	—
(13)	Shielding of high pressure fuel pipes, assembly (all engines)	○

(14)	High pressure parts for fuel oil injection system The documentation to contain specifications for pressures, pipe dimensions and materials.	○
(15)	Arrangement and details of the crankcase explosion relief valve (only for engines of a cylinder diameter of 200 *mm* or more or a crankcase volume of 0.6 m^3 or more)	○
(16)	Oil mist detection and/or alternative alarm arrangements	○
(17)	Connecting rod with cap (four-stroke design)	○
(18)	Arrangement of foundation (for main engines only)	○
(19)	The drawings, data, etc. required by **2.1.4**.	○
(20)	The following drawings and data for exhaust driven turbochargers: (a) Category *A* turbochargers (upon request) i) Sectional assembly (including principal dimensions and names of components) ii) Containment test report iii) Test procedures (b) Category *B* turbochargers i) Sectional assembly (including principal dimensions and materials of housing components for containment evaluation.) ii) Documentation of containment in the event of the disc fracture specified in **2.5.1-6** iii) Documentation of following operational data and limitations ・Maximum permissible operating speed (*rpm*) ・Maximum permissible exhaust gas temperature at the turbine inlet ・Minimum lubrication oil inlet pressure ・Maximum permissible vibration levels (self- and externally generated vibrations) ・Alarm level for exhaust gas temperature at the turbine inlet (levels are also to be indicated on engine control system diagrams) ・Lubrication oil inlet pressure low alarm set point (levels are also to be indicated on engine control system diagrams) ・Lubrication oil outlet temperature high alarm set point (levels are also to be indicated on engine control system diagrams) iv) Diagram of lubrication oil systems (diagrams included in piping arrangements fitted to engines may be accepted instead) v) Test report of type test (only for type tests) vi) Test procedure (only for type tests) (c) Category *C* turbochargers i) Drawings listed in **(b)** above ii) Drawings of the housing and rotating parts (including details of blade fixing) iii) Material specifications (including mechanical properties and chemical composition) of the parts mentioned in **ii)** above iv) Welding details and welding procedures for the parts mentioned in **ii)** above, if made of welded construction	—
(21)	Other drawings and data deemed necessary by the Society	○

Notes:

(1) For approval of materials and weld procedure specifications, the weld procedure specification is to include details of pre- and post-weld heat treatments, weld consumables and fit-up conditions.

Table D2.1(b) Drawings and Data for Reference

	Items	For inspection and testing
(1)	A list containing all drawings and data submitted (including relevant drawing numbers and revision status)	○
(2)	Bolts and studs for main bearings	○
(3)	Connecting rod bearings (two-stroke design)	–
(4)	Bolts and studs for cylinder heads and exhaust valve (two-stroke design)	○
(5)	Bolts and studs for connecting rods (two-stroke design)	○
(6)	Tie rods	○
(7)	Piston pins	–
(8)	Construction of accumulators for hydraulic oil and fuel oil	○
(9)	Cylinder head fixing bolts and valve box fixing bolts	–
(10)	Rocker valve gears	–
(11)	Cylinder head	○
(12)	Cylinder block, engine block	○
(13)	Cylinder liner	○
(14)	Counterweights (if not integral with crankshaft), including fastening	○
(15)	Connecting rod with cap (two-stroke design)	○
(16)	Crosshead	○
(17)	Piston rod	○
(18)	Piston, assembly, including identification (e.g. drawing number) of components	○
(19)	Piston head	○
(20)	Camshaft drive, assembly, including identification (e.g. drawing number) of components	○
(21)	Flywheel	○
(22)	Fuel oil injection pump	○
(23)	Shielding and insulation of exhaust pipes and other parts of high temperature which may be impinged as a result of a fuel system failure, assembly	○
(24)	Construction and arrangement of dampers	○
(25)	Construction and arrangement of detuners, balancers or compensators, bracings as well as all calculation sheets related to engine balancing and engine vibration prevention	–
(26)	For electronically controlled engines, assembly drawings or arrangements of the following **(a)** to **(d)**: (a) Control valves (b) High-pressure pumps (c) Drive for high pressure pumps (d) Valve bodies, if applicable	○
(27)	Operation and service manuals(1)	○
(28)	Engine control system diagram (including the monitoring, safety and alarm systems)	–
(29)	Test program resulting from FMEA (for engine control system) in cases of engines that rely on hydraulic, pneumatic or electronic control of fuel injection and/or valves	○
(30)	Production specifications for castings and welding (sequence)	○
(31)	Certification of an approval of use for environmental tests, control components(2)	○
(32)	Quality requirements for engine production	○
(33)	Location of measures preventing oil from spraying out from joints in flammable oil piping (if fitted)	–
(34)	The following drawings and data for exhaust driven turbochargers (only for category *C* turbochargers): (a) Documentation of the safe torque transmission specified in **2.5.1-6** when the disc is connected to the	–

	shaft by an interference fit (b) Information on expected lifespan (creep, low cycle fatigue and high cycle fatigue are to be considered) (c) Operation and maintenance manuals	
(35)	Other drawings and data deemed necessary by the Society	○

Notes:

(1) Operation and service manuals are to contain maintenance requirements (servicing and repair) including details of any special tools and gauges that are to be used with their fitting/settings together with any test requirements on completion of maintenance.

(2) Drawings and data modified for a specific application are to be submitted to the Society for reference or approval, as applicable

2.1.4 Approval of Reciprocating Internal Combustion Engines

1 Reciprocating internal combustion engines are to be approved in accordance with the following **(1)** to **(6)**:

(1) Development of documents and data for engine production

(a) Prior to the start of the reciprocating internal combustion engine approval process in accordance with the following **(c)** and subsequent sub-paragraphs of this paragraph, a design approval is to be obtained as specified separately by the Society.

(b) Each type of reciprocating internal combustion engine is to be provided with a certificate of approval of use obtained by the licensor in accordance with **2.1.1-3**. For the first engine of a type or for those with no service records, the process of an approval of use and the approval process for production by the licensee may be performed simultaneously.

(c) The licensor is to review the drawings and data of the reciprocating internal combustion engine whose approval of use has been obtained for the application and develop, if necessary, application specific drawings and data for production of reciprocating internal combustion engines for the use of the licensee in developing the reciprocating internal combustion engine specific production drawings and data for the inspection and testing specified in **2.1.3-1**.

(d) If substantive modifications to the drawings and data of the reciprocating internal combustion engine whose approval of use has been obtained have been made in the drawings and data of reciprocating internal combustion engines to be produced, the affected drawings and data are to be resubmitted to the Society as specified separately by the Society.

(2) Drawings and data for the inspection and testing of reciprocating internal combustion engines

(a) The licensee is to develop the drawings and data for the inspection and testing specified in **2.1.3-1** and a comparison list of these drawings and data to the drawings and data of the reciprocating internal combustion engine whose approval of use has been obtained by the licensor and submit these drawings and the comparison list to the Society.

(b) As for the drawings and data for the inspection and testing specified in **2.1.3-1**, if there are differences in the technical content on the licensee's production drawings and data of the reciprocating internal combustion engine compared to the drawings and data of the reciprocating internal combustion engine whose approval of use has been obtained by the licensor, the licensee is to submit "Confirmation of the licensor's acceptance of licensee's modifications" approved by the licensor and signed by the licensee and licensor. If the licensor acceptance is not confirmed, the reciprocating internal combustion engine manufactured by the licensee is to be regarded as a different engine type and is **2.1.1-3** is to apply to the reciprocating internal combustion engine.

(c) In applying **(b)** above, modifications applied by the licensee are to be provided with appropriate quality requirements.

(d) The Society returns the drawings and data specified in **(a)** and **(b)** above to the licensee with confirmation that the design has been approved.

(e) The licensee or its subcontractors are to prepare to be able to provide the drawings and data specified in **(a)** and **(b)** above so that the Surveyor can use the information for inspection purposes during manufacture and testing of the reciprocating internal combustion engine and its components.

(3) Additional drawings and data

In addition to the drawings and data for the inspection and testing specified in **2.1.3-1**, the licensee is to be able to provide to the Surveyor performing the test specified in **2.6.1** upon request the relevant detail drawings, production quality control specifications and acceptance criteria. These drawings and data are for supplemental purposes to the survey only.

(4) Licensee approval

(a) The Society assesses conformity of production with the Society's requirements for production facilities comprising manufacturing facilities and processes, machining tools, quality assurance, testing facilities, etc. as specified separately by the Society.

(b) Satisfactory conformance with **(a)** above results in the issue of a document showing the licensee has been approved by the Society.

(5) Engine assembly and testing

The licensee is to assemble and test the reciprocating internal combustion engine according to the Society's technical rules each of the reciprocating internal combustion engine assembly and testing procedure is to be witnessed by the Surveyor unless the manufacturer of the reciprocating internal combustion engine is one approved in accordance with the **Rules for Approval of Manufacturers and Service Suppliers** and use of a mass production system is agreed between the manufacturer and the Society.

(6) Issue of certificates of reciprocating internal combustion engines and components

(a) The attending Surveyors, at the licensee/subcontractors, will issue product certificates as necessary for components manufactured upon satisfactory inspections and tests.

(b) An engine certificate is issued by the Surveyor upon satisfactory completion of assembly and tests specified in **(5)** above.

2 In applying **-1** above, for those cases when a licensor - licensee agreement does not apply, a "licensor" is to be understood as the following **(1)** or **(2)**:

(1) The entity that has the design rights for the reciprocating internal combustion engine type; or

(2) The entity that is delegated by the entity having the design rights of **(1)** above to modify the design.

3 Components of licensor's design which are covered by the certificate of approval of use of the relevant engine type are regarded as approved whether manufactured by the reciprocating internal combustion engine manufacturer or sub-supplied.

4 For components of subcontractor's design, necessary approvals are to be obtained by the relevant suppliers (*e.g.* exhaust gas turbochargers, charge air coolers, etc.).

2.2 Materials, Construction and Strength

2.2.1 Materials

1 Materials intended for the principal components of reciprocating internal combustion engines and their non-destructive tests as well as surface inspections and dimension inspections are to conform to the requirements given in **Table D2.2**. However, with respect to ultrasonic testing as well as surface inspections and dimension inspections, submission or presentation of test results to the Surveyor may be considered sufficient. In cases where deemed necessary by the Society, tests or inspections may also be required for any parts not specified in **Table D2.2**.

2 Cylinders, pistons and other parts subjected to high temperature or pressure as well as any parts used for transmitting propulsion torque are to be of materials suitable to sufficiently withstand high temperature and loads.

2.2.2 Construction, Installation and General*

1 Cylinders, pistons and other parts subjected to high temperature or pressure are to be of a construction that is suitable to sufficiently withstand the mechanical and thermal stresses.

2 Where the principal components of a reciprocating internal combustion engine are of welded construction, they are to comply with the requirements of **Chapter 11**.

3 The frames and bedplates are to be of rigid and oiltight construction. The bedplate is to be provided with a sufficient number of foundation bolts to secure it firmly to the entire length of the engine seating.

4 Crankcase and crankcase doors are to be of sufficient strength to withstand a crankcase explosion. Crankcase doors are to be fastened sufficiently securely for them not to be readily displaced by a crankcase explosion.

5 A warning notice is to be fitted on a prominent position, preferably on a crankcase door on each side of the engine, or alternatively at the engine room control station. This warning notice is to specify that whenever overheating is suspected within the crankcase, the crankcase doors or sight holes are not to be opened until a reasonable time, sufficient to permit adequate cooling has elapsed after stopping the engine.

6 Ventilation of crankcase, and any arrangement which could produce a flow of external air into the crankcase, is not permitted except in cases **(1)** to **(3)** below.

(1) Ventilation pipes, where provided, are to be as small as practicable to minimise the inrush of air after a crankcase explosion. In addition, ventilation pipes for each engine are to be independent of any other engine. Ventilation pipes from the crankcase of

main propulsion engine are to lead to a safe position on deck or to some other approved position.

(2) If provision is made for the extraction of gases from the crankcase (*e.g.* for oil mist detection purposes), the vacuum in the crankcase is not to exceed 2.5×10^{-4} *MPa*.

(3) In cases where gas-fuelled engines are provided with crankcase ventilation for preventing the accumulation of leaked gas.

7 The ambient reference conditions for the purpose of determining the power of reciprocating internal combustion engines are to be as follows:

Total barometric pressure: 0.1 *MPa*

Air temperature: 45 °C

Relative humidity: 60 %

Seawater temperature (charge air intercooler-inlet): 32 °C

8 Essential components are to be so arranged that normal operation of main propulsion machinery is capable of being sustained or restored even though one of these components becomes inoperable, except in cases where special consideration and approval is given by the Society to the reliability of single arrangements. Single components provided for cylinders, which do not require a spare, may be acceptable in cases where any failed parts can be isolated.

Table D2.2 Application of Materials and Non-destructive Tests as well as Surface Inspections and Dimension Inspections to Principal Components of Reciprocating Internal Combustion Engines

Principal components			Cylinder bore *D* (*mm*)								
			$D\le300$			$300<D\le400$			$400<D$		
			I	II	III	I	II	III	I	II	III
1	Welded bedplate		○	○		○	○		○	○	
2	Bearing transverse girders (cast steel)		○	○		○	○		○	○	
3	Welded frame box		○	○		○	○		○	○	
4	Welded cylinder frames [5]		○	○		○	○		○	○	
5	Engine block (spheroidal graphite cast iron)[6]		○			○			○		
6	Cylinder liner					○[7]			○[7]		
7	Cylinder head (cast steel or forged steel)					○	○		○	○	
8	Piston crown (cast steel or forged steel)								○	○	
9	Crankshaft	made in one piece	○	○	○	○	○	○	○	○	○
		Web, pin and journal for all built-up and semi-built-up types	○	○	○	○	○	○	○	○	○
		Others (including coupling bolts)	○	○	○	○	○	○	○	○	○
10	Piston rod [5]								○	○	
11	Cross head [5]		○	○		○	○		○	○	
12	Connecting rods together with connecting rod bearing caps		○	○	○	○	○	○	○	○	○
13	Bolts and studs (for cylinder heads, connecting rods, main bearings)					○	○	*TR*[8]	○	○	*TR*[8]
14	Tie rod [5]		○	○	*TR*[8]	○	○	*TR*[8]	○	○	*TR*[8]
15	Fuel injection pump body		○[9]			○[9]			○[9]		
16	High pressure fuel injection pipes including common fuel rail		○			○			○		
17	High pressure common servo oil system		○			○			○		
18	Heat exchanger, both sides [10]					△			△		
19	Accumulator [11]		○			○			○		
20	Piping, pumps, actuators, etc. for hydraulic drive of valves [12]		○[13]			○[13]			○[13]		

21	Pipes, valves and fittings attached to engines classified in **Chapter 12** as either Group I or Group II. (excluding items listed in this table)		○			○			○		
22	Bearings for main, crosshead, and crankpin [12]		*TR*[14]	*TR*[15]	○	*TR*[14]	*TR*[15]	○	*TR*[14]	*TR*[15]	○
23	Turbine discs, blades, blower impellers and rotor shafts of exhaust driven turbochargers [16]	Category *A*	○[9]			○[9]			○[9]		
		Category *B*	○	○	○[17]	○	○	○[17]	○	○	○[17]
		Category *C*	○	○	○[17]	○	○	○[17]	○	○	○[17]
24	Casings of exhaust driven turbochargers [16][18]	Category *A*	○[9]			○[9]			○[9]		
		Category *B*	○			○			○		
		Category *C*	○			○			○		

Notes:

(1) Materials intended for the components marked by "○" or "*TR*" in Column I are to comply with the requirements in **Part K**. However, the components marked by "*TR*" in Column I may be in accordance with **Note (9)**. In addition, materials intended for the components marked by "△" in Column I are to comply with the requirements in **Chapter 10**.

(2) Materials intended for the components marked by "○" or "*TR*" in Column II are to be tested by a magnetic particle test or a liquid penetrant test as well as an ultrasonic test. When a certain non-destructive test method on the finished component is impractical (for example UT for cylinder heads), the non-destructive test method may be performed at an appropriate earlier stage of production for the component.

(3) Materials intended for the components marked by "○" or "*TR*" in Column III are to be tested by a surface inspection and a dimension inspection.

(4) For items marked by "*TR*", submission of a test report which compiles all test and inspection results in an acceptance protocol issued by the manufacturer may be accepted. The test report is to include the following. Tests or inspections may be carried out on samples from the current production.

(a) Signature of the manufacturer

(b) Statement that components comply with specifications stipulated by the manufacturer

(5) Only for crosshead reciprocating internal combustion engines.

(6) Only when engine power exceeds 400 *kW/cyl*. Chemical composition analysis may be omitted.

(7) Materials may be in accordance with **Note (9)** except when used for steel parts.

(8) Only for threaded bolts and studs used for connecting rods or tie rods.

(9) Materials which comply with the requirements of national or international standards such as *ISO*, *JIS*, etc. may be used.

(10) Charge air coolers need only be tested on the water side.

(11) Only when capacity exceeds 0.5 *l*.

(12) Only when engine power exceeds 800 *kW/cyl*.

(13) Materials intended for pumps and actuators may be in accordance with **Note (9)**.

(14) Mechanical property test may be omitted.

(15) Magnetic particle tests and liquid penetrant tests may be omitted. An ultrasonic test is to be carried out for full contact between the base material and bearing metal

(16) In cases where the manufacturer has a quality system deemed appropriate by the Society, materials and non-destructive tests as well as a surface inspection and a dimension inspection for categories *A* and *B* turbochargers may not require the presence of a Society surveyor. In such cases, the submission or presentation of test records may be required by the Society.

(17) Surface inspection may be omitted.

(18) Chemical composition analysis may be omitted.

2.2.3 Crankpin Bearings of 4-Stroke Cycle Engines

The crankpin bearings of 4-stroke cycle engines are to be designed and constructed so as to keep a fair contact pressure upon the contact face of the bearing caps as well as not to cause any excessive stress on the crankpin bolts against the alternating load acting on the connecting rod.

2.2.4 Flywheel Shafts and Other Shafts

Where flywheels or eccentric sheaves for pumps are fitted onto crankshafts or additional shaft located between the aftermost journal bearing and the thrust shaft, the shaft diameter in way of the part is not to be less than the required diameter of the crankshaft determined by the formula in **2.3**.

2.3 Crankshafts

2.3.1 Solid Crankshafts and Semi-Built Crankshafts*

1 The requirements in this paragraph apply to solid-forged and semi-built crankshafts made of forged or cast steel, with one crank throw between main bearings that are used for reciprocating internal combustion engines for propulsion and auxiliary purposes in cases where such engines are capable of continuous operation at their rated power when running at their rated speed.

2 The torsional stress in crankpins and journals is to be evaluated by carrying out forced vibration calculations including the stern shafting and the values of the acceptability factor Q calculated by **Annex 2.3.1** "**CALCULATION METHOD OF CRANKSHAFT STRESS**" are to comply with the following formula:

$$Q \geq 1.15$$

3 In cases where a crankshaft design involves the use of surface treated fillets, where fatigue parameter influences are tested, or where working stresses are measured, relevant documents for such calculations and analyses are to be submitted to the Society in order to demonstrate equivalence to **-2** above.

4 Approval of crankshafts other than those specified in the requirements of this paragraph is to be as deemed appropriate by the Society.

2.3.2 Built-up Crankshafts*

Built-up crankshaft approval is to be as deemed appropriate by the Society.

2.3.3 Shaft Couplings and Coupling Bolts*

1 The diameter of coupling bolts at the joining face of the coupling between crankshafts, between a crankshaft and a thrust shaft, or between a crankshaft and a shaft mentioned in **2.2.4** is to be not less than the value obtained by the following formula.

$$d_b = 0.75\sqrt{\frac{(0.95d_c)^3}{nD}\left(\frac{440}{T_b}\right)}$$

where

d_b : Diameter of coupling bolts (mm)

n : Number of bolts

D : Diameter of pitch circle (mm)

d_c : Required diameter of crankshaft (mm), as deemed appropriate by the Society.

T_b : Specified tensile strength of bolt material (N/mm^2)

When the specified tensile strength of the bolt material exceeds 1000 N/mm^2, the value used for the formula is to be as considered appropriate by the Society.

2 Shaft couplings are to be of sufficient strength to withstand working stresses. The fillets of shaft couplings are to have enough of a radius to avoid any excessive stress concentration. Where shaft couplings are separate from the shafts, both the fitting method and the construction of the couplings are to be sufficient enough to resist astern pull. In cases where keys are used for fitting shaft couplings to shafts, the grooves for the keys are to be constructed so as to avoid any excessive stress concentration.

2.4 Safety Devices

2.4.1 Speed Governors and Overspeed Protective Devices

1 For ship in which reciprocating internal combustion engines are used as main propulsion machinery (excluding electric propulsion ships), each of such reciprocating internal combustion engines is to be provided with a speed governor so adjusted to prevent the engine speed from exceeding the number of maximum continuous revolutions by more than 15 %.

2 In addition to this speed governor, each reciprocating internal combustion engine as specified in **-1** above that has a continuous maximum output of 220 *kW* or above, and which can be declutched or which drives a controllable pitch propeller, is to be provided with a separate overspeed protective device. The overspeed protective device, including its driving gear, are to be independent from the governor required by **-1**, and be so adjusted that the engine speed may not exceed the number of maximum continuous revolutions by more than 20 %.

3 Reciprocating internal combustion engines used to drive generators are to be provided with the governors specified in the requirements in **-5**. However, if a reciprocating internal combustion engine which is used as main propulsion machinery of an electric propulsion ship drives a generator used to supply electrical power exclusively to propulsion motors, the requirements specified in **5.1.2-2, Part H** are to be applied.

4 In addition to the speed governor, each reciprocating internal combustion engine used as main propulsion machinery of electric propulsion ships and those reciprocating internal combustion engines used to drive generators that have a maximum continuous output of 220 *kW* or above are to be provided with a separate overspeed protective device. The overspeed protective device, including its driving gear, are to be independent from the governor required by **-3**, and be so adjusted that the engine speed may not exceed the number of maximum continuous revolutions by more than 15 %.

5 Speed governors of reciprocating internal combustion engines driving generators are to be have the following characteristics:

(1) Reciprocating internal combustion engines driving main generators

(a) Momentary speed variations are, in principle, to be 10 % or less of the maximum rated speed when the rated loads of generators are suddenly thrown off. However, in cases where it is difficult to meet the above requirements, the characteristics of such governors may be acceptable in the following cases.

i) In cases where momentary variations are 10 % or less of the rated speed when the maximum load on board is suddenly thrown off and the speed is returned to within 1 % of the final steady speed in not more than 5 *seconds*, momentary variations in excess of 10 % of rated speeds may be acceptable in cases where rated loads of such generators are suddenly thrown off.

ii) The momentary variations given in **i)** above, in cases where the rated loads of generators suddenly thrown off are less than any adjusted values of the intervention of overspeed devices as required by **-4**.

(b) Momentary speed variations are, in principle, to be 10 % or less of the maximum rated speed when 50 % of the rated loads of generators are suddenly thrown-on followed by the remaining 50 % of such loads suddenly being thrown-on after an interval to restore the steady state. Speeds are to return to within 1 % of final steady speeds in not more than 5 *seconds*.

(c) In cases where the throwing-on methods are difficult according to the requirements in **(b)** above, and where a three-stage or more throwing-on method is adopted, throw-on power calculation sheets which take into consideration **i)** to **iv)** are to be submitted to the Society for approval:

i) power restoration after blackout,

ii) sequential starting,

iii) starting with large start-up loads, or

iv) instantaneous load transfers in cases where one set of generators fails (during parallel running).

(d) At all loads in ranges between no loads and rated loads, all permanent speed variations are to be within ± 5 % of the maximum rated speed.

(2) Reciprocating internal combustion engines driving emergency generators

(a) Momentary speed variations are not to exceed the values specified in **(1)(a)** in cases where total emergency consumer loads are suddenly thrown off.

(b) Momentary speed variations are, in principle, not to exceed the values specified in **(1)(b)** and speeds to return to within 1 % of final steady speeds in not more than 5 *seconds* in cases where total emergency consumer loads are suddenly thrown-on. However, if it is difficult to meet the above requirements and in cases where the following **i)** through **iii)** requirements are adopted, a throwing-on in steps method may be used.

i) Total emergency consumer loads are to be thrown-on within 45 *seconds* after blackout.

ii) Prime movers are to be designed so that the maximum step loads in emergency consumer loads are to be thrown-on at one time.

iii) Documents such as thrown-on power calculations specifying the adoption of throwing-on in steps are to be submitted.

(c) At all loads in ranges between no loads and total emergency consumer loads, all permanent speed variations are not to exceed the values specified in **(1)(d)** above.

(3) Reciprocating internal combustion engines driving *a.c.* generators operating in parallel

(a) The load sharing specified in **2.4.14-4** and **-5, Part H**, is ensured, and

(b) Facilities are to be provided to adjust the governor sufficiently enough to permit adjustments of loads not exceeding 5 % of rated loads at normal frequencies.

2.4.2 Alarm for Overpressure in the Cylinders

Each cylinder of reciprocating internal combustion engines having a bore exceeding 230 *mm* is to be provided with an effective sentinel valve or other means for overpressure.

2.4.3 Protection against Crankcase Explosion*

1 Reciprocating internal combustion engines having a cylinder bore not less than 200 *mm* or a crankcase with a gross volume not less than 0.6 m^3 are to be provided with crankcase explosion relief valves of an approved type for preventing any overpressure in the event of an explosion within the crankcase. Crankcase explosion relief valves are to be in accordance with the following requirements:

(1) The valves are to 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.

(2) The valve discs are to be made of ductile material capable of withstanding the shock of contact with stoppers at the full open position.

(3) The valves are to be designed and constructed to open quickly and be fully open at a pressure not greater than 0.02 *MPa*.

(4) The valves are to be provided with a flame arrester that permits flow for crankcase pressure relief and prevents passage of flame following a crankcase explosion.

(5) The valves are to be provided with a copy of the manufacturer's installation and maintenance manual. This copy is to be provided on board ship.

2 The number and locations of the explosion relief valves specified in **-1** are to be in accordance with **Table D2.3**.

3 Additional explosion relief valves corresponding to **-1** above are to be fitted on separate spaces of crankcase such as gear or chain cases for camshaft or similar drives, when the gross volume of such spaces is not less than 0.6 m^3.

4 The explosion relief valves given in **-1** and **-3** above are to conform to the following requirements **(1)** to **(3)**:

(1) The free area of each explosion relief valve is to be not less than 45 cm^2.

(2) The combined free area of the valves fitted on an engine is to be not less than 115 cm^2 per cubic metre of the crankcase or similar drive case specified in **-3** gross volume. The total volume of the stationary parts within the crankcase or separate space may be discounted in estimating the gross volume of the case.

Table D2.3 Number and Location of Explosion Relief Valves

Cylinder bore(*mm*)	Number and location of explosion relief valves
200 to 250	At least one valve near each end, but, an additional valve is to be fitted near the middle of the engine in the case of more than 8 crankthrows.
over 250 to 300	At least one valve in way of each alternate crankthrow, with a minimum of two valves.
over 300	At least one valve in way of each crankthrow.

2.4.4 Protection against Scavenging Spaces

1 Scavenging spaces in open connection to the cylinders are to be provided with relief valves designed to prevent explosions that might be caused by the abnormal buildup of pressure. These devices are to be arranged so that any discharge from them does not pose any danger to the operators working in that space.

2 Scavenging spaces in open connection to the cylinders are to be connected to an approved fire extinguishing system completely independent of the system in the engine room.

2.4.5 Crankcase Oil Mist Detection Arrangements*

1 Crankcase oil mist detection arrangements are required for reciprocating internal combustion engines of 2,250 *kW* maximum continuous power and above or having cylinders of more than 300 *mm* bore, and in cases of engine failure, the following means are to automatically be employed. However, in cases where alternative devices deemed appropriate by the Society are provided, such devices may be used instead of crankcase oil mist detection arrangements. In this case, the following means are also to be automatically employed.

(1) In the case of low speed engines (a rated speed of less than 300 *rpm*), alarms are to activate and speeds be reduced. (However, in cases where alternative measures such as activating alarms to request such speed reductions are taken, the manual reduction of speeds may be accepted).

(2) In the case of medium speed engines (a rated speed of 300 *rpm* and above, but less than 1,400 *rpm*) and high speed engines (a rated speed of 1,400 *rpm* and above), alarms are to activate and engines are to be stopped or have their fuel supply shut off.

2 The crankcase oil mist detection arrangements required in **-1** above are to be of an approved type and in accordance with the following requirements:

(1) Oil mist detection arrangements are to provide an alarm indication in the event of a foreseeable functional failure in the equipment and installation arrangements.

(2) Oil mist detection arrangements are to provide an indication 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.

(3) Oil mist detection arrangements are to be capable of being tested on the test bed and board under engine standstill and engine running at normal operating conditions.

(4) Each engine is to be provided with independent oil mist detection and monitoring and a dedicated alarm. Oil mist detection and alarm information is to be able to be confirmed from a safe location away from the engine. In addition, in the case of ships which apply the **Rules for Automatic and Remote Control Systems**, the concentration of crankcase oil mist is also to be capable of being read by a monitoring panel.

(5) The layout of the arrangements, pipes and cables, pipe dimensions, the location of engine crankcase sample points, sample extraction rate and the way of maintenance and test are to be in accordance with the engine designer's and oil mist manufacturer's instructions.

(6) Where sequential oil mist detection arrangements are provided the sampling frequency and sampling time is to be as short as reasonably practicable.

(7) A copy of the maintenance and test manual is to be provided on board ship.

2.5 Associated Installations

2.5.1 Exhaust Driven Turbochargers*

1 Manufacturers are to adhere to a quality system designed to ensure that designer specifications are met, and that manufacturing is in accordance with the approved drawings.

2 For main propulsion engine equipped with exhaust driven turbochargers, means are to be provided to ensure that the engine can be operated with sufficient power to give the ship a navigable speed in case of failure of one of the turbochargers.

3 Where the main propulsion engine can not be operable only with the exhaust driven turbochargers in case of starting or low speed range, an auxiliary of scavenging air system is to be provided. For the event of failure of such an auxiliary system, proper means are to be provided so that the main propulsion engine can be brought into the condition that its output increases enough as the exhaust driven turbochargers show their function.

4 Exhaust driven turbochargers are to be designed to operate under the conditions given in **1.3.1-6** and **2.2.2-7**. Component lifetime and the alarm level for speed are to be based upon an air inlet temperature of 45° C.

5 The air inlets of exhaust driven turbochargers are to be fitted with filters.

6 Exhaust driven turbochargers are to be capable of containment in the event of a rotor burst. This means that no parts are to penetrate the casing of exhaust driven turbochargers or escape through the air intake in the case of a rotor burst. It is to be assumed that the discs disintegrate in the worst possible way.

7 In the case of category *C* turbochargers where the disc is connected to the shaft by an interference fit, calculations are to substantiate safe torque transmission during all relevant operating conditions such as maximum speed, maximum torque and maximum temperature gradient combined with minimum shrinkage amount.

8 For categories *B* and *C* turbochargers, the indications and alarms listed in the **Table D2.4** are to be provided. Indications may be provided at local locations, monitoring stations or control stations. Alarm levels may be equal to permissible limits, but are not to be reached when operating the engine at 110 % power, or at any approved intermittent overload beyond 110 % in cases where the turbochargers are fitted to engines for which intermittent overload power is approved.

9 Turbochargers are to have compressor characteristics that allow the engines, for which they are intended, to operate without any audible high pitch vibrations or explosion-like noises from the scavenger area of the engine (hereinafter referred to as "surging" in this Part) during all operating conditions and also after extended periods of operation. For abnormal, but permissible, operation conditions such as misfiring and sudden load reduction, repeated surging (hereinafter referred to as "continuous surging") is not to occur.

10 Certificates for categories *B* and *C* turbochargers issued by the Society will, at a minimum, cite the applicable type approval.

11 Certification and test requirements specified in this chapter apply to the replacement of rotating parts and casings.

Table D2.4 Alarms and Indications of Turbochargers

Monitoring Item	Category *B* Turbochargers		Category *C* Turbochargers		Remarks
	Alarm	Indication	Alarm	Indication	
Speed	H(1)	○(1)	H(1)	○(1)	Alarm level is to be based upon an air inlet temperature of 45 °C.
Exhaust gas temperature at each turbocharger inlet	H(2)	○(2)	H	○	High temperature alarms for each cylinder at engine are acceptable.(3)
Lubrication oil temperature at turbocharger outlet	—	—	H	○	If not forced lubrication system, oil temperature near bearings is to be monitored.
Lubrication oil pressure at turbocharger inlet	L	○	L	○	Only for forced lubrication systems.(4)

Notes:

(1) For turbocharging systems where turbochargers are activated sequentially, speed monitoring is not required for the turbocharger(s) being activated last in sequence, provided all turbochargers share the same intake air filter and are not fitted with waste gates.

(2) Exhaust gas temperature may be alternatively monitored at the turbocharger outlet, provided that the alarm level is set to a safe level for the turbine and that correlation between inlet and outlet temperatures is substantiated.

(3) Alarms and indications for exhaust gas temperatures at turbocharger inlets may be omitted if the alarms and indications for individual exhaust gas temperatures are provided for each cylinder and the alarm level is set to a value safe for the turbocharger.

(4) Separate sensors are to be provided when the lubrication oil system of the turbocharger is not integrated with the lubrication oil system of the reciprocating internal combustion engine, or when it is separated from the reciprocating internal combustion engine lubrication oil system by a throttle or pressure reduction valve.

(5) "H" and "L" mean "high" and "low", respectively.

2.5.2 Exhaust Gas Arrangements

1 Exhaust gas pipes with a surface temperature exceeding 220 ℃ are to be water-cooled or sufficiently covered with thermal insulation. However, in case where no fire is likely to occur, the requirements may be dispensed with.

2 Exhaust gas arrangements are also to comply with the requirements specified in **13.16** in this Part.

2.5.3 Starting Arrangements*

1 The starting air mains are to be protected against explosion caused by back-fire from the cylinders or excessive temperature rise in the starting air manifold at the time of starting by the following **(1)** through **(5)** arrangements:

(1) An isolating non-return valve or equivalent thereto is to be provided at the starting air supply connection to each engine.

(2) An adequate rupture disc device or a flame arrester is to be fitted at the starting valve on each cylinder for direct reversing engines having a starting air manifold. At least one such device is to be fitted at the supply inlet to the starting air manifold for each non-reversing engine. However, the above mentioned device may be omitted for engines having cylinder bore not exceeding 230 *mm*.

(3) An adequate rupture disc device is to be fitted at an appropriate position on the starting air manifold as an emergency means of relieving a pressure caused by explosion for direct reversing engines fitted with flame arresters in accordance with **(2)** above.

(4) For rupture disc devices of which ruptured discs cannot be easily replaced, a mechanism of blocking up the exhaust way is to be provided for the purpose of quick restart of the engine. This blocking mechanism is to be fitted with a means of indicating whether it is blocking or not.

(5) An effective arrangement to prevent the accumulation of combustibles (fuel oil, lubrication oil, system oil, etc.) in the starting air manifold or to prevent the excessive temperature rise in the starting air manifold is to be provided for direct reversing engines.

2 Where main propulsion engines are arranged for starting by compressed air, at least two starting air reservoirs are to be provided. These reservoirs are to be connected so that usage can be readily switched from one to the other. In this case, the total capacity of the starting air reservoirs is to be sufficient to provide, without replenishment, the number of consecutive starts not less than that specified in **(1)**, **(2)** and **(3)** below. Where the arrangements of the main propulsion engines and shafting systems are other than shown below, the required number of starts is to be as deemed appropriate by the Society. When other consumers such as auxiliary machinery starting systems, pneumatic piping systems for essential services (refer to **13.13.6(2)**), control systems, whistles, etc. are to be connected to starting air reservoirs, their air consumption is also to be taken into account.

(1) For direct reversible engines

$Z = 12C$

where

Z : Total number of starts of engine

C : Constant determined by the arrangement of main propulsion engines and shafting systems, where the following values are to be referred to as the standard;

$C = 1.0$ For single screw ships, where one engine is either coupled with the shaft directly or through reduction gears.

$C = 1.5$ For twin screw ships, where two engines are either coupled with the shafts directly or through reduction gear. Or, for single screw ships, where two engines are coupled with the shaft through declutchable coupling provided between engine and reduction gear.

$C = 2.0$ For single screw ships, where two engines are coupled with one shaft without any declutchable coupling between engine and reduction gear.

(2) For non-reversible type engines using a separate reversing gear or controllable pitch propeller, 1/2 of the total number of starts specified in **(1)** above may be accepted.

(3) For electric propulsion ships:

$Z = 6 + 3(k - 1)$

where

Z : Total number of starts of engine

k : Number of engines (In the case of more than 3 engines, the value of k to be used is 3.)

3 The capacities of the reservoirs specified in **-2** above are to be about the same.

4 Starting air reservoirs and starting air systems are also to comply with **13.13**.

5 Internal combustion engines which are arranged for electrical starting are to comply with the requirements specified in **Part H**, in addition to the following **(1)** to **(3)**:

(1) Two separate batteries are to be fitted to starting arrangements for main propulsion machinery. Arrangements are to be such that the batteries cannot be connected in parallel, and each battery is to be capable of starting the main propulsion machinery under the cold and ready-to-start condition. The combined capacity of the batteries is to be sufficient (without recharging) to provide the number of consecutive starts specified in **-2** above within 30 *minutes*.

(2) Electric starting arrangements for internal combustion engines driving generators and auxiliary machinery are to have two separate batteries but may be supplied by separate circuits from the batteries for main propulsion machinery. In the case of single auxiliary engines, only one battery needs to be fitted. The capacity of each set of batteries is to be sufficient for at least three starts for each engine.

(3) Starting batteries are to be used for starting and engine self-monitoring purposes only. Provisions are to be made to continuously maintain stored energy at all times.

2.5.4 Fuel Oil Arrangements

1 Where a reciprocating internal combustion engine is mounted on an elastic support, flexible joints approved by the Society are to be provided at the connections between the engine and the fuel oil supply pipe.

2 The fuel oil arrangements for reciprocating internal combustion engines are additionally to comply with the requirements in **13.9, Part D** and **4.2.2, Part R**.

2.5.5 Lubricating Oil Arrangements

1 The lubricating oil arrangements of reciprocating internal combustion engines with a maximum continuous output exceeding 37 *kW* are to be provided with alarm devices which give visible and audible alarming in the event of failure of the supply of lubricating oil or an appreciable reduction in lubricating oil pressure. Also, devices to stop the operation of the engine automatically by lower pressure after such alarms are to be provided.

2 The lubricating oil arrangements are to be provided with lubricating oil sampling connections at proper locations.

3 Lubricating oil arrangements for rotor shafts of exhaust driven turbochargers are to be designed so that the lubricating oil may not be drawn into charging air.

4 Lubricating oil drain pipes from the engine crankcase sump to the sump tank are to be submerged at their outlet ends.

5 The lubricating oil drain pipes shown in **-4**, above of two or more engine units are not to be interconnected.

6 Arrangements for lubricating oil system are additionally to comply with the requirements in **13.10, Part D** and **4.2.3, Part R**.

2.5.6 Cooling Arrangements

Cooling arrangements are to comply with the requirements in **13.12** in addition to the requirements in the following **(1)** and **(2)**:

(1) In engines having more than one cylinder, an adequate means is to be provided to make cooling uniform for each cylinder and piston.

(2) Drain cocks are to be fitted to water jackets and water pipe lines at their lowermost position.

2.5.7 Control Valves for Electronically-controlled Engines which are used as the Main Propulsion Machinery

1 Control valves are to be capable of retaining their expected ability to function properly for a period of time set by manufacturers.

2 Control valves are to be independently provided for each function (e.g. fuel injection, exhaust valve driving).

3 Means are to be provided to prevent fuel oil from continuously flowing into cylinders due to control valve failure.

2.5.8 Accumulators and Common Accumulators for Electronically-controlled Engines which are used as the Main Propulsion Machinery

1 Accumulators and common accumulators are to comply with the requirements in **Chapter 10**. However, notwithstanding this requirement, materials and non-destructive tests as well as surface inspections and dimension inspections are to be in accordance with **Table D2.2** and hydrostatic tests are to be in accordance with **Table D2.6**.

2 Accumulators are to be capable of retaining their expected ability to function properly for a period of time set by manufacturers.

3 In principle, at least two common accumulators are to be provided. However, in cases where results of fatigue analysis upon fluctuating stress are submitted and approved by the Society, a single arrangement may be acceptable.

2.5.9 Fuel Oil Piping Systems and Hydraulic Oil Piping Systems for Electronically-controlled Engines which are used as the Main Propulsion Machinery

1 At least two fuel oil pressure pumps and hydraulic oil pressure pumps are to be provided for their respective lines and are to be capable of supplying a sufficient amount of oil at the maximum continuous output of main propulsion machinery. In such cases, even though a single one of these pumps may become inoperable, the remaining pumps are to be capable of supplying a sufficient amount of fuel under normal service conditions. In cases where one or more of these pumps are provided as a stand-by pump, the pumps are to always be connected and ready for use.

2 Piping arrangements from fuel oil pressure pumps to the fuel injection devices and from hydraulic oil pressure pumps to exhaust valve driving gears are to be protected with jacketed piping systems or oil tight enclosures, to prevent any spread of oil from igniting.

3 Two common piping arrangements from fuel oil pressure pumps or a hydraulic oil pressure pumps to common accumulators, from one common accumulator to another common accumulator and from common accumulators to those positions where distribution to cylinders are to be respectively provided. In cases where results of fatigue analysis upon fluctuating stress are submitted and approved by the Society, a single arrangement may be acceptable.

4 Valves or cocks provided on piping connected to equipment, e.g. accumulators or pumps, are to be located as close to such equipment as practicable.

5 In high-pressure piping, high-pressure alarms are to be provided. Relief valves are also to be provided at proper positions, so as to lead any released oil to lower-pressure sides.

6 In cases where pressure gauges using bourdon-tubes are provided in high-pressure piping, such gauges are to be ones that comply with recognized industrial standards, e.g. *JIS*, and be vibration-proof and heat-resistant types.

2.5.10 Electronic Control Systems for Electronically-controlled Engines which are used as the Main Propulsion Machinery

1 Systems are to be so arranged that the function of an entire system is capable of being sustained or restored in cases where there is a single failure in any equipment part or circuit.

2 Controllers for systems are to comply with the following:

(1) At least two main controllers which are integrated to control every function, e.g. fuel injection, exhaust valve drive, cylinder lubrication and supercharge, are to be provided.

(2) Notwithstanding the requirement in **(1)** above, a single main controller may be acceptable, in cases where normal operation of main propulsion machinery is available by using control systems independent from main controllers.

3 At least two sensors essential for the operation of main propulsion machinery, e.g. for the following uses, are to be independently provided. In cases where normal operation of main propulsion machinery is available without any feedback from such sensors, single arrangements may be acceptable.

(1) Number of revolutions

(2) Crank angles

(3) Fuel pressure in common accumulators

4 Power for control systems is to be supplied from two independent sources, one of which is to be supplied from a battery, and through two independent circuits.

5 Power for driving solenoid valves is to be supplied from two independent sources, and through two independent circuits.

6 Electronic-control systems of main propulsion machinery which comply with the requirements given in **-1** through **-5** above are regarded as the same as those which comply with the following requirements.

(1) **18.2.4-5(1)**

(2) **18.3.2-3(3)**

2.5.11 Failure Mode Effect Analysis for Electronically-controlled Engines which are used as the Main Propulsion Machinery

Failure Mode Effect Analysis (FMEA) is to be carried out, for electronic control systems, in order to confirm that any one equipment or circuits in such systems which lose function may not cause any malfunction or deterioration in other equipment or circuits, in accordance with the following:

(1) Systems are to be divided into functional blocks and drawn out in reliability block diagrams in which such functional blocks are systematically organized.

(2) Analytic levels are to be sufficient up to the extent of those functional blocks regarding sub-systems and components.

(3) FMEA results are to be created in table form as shown in **Table D2.5** or be of equivalent forms thereto.

(4) If FMEA results show that corrective action is demanded, then FMEA is to be carried out again after the corrective action to confirm the effectiveness of the corrective action.

(5) For failure modes, every possible failure from minor to catastrophic is to be considered.

Table D2.5 Failure Mode Effect Analysis Table for Electronically-controlled Engines which are used as the Main Propulsion Machinery

Systems				Elements									
ID Number	Component	Sub-system	Operating mode	Failure mode	Failure cause	Failure detection Means	Alarm / Notification Means	Effect of failure			Failure severity	Corrective action	Remarks
								On component	On sub-system	On system			

Examples of Operating Mode: back-up operations, fuel cost priority operations, NOx reduction operations, etc.

Examples of Failure Mode: piston pin stuck, connecting rod broken, lubricating oil leaked out, etc. (Failed parts are to be shown.)

Failure Severity:
(a) Catastrophic: loss of complete function, explosion, loss of life (Design change is to be compulsory.)
(b) Major: loss or deterioration of part of the ability to function properly (Possible design change is to be investigated.)
(c) Minor: negligible affect on ability to function properly (Design change may not be required.)

2.6 Tests

2.6.1 Shop Tests*

1 For components or accessories specified in **Table D2.6**, hydrostatic tests are to be carried out on the water or oil side of the component at the pressures shown in the Table. In cases deemed necessary by the Society, tests may also be required for any components not specified in **Table D2.6**.

2 For reciprocating internal combustion engines, the purpose of the shop trials is to verify design premises such as engine power, safety against fire, adherence to approved limits such as maximum pressure, and functionality as well as to establish reference values or base lines for later reference in the operational phase. The programme is to be in accordance with the following:

(1) The following preparations are to be made before carrying out the engine tests:
   (a) All relevant equipment for the safety of attending personnel such as oil mist detection arrangements, overspeed protective devices and any other shut down functions are to be made available and are to be operational.
   (b) The overspeed protective device is to be set to a value which is not higher than the allowable overspeed value. This set point is to be verified by the surveyor.
   (c) The engines are to be run as prescribed by the engine manufacturer.
   (d) All fluids used for testing purposes (fuel oils, lubrication oils, cooling water, etc., including all fluids used temporarily or repeatedly for testing purposes only) are to be suitable for their intended purposes (i.e., they are to be clean, preheated if necessary and cause no harm to engine parts).

(2) For all stages of testing, the following **(a)** to **(c)** ambient conditions are to be recorded and the pertaining operation values (normally the following **(d)** to **(k)** items) for each load point are to be measured and recorded by the engine manufacturer. All results are to be compiled in an acceptance protocol to be issued by the manufacturer. Calibration records for the instrumentation are to be presented to the attending surveyor. In addition, crankshaft deflection is to be checked and recorded in the results in cases where such a check is required by the manufacturer during the operating life of the engine.
   (a) Ambient air temperature
   (b) Ambient air pressure

(c) Atmospheric humidity
(d) Power
(e) Speed
(f) Fuel index (or equivalent reading)
(g) Maximum combustion pressures (only when the cylinder heads installed are designed for such measurement)
(h) Exhaust gas temperature at the turbine inlet and from each cylinder
(i) Charge air temperature
(j) Charge air pressure
(k) Turbocharger speed

(3) All measurements conducted at the various load points are to be carried out under steady operating conditions. However, provision is to be made for time needed by the surveyor to carry out visual inspections for all load points. The readings for 100 % power (rated power at rated speed) are to be taken twice at an interval of at least 30 *minutes*.

(4) In cases where a no-load operation is conducted for adjusting engine conditions, the fuel delivery system, manoeuvring system and safety devices are to be properly adjusted by the manufacturer before the operation.

(5) The programme shown in **Table D2.7** is to be used for the shop trials of reciprocating internal combustion engines. In this case, refer to the *JIS* specified below or those considered equivalent thereto for more details on each respective testing procedure. However, additional tests may be requested by the Society depending on the engine application, service experience, or other relevant reasons. In addition, alternatives to the detailed tests may be agreed between the manufacturer and the Society when the overall scope of tests is found to be equivalent.

(a) In the case of reciprocating internal combustion engines used as main propulsion machinery (including those used as main propulsion machinery for electric propulsion ships);
JIS F 4304 "Shipbuilding - Internal combustion engines for propelling use-shop test code"

(b) In the case of reciprocating internal combustion engines driving other generators or essential auxiliary machinery;
JIS F 4306 "Shipbuilding - Water cooled four-cycle generator diesel engines"

(6) The following **(a)** to **(c)** are to be inspected. However, a part of or all of these inspections may be postponed until shipboard testing when agreed to by the Society.

(a) Jacketing of high-pressure fuel oil lines, including the system used for the detection of leakage
(b) Screening of pipe connections in piping containing flammable liquids
(c) Temperature of hot surface insulation
Random temperature readings are to be compared with corresponding readings obtained during the type test. This is to be done while running at the rated power of engine. If the insulation is modified subsequently to the type test, the Society may request temperature measurements as required by the type test.
In the case of reciprocating internal combustion engine with an application for approval of use dated before 1 July 2016 which is an engine type that does not have the results of temperature measurements required by the type test, temperature measurements are to be performed by a procedure deemed appropriate by the Society.

(7) Category *C* turbochargers installed on reciprocating internal combustion engines used as main propulsion machinery are to be checked for surge margins in accordance with the following. However, if successfully tested earlier on an identical configuration of the engine and turbocharger (including the same nozzle rings), submission of this test report may be accepted instead.

(a) For 4-stroke engines, the operations given in the following **i)** and **ii)** are to be performed without any indication of surging.
i) While at maximum continuous rating (maximum continuous power and speed), speed is to be reduced with the constant torque (fuel index) down to 90 % power.
ii) While at 50 % power and 80 % speed, speed is to be reduced to 72 % while keeping constant torque (fuel index).

(b) For 2-stroke engines, the surge margin is to be demonstrated by at least one of the following **i)** to **iii)**:
i) The engine working characteristics established at shop tests of the engine is to be plotted into the compressor chart of the turbocharger (established in a test rig). There is to be at least a 10 % surge margin in the full load range, i.e., working flow is to be 10 % above the theoretical mass flow at the surge limit where there are no pressure fluctuations.
ii) A sudden fuel cut-off to at least one cylinder at the following **1)** and **2)** loads is not to result in continuous surging and the turbocharger is to be stabilised at the new loads within 20 *seconds*. For applications with more than one

turbocharger, the fuel supply to the cylinders closest upstream to each turbocharger is to be cut off.

1) The maximum power permitted for one cylinder misfiring.

2) The engine load corresponding to a charge air pressure of about 0.06 *MPa*, but without auxiliary blowers running.

iii) No continuous surging and the turbocharger is to be stabilised at the new load within 20 *seconds* when the power is abruptly reduced from 100 % to 50 % of the maximum continuous power.

(8) For electronically controlled engines, integration tests are to be made to verify that the response of the complete mechanical, hydraulic and electronic system is as predicted. The scope of these tests is to be determined based on a risk analysis by a method deemed appropriate by the Society and agreed with the Society, prior to the tests. The tests may be carried out using other alternative methods, subject to special consideration by the Society.

3 For gas-fuelled engines (specified in **4.2.2, Annex 1.1.3-3, Part GF** or **5.2.2, Annex 16.1.1-3, Part N**), the following requirements are to be complied with. In addition, the scope of the tests may be expanded depending on the engine application, service experience, or other relevant reasons.

(1) The requirements specified in **-2(1)** to **(7)** apply subject to following **(2)** to **(5)** requirements.

(2) For dual fuel engines, the tests specified in **Table D2.7** are to be carried out for both diesel and gas mode. However, for loads considered by the Society not to be designed to operate, the load test may be omitted. For load tests for the gas mode, test loads are to be determined based on the maximum continuous power available in the gas mode (see **2.5.1-1(1), Annex 1.1.3-3, Part GF** or **2.5.1-1(1), Annex 16.1.1-3, Part N**). The 110 % load test is not required for the gas mode provided that changeover to oil fuel mode is automatically performed in case of overload.

(3) In addition to the preparations specified in **-2(1)**, measures to verify that gas fuel piping for the engine is gas tight are to be carried out prior to the start-up of the engine.

(4) In addition to **-2(2)** and **(3)**, the following engine data are to be recorded.

(a) The item listed in **-2(2)(f)** is to be measured and recorded for both gas and diesel, as applicable

(b) Gas pressure and temperature

(c) Pilot fuel temperature and pressure (supply or common rail as appropriate)

(5) The engines are to undergo integration tests to verify that the responses of the complete mechanical, hydraulic and electronic systems are as predicted for all intended operational modes. The scope of these tests is to be agreed to with the Society for selected cases based upon risk analysis by a procedure deemed appropriate by the Society and is to at least include the following incidents. The tests may be carried out using simulation or other alternative methods, subject to special consideration by the Society.

(a) Failure of ignition (spark ignition or pilot injection systems)

(b) Failure of a cylinder gas supply valve

(c) Failure of combustion (to be detected by e.g. misfiring, knocking, exhaust temperature deviation, etc.)

(d) Abnormal gas pressure

(e) Abnormal gas temperature

4 For rotating assemblies of exhaust driven turbochargers of categories *B* and *C*, dynamic balancing tests are to be carried out.

5 For the impellers and inducers of exhaust driven turbochargers of categories *B* and *C*, overspeed tests for a duration of 3 *minutes* at either of the following **(1)** or **(2)** are to be carried out. For forged impellers and inducers subject to quality control through an approved non-destructive test method, overspeed tests may be dispensed with.

(1) 120 % of the alarm level speed at room temperature; or

(2) 110 % of the alarm level speed at an inlet temperature of 45 °C when tested in the actual housing with the corresponding pressure ratio.

6 For categories *B* and *C* turbochargers, tests are to be carried out to verify durability according to procedures deemed appropriate by the Society.

Table D2.6 Hydrostatic Test Pressure

Part		Cylinder bore *D* (*mm*)		Test Pressure(2)(*MPa*)
		D≤300	300<*D*	
Cylinder block (gray cast iron or spheroidal graphite cast iron) (3) (4)		○	○	1.5*P*
Engine block (gray cast iron or spheroidal graphite cast iron) (3) (4)		○	○	1.5*P*
Cylinder liner (4)			○	1.5*P*
Cylinder head (gray cast iron, spheroidal graphite cast iron, cast steel or forged steel)			○	1.5*P*
High pressure fuel line	Fuel injection pump body	*TR*(6)	○	1.5*P* or *P* +30, whichever is smaller
	fuel injection valves (5)			
	fuel injection pipes including common fuel rail (5)	*TR*(6)	○	
High pressure common servo oil system		*TR*(6)	○	1.5*P*
Turbocharger, cooling space (7)	Category *A*			0.4 or 1.5*P*, whichever is greater
	Category *B*	○	○	
	Category *C*	○	○	
Heat exchanger, both sides			○	1.5*P*
Exhaust gas valve cage (8)		○	○	1.5*P*
Accumulator (9)		○	○	1.5*P*
Piping, pumps, actuators, etc. for hydraulic drive of valves (10)		○	○	1.5*P*
Engine driven pumps (oil, water, fuel, bilge) (10)		○	○	1.5*P*
Piping system other than those listed in this Table		○	○	Apply the requirements in **12.6**

Notes:

(1) Materials intended for the components marked by "○"or "*TR*" are to be tested by hydrostatic test.

(2) *P* is the maximum working pressure (*MPa*).

(3) Only when engine power exceeds 400 *kW*/*cyl*.

(4) Hydrostatic tests are also required for those parts filled with cooling water that have the ability to contain water which is in contact with the cylinder or cylinder liner.

(5) Only when not autofretted.

(6) For items marked by "*TR*", submission of a test report which compiles all test and inspection results in an acceptance protocol issued by the manufacturer may be accepted. The test report is to include the following. Tests or inspections may be carried out on samples from the current production.

(a) Signature of the manufacturer

(b) Statement that components comply with specifications stipulated by the manufacturer

(7) In cases where the manufacturer has a quality system deemed appropriate by the Society, hydrostatic tests for category *B* turbochargers may be substituted for by manufacturer tests. In such cases, the submission or presentation of test records may be required by the Society.

(8) Only for crosshead reciprocating internal combustion engines.

(9) Only when capacity exceeds 0.5 *l*.

(10) Only when engine power exceeds 800 *kW*/*cyl*.

Table D2.7 Programme for Shop Trials of Engines

<table>
<tr><td colspan="2" rowspan="2">Test items</td><td colspan="3">Use of engines</td></tr>
<tr><td>Reciprocating internal combustion engines used as main propulsion machinery(1)</td><td>Reciprocating internal combustion engines driving generators (including those used as main propulsion machinery of electric propulsion ships) (2)</td><td>Reciprocating internal combustion engines driving auxiliaries (excluding auxiliary machinery for specific use etc.) (1)</td></tr>
<tr><td rowspan="6">Load test</td><td>110 % power run (3)</td><td>15 minutes or until steady conditions have been reached, which is shorter, at 1.032 n_0 or more (where n_0 is the rated engine speed)(4), (5)</td><td>15 minutes after having reached steady conditions at n_0</td><td>15 minutes after having reached steady conditions at n_0</td></tr>
<tr><td>100 % power run</td><td>60 minutes at n_0</td><td>60 minutes at n_0</td><td>30 minutes at n_0</td></tr>
<tr><td>90 % power run (or normal continuous cruise power) (6), (7)</td><td rowspan="4">20 minutes at engine speed in accordance with the nominal propeller curve</td><td>-</td><td>-</td></tr>
<tr><td>75 % power (6), (7)</td><td rowspan="3">20 minutes at n_0</td><td rowspan="3">20 minutes in accordance with the nominal power consumption curve (8)</td></tr>
<tr><td>50 % power (6), (7)</td></tr>
<tr><td>25 % power (6), (7)</td></tr>
<tr><td colspan="2">Idle run (6)</td><td>-</td><td>An adequate time at n_0</td><td>-</td></tr>
<tr><td colspan="2">Reversing manoeuvres (9)</td><td>○</td><td>-</td><td>-</td></tr>
<tr><td colspan="2">Intermittent overload (10)</td><td>○</td><td>-</td><td>○</td></tr>
<tr><td colspan="2">Governor test</td><td>-</td><td>○</td><td>-</td></tr>
<tr><td colspan="2">Performance of monitoring, alarm and safety devices</td><td>○</td><td>○</td><td>○</td></tr>
<tr><td colspan="2">Open-up inspection (11)</td><td>○</td><td>○</td><td>○</td></tr>
</table>

Notes:

(1) After testing has been completed, the fuel delivery system is to be blocked so as to limit the engines to run at not more than 100 % power, unless intermittent overload power is approved by the Society. In the case of propulsion engines also driving power take-off generators, the fuel delivery system is to be adjusted so that overload of generator (110 % power) can be given in service and the electrical protection of downstream system components is activated before the engine stalls.

(2) After testing has been completed, the fuel delivery system is to be adjusted such that overload (110 % power) can be given in service after installation on board so that the governing characteristics (including the activation of generator protective devices) can be fulfilled at all times.

(3) For dual fuel engines, tests in the gas mode are not required in accordance with **2.6.1-3(2)**.

(4) Submission of test reports for identical engines and turbocharger configurations proving their compatibility for over-loaded operation may be accepted as substitutions for the 110 % power run.

(5) In the case of propulsion engines also driving power take-off generators, tests are to be carried out at n_0 for 15 *minutes* after having reached a steady operating condition.

(6) The sequence is to be selected by the engine manufacturer.

(7) A shorter time may be considered by the Society provided that the time specified in **2.6.1-2(3)** is allowed.

(8) Only for variable speed engines.

(9) The test item applies only to direct reversible engines.

(10) Only for engines for which intermittent overload is approved, and tests are to be for the duration agreed upon with the manufacturer.

(11) The scope of the open-up inspection is to be as deemed appropriate by the surveyor. The omission of the open-up inspection may be considered by the Society provided that all of the following **(a)** through **(g)** are met:

(a) It is not the open-up inspection to be carried out during the approval test specified in **Chapter 8, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

(b) No abnormality is found in the temperature measurement for each bearing of the main bearings and the crank pin bearings after the load test, and in the visual inspection of the inner surfaces of the cylinder liners from the inspection ports of the crankcase. (In the case of a 2-stroke engine, the cylinder liners, pistons, piston rings and piston rods are to be inspected from the scavenging space.)

(c) No abnormality is found in the visual inspection of the lubrication oil after the load test (including the visual inspection of the filter in cases where the open-up of the strainer is reasonable).

(d) Flushing of the parts through which the lubrication oil passes is carried out during the manufacturing process.

(e) The manufacturer of the reciprocating internal combustion engine is approved by the Society in accordance with the **Rules for Approval of Manufacturers and Service Suppliers**.

(f) There is agreement between the involved parties. (manufacturer, shipyard, prospective owner, etc.)

(g) Other items deemed necessary by the Society.

Chapter 3 STEAM TURBINES

3.1 General

3.1.1 Scope

The requirements in this Chapter apply to steam turbines used as main propulsion machinery, or used to drive generators and auxiliaries (hereinafter referred to in this Chapter as all auxiliaries excluding auxiliary machinery for specific use etc.).

3.1.2 Drawings and Data

Drawings and data to be submitted are as follows:

(1) Drawings and data for approval
 (a) Turbine casings
 (b) Turbine rotors
 (c) Turbine blades
 (d) Details of turbine installation
 (e) Shaft coupling and bolts
 (f) Piping arrangements fitted to turbine (including steam, lubricating oil and drain system, and information regarding pipe materials, pipe sizes and service pressures)
 (g) Turbine particulars (power and number of revolutions at maximum continuous rating of turbines, steam pressure and temperature of turbine inlet, degree of vacuum at condenser top or steam condition in exhaust chamber)
 (h) Critical speed of each turbine rotor
 (i) Number of blades in each stage
 (j) Number of nozzles and their arrangements in each stage
 (k) Technical data for strength calculations specified in **3.2.3**
 (l) Material specifications of principal components
 (m) Details of welding procedures for principal components

(2) Drawings and data for reference
 (a) Sectional assembly
 (b) Control system diagram
 (c) Drawings and data which are deemed necessary by the Society

3.2 Materials, Construction and Strength

3.2.1 Materials

1 Materials of the components of steam turbine specified below (hereinafter referred to as "the principal components of steam turbine") are to comply with the requirements in **Part K**.

(1) Turbine rotors
(2) Turbine blades
(3) Turbine casings
(4) Shaft couplings and coupling bolts
(5) Pipes, valves and fittings attached to steam turbine classified as either Group I or II as specified in **Chapter 12**

2 The principal components of steam turbines (excluding coupling bolts, pipes, valves and fittings) are to have been subjected to the non-destructive tests specified in **5.1.10** and **6.1.10, Part K**.

3 The materials used in high temperature parts are to have properties suitable for the design, performance and service life against corrosions, thermal stresses, creeps and relaxations.

4 Cast iron is not to be used for the turbine casing and others subjected to pressure where the steam temperature exceeds 230℃.

3.2.2 General Construction

1 For ships provided with one cross-compound type main steam turbine, the turbine is to be so constructed as to be capable of ensuring operation at navigable speed even when the steam led to any one of the cylinders is cut off in an emergency. For this operation, the following **(1)** and **(2)** are to be complied with:

(1) The permissible steam pressures and temperatures, speeds, etc. are to be specified and information is to be provided on board considering the safety of the turbine and condenser as well as any potential influence on shaft alignment and gear teeth loading conditions.

(2) All necessary pipes and valves are to be readily available and properly marked, and the procedure is to be documented locally.

2 Each part of a turbine is to be so constructed that no detrimental deformations are caused by their thermal expansions. Turbines are to be installed on the seatings so that no excessive structural constraints are caused by thermal expansions.

3 Where the principal components of steam turbine are of welded construction, they are to comply with the requirements in **Chapter 11**.

4 Turbine casings are to be provided with drain connections at suitable locations.

5 Non-return valves or other approved means which prevent steam and drain from returning to the turbines, are to be fitted in bled steam connections.

6 In steam turbines used as main propulsion machinery, steam strainers are to be provided at the turbine inlet or the inlet to the manoeuvring valves.

7 The construction of main condensers is to conform to the requirements in **Chapter 10**.

3.2.3 Strength of Turbine Rotors and Blades

1 The strength of turbine rotors is to comply with the requirements of the following **(1)** and **(2)**:

(1) Turbine rotors (or discs) are to be so designed that no excessive vibration is induced within the operating speed range.

(2) Mean tangential stress of turbine rotors are to satisfy the following conditions. Since these conditions do not take into account factors of creep and other design considerations for the materials, special consideration is to be given to these stress conditions as considered necessary.

$$T_m = \frac{n^2(1.10\rho I + 0.1766mr)}{A}$$

$$T_m \leq Y/3$$

$$T_m \leq T_s/4$$

where

T_m : Mean tangential stress (N/mm^2)

n : Number of maximum continuous revolutions per minute divided by 1,000

A : Sectional area of wheel profile on one side of axis of rotation (cm^2)

I : Moment of inertia of area A about the axis of rotation (cm^4)

ρ : Density of turbine disc or rotor (kg/cm^3)

m : Total mass of blades including roots (kg)

r : Distance between the centre of gravity of blade (including root) and the centre line of shaft (cm)

Y : Specified yield strength or proof stress of the material (N/mm^2)

T_s : Specified tensile strength of the material (N/mm^2)

2 Strength of turbine blades is to comply with the requirements of the following **(1)** and **(2)**:

(1) Turbine blades are to be so designed as to avoid abrupt changes in section and to provide sufficient rigidity to minimize deflection and vibration.

(2) The sectional area at the root of the blade is not to be less than the value obtained from the following formula. Where, however, deemed appropriate by the Society, the formula may be modified.

$$A = \frac{4.395mrn^2}{T_s}$$

where

A : Required minimum sectional area at root of blade (cm^2)

m : Mass of a blade upward of the section with area A and shroud (kg)

r : Distance between the centre of gravity of blade (including root) and the centre line of shaft (cm)

T_s : Specified tensile strength of blade material (N/mm^2)

n : Number of maximum continuous revolutions per minute divided by 1,000

3.3 Safety Devices

3.3.1 Governors and Overspeed Protective Devices

1 All main and auxiliary steam turbines are to be provided with overspeed protective devices to prevent the engine speed from exceeding the number of maximum continuous revolutions by more than 15 %. Where two or more steam turbines are coupled to the same main gear wheel, only one overspeed protective device provided for all the turbines may be accepted.

2 In addition to this speed governor, for ships in which steam turbines are used as main propulsion machinery (excluding electric propulsion ships) which can be declutched, or which drives a controllable pitch propeller, each of such steam turbines is to be provided with a separate and independent speed governor in addition to the overspeed protective device specified in **-1** above. This additional speed governor is to be capable of controlling the speed of the unloaded turbine without bringing the overspeed protective device into action.

3 Steam turbines to drive generators are to be provided with governors complying with the requirements in **-4** in addition to the overspeed protective device specified in **-1** above. However, if steam turbines used as main propulsion machinery in electric propulsion ships are used to drive generators for supplying electrical power exclusively to propulsion motors, the requirements in **5.1.2-2, Part H** are to be applied.

4 Speed governors of steam turbines driving generators are to be as follows:

(1) Characteristics of governors are to comply with the requirements specified in **2.4.1-5** (in this case the term "reciprocating internal combustion engines" is to be read as "steam turbines").

(2) In cases where a steam turbine-driven *d.c.* generator is arranged to run in parallel with other generators, a switch is to be fitted on each steam turbine emergency governor for the purpose of opening the generator circuit-breaker when the emergency governor functions.

3.3.2 Steam Shut-off Devices

1 Steam turbines used as main propulsion machinery are to be provided with devices which automatically shut off the steam supply to the ahead turbine (for steam turbines used as main propulsion machinery of electric propulsion ships, the turbines used for that purpose) in the following cases:

(1) In the case of low lubricating oil pressure

(2) In the case of low main condenser vacuum

2 Steam turbines to drive generators or auxiliaries are to be provided with devices which automatically shut off the steam supply in the case of low lubricating oil pressure.

3 Arrangements are to be provided for shutting off the steam supply to steam turbines used as the main propulsion machinery by suitable manually operated gears installed at the manoeuvring stand and at the turbine respectively. Manually operated gears for turbines to drive generators or auxiliaries are to be arranged in the vicinity of the turbines.

3.3.3 Lubricating Oil Supply System

1 Steam turbines used as main propulsion machinery are to be provided with a satisfactory emergency supply of lubricating oil which comes into service automatically when the pressure drops below a predetermined pressure level. This emergency supply may be obtained from a gravity tank or equivalent means (e.g. attached pump) with sufficient amount of oil to ensure adequate lubrication until the turbine is brought to rest.

2 The lubricating oil arrangements of a steam turbine are to be provided with alarm devices which give visual and audible alarms in the event of failure of the supply of lubrication oil or an appreciable reduction of lubricating oil pressure before the function of the steam shut off devices specified in **3.3.2-1(1)** and **-2**.

3.3.4 Sentinel Valve for Exhaust Steam Outlet

A sentinel valve is to be provided at the exhaust end of all turbines to prevent against the abnormal rise of the exhaust steam pressure.

3.4 Tests

3.4.1 Shop Tests*

1 The following components are to be subjected to hydrostatic tests at pressures specified below:

(1) Turbine casings:

: 1.5 times the design-steam pressure for the turbine casing or 0.2 *MPa*, whichever is greater.

(2) High pressure turbine steam chests:

: 1.5 times the nominal pressure of the boiler.

(3) Steam receivers, pipes and valve chests etc.:

The same pressure as the hydrostatic test pressure that is applicable to the turbine casing to which they belong.

(4) Steam strainers, and manoeuvring valve chests:

: 2 times the nominal pressure of the boiler

(5) Steam space of main condenser:

: 0.1 *MPa*

Cooling water space:

: 0.2 *MPa* or 0.1 *MPa* plus the maximum discharge pressure which the circulating pumps can develop with the discharge valve closed and the maximum suction pressure which develops under the full draught condition, whichever is greater.

Where the conditions of service are unknown and the pressure under the conditions above is unable to be calculated, the test pressure is not to be less than 0.34 *MPa*.

2 For turbine rotors, dynamic balancing tests are to be carried out by the test procedure deemed appropriate by the Society.

3 For steam turbines, shop trials are to be carried out, including the test of the safety devices specified in **3.3** above, by procedures deemed appropriate by the Society.

3.4.2 Tests after Installation On Board

A fit up test, to ensure the availability of the operation in compliance with **3.2.2-1**, is to be carried out prior to the sea trials. This test may be carried out at the shop tests.

Chapter 4 GAS TURBINES

4.1 General

4.1.1 Scope

1 The requirements in this Chapter apply to open cycle gas turbines (i.e., thermodynamic cycle in which the working fluid enters the gas turbine from the atmosphere and is discharged into the atmosphere) used as main propulsion machinery, or used to drive generators and auxiliaries (hereinafter referred to in this Chapter as all auxiliaries excluding auxiliary machinery for specific use, etc.). The requirements of this Chapter apply mutatis mutandis to other cycle types gas turbines.

2 Gas turbines for driving emergency generators are to comply with the requirements in **3.3** and **3.4, Part H**, in addition to the requirements (excluding **4.2.1-1**, **4.2.1-2**, **4.3.1-1**, **4.3.2** and **4.3.3**) in this Chapter.

4.1.2 Terminology

The terminology used in this Chapter is as specified in the following **(1)** to **(5)**:

(1) “Gas generator” is an assembly of gas turbine components that produces heated pressurized gas to a process or to a power turbine.

(2) “Power turbine” is a turbine which is driven by the gases from a gas generator, producing power output from the gas turbine through an independent shaft.

(3) “Combustion chamber” is a component of a gas turbine in which fuel (heat source) reacts with the working fluid to increase its temperature.

(4) “Enclosure” is barriers, used to protect personnel, protect equipment from the environment, contain fires and possibly provide sound attenuation.

(5) “Principal components” of gas turbines are those listed in the following **(a)** to **(h)**:

(a) Discs (or rotors), stationary blades and moving blades of the turbine

(b) Discs, stationary blades and moving blades of the compressor

(c) Turbine and compressor casings

(d) Combustion chambers

(e) Turbine output shafts

(f) Connecting bolts for main turbine components

(g) Shaft couplings and bolts

(h) Pipes, valves and fittings attached to a gas turbine classified in **Chapter 12** as either Group I or II

4.1.3 Drawings and Data*

Drawings and data to be submitted are as follows:

(1) Drawings and data for approval

(a) Discs (and/or rotors) of the turbine and compressor

(b) Combustion chambers

(c) Details of the fixing of moving and stationary blades

(d) Shaft couplings and bolts

(e) Piping arrangements fitted to the turbine (including fuel, lubricating oil, cooling water, pneumatic and hydraulic systems and information on materials, sizes and working pressures of pipes)

(f) Pressure vessels and heat exchangers (classified as Group I and Group II as defined in **10.1.3**) attached to the turbine

(g) Details of turbine installation

(h) Particulars (type and product number of the turbine, power and number of revolutions per minute of the turbine and compressors at maximum continuous rating, temperatures and pressure at turbine inlet and outlet, pressure losses in inlet air and exhaust gas arrangements, ambient condition intended for operation, fuel oil and lubricating oil to be used)

(i) Material specifications of principal components

(j) Critical speeds of turbine rotors and compressors

(k) Number of moving blades in each stage
(l) Number and arrangements of stationary blades
(m) Lists of safety devices, including those specified in **4.3.5**
(n) In the case of a gas turbine without service records for Society-classed ships or the modification of specifications of a gas turbine with such service records, the following **i)** and **ii)**:
i) Welding details of principal components
ii) Maintenance instructions

(2) Drawings and data for reference
(a) A list containing all drawings and data submitted (with relevant drawing numbers and revision status)
(b) Sectional assembly
(c) Moving blades and stationary blades
(d) General arrangement
(e) Starting arrangement
(f) Inlet air and exhaust gas arrangements
(g) Diagram of engine control systems
(h) Documents containing strength considerations made for principal components
(i) Calculation sheets for vibration of turbine blades
(j) Documentation on the failure mode and effect analysis
(k) In the case of a gas turbine without service records for Society-classed ships or modification specifications of a gas turbine with such service records, the following **i)** and **ii)**:
i) Operation instructions for fuel oil control systems
ii) Illustrative drawing of cooling method for each part of turbine
(l) Other drawings and data deemed necessary by the Society

4.2 Materials, Construction and Strength

4.2.1 Materials

1 Materials intended for the principal components of gas turbines are to comply with the requirements in **Part K**.

2 The principal components of gas turbines (excluding bolts, pipes, valves and fittings) are to be subjected to the non-destructive tests specified in **5.1.10** and **6.1.10, Part K**.

3 Materials used for high temperature parts are to have properties suitable against corrosion, thermal stress, creep and relaxation in order to maintain intended performance and achieve the intended service life. In cases where the base material is coated, for example, with corrosion-resistant surfacing, the coating material is to have properties such that it is hard to detach from the base material the strength of the base material is not impaired.

4.2.2 Construction and Installations

1 Gas turbines are to be so designed that no excessive vibration and surging, etc. are induced within the operating speed range.

2 Each part of gas turbines is to be so constructed that no detrimental deformations are caused by its thermal expansions.

3 Where the principal components of gas turbines are of welded construction, they are to comply with the requirements in **Chapter 11**.

4 Gas turbines used as main propulsion machinery are to be so designed that they can restart immediately when the electrical power supply is resumed after any stoppage resulting from a temporary failure of the main source of electrical power.

5 Gas turbines are to be installed so that no excessive structural constraints are caused by thermal expansion.

6 Gas turbines are to be installed so that any turbine or compressor blade loss or any failure of other principal components does not endanger persons and machinery in the vicinity of the gas turbine. In addition, gas turbines are to be constructed to contain, as far as possible, turbine or compressor blades and any blade debris in the event of blade loss.

4.3 Safety Devices

4.3.1 Governors and Overspeed Protective Devices

1 Gas turbines are to be provided with an overspeed protective device. This device is to be so adjusted that the output shaft speed may not exceed the maximum continuous speed by more than 15 % and is to have the functions specified in **4.3.2-2**.

2 Gas turbines are to be provided with a speed governor independent of the overspeed protective device specified in **-1** above. The speed governor is to be capable of controlling the speed of the unloaded gas turbine without bringing the overspeed protective device into action.

3 The governors of gas turbines used to drive generators are to comply with the requirements in **-4**. However, when gas turbines used as main propulsion machinery in electric propulsion ships are used to drive generators to supply electric power exclusively to propulsion motors, the requirements in **5.1.2-2, Part H** are to be applied.

4 Speed governors of gas turbines driving generators are to be as follows:

(1) Characteristics of governors are to comply with the requirements specified in **2.4.1-5** (in this case the term "reciprocating internal combustion engines" is to be read as "gas turbines").

(2) In cases where a gas turbine-driven *d.c.* generator is arranged to run in parallel with other generators, a switch is to be fitted on each gas turbine emergency governor for the purpose of opening the generator circuit-breaker when the emergency governor functions.

4.3.2 Shut-down Devices

1 Gas turbines are to be provided with hand trip gear for shutting off the fuel in an emergency which is to be provided at the control station.

2 Unless the FMEA proves that the adverse effects due to failures occurring are within acceptable ranges, the shut-down functions for gas turbines are to be provided in accordance with **Table D4.1**.

3 Gas turbines are to be provided with a quick closing device (shut-down device) which automatically shuts off the fuel supply to the turbines at least in the cases of the following **(1)** to **(7)**. In addition, means are to be provided so that alarms are operated at the control station by the activation of these shut-down devices.

(1) Over speed

(2) Unacceptable lubricating oil pressure drop (for gas turbines other than the main gas turbines, only in the case where forced lubrication is adopted.)

(3) Failure of the lubricating oil system

(4) Failure in automatic starting

(5) Loss of flame during operation

(6) Excessive vibrations

(7) Excessive high temperature of gas at the turbine inlet or outlet

4 In addition to the requirements specified in **-3** above, gas turbines used as main propulsion machinery are to be provided with a quick closing device (shut-down device) which automatically shuts off the fuel supply to the turbines in at least the following **(1)** to **(3)** cases. In addition, means are to be provided so that alarms are operated at the control station by the activation of these shut-down devices.

(1) Excessive axial displacement of each rotor (except for gas turbines with roller bearings)

(2) Unacceptable lubricating oil pressure drop of reduction gear

(3) Excessive high vacuum pressure at the compressor inlet

4.3.3 Alarms

Gas turbines are to be provided with alarm devices as required by **Table D4.1**. The addition or omission of alarm devices, however, may be accepted taking into account the results of failure mode and effects analysis (FMEA).

4.3.4 Fire Detection and Extinction Systems in Enclosures

Where gas generators and the high pressure oil pipes of gas turbines are surrounded by an enclosure, the enclosure is to be provided with fire detection systems and a fire extinguishing system which complies with the requirements of **Part R**.

4.3.5 Additional Safety Devices

Gas turbines may be required to be provided with additional safety devices as required in order to safeguard against hazardous conditions arising in the event of malfunctions in the gas turbine installation. Such hazardous conditions are to be verified by the manufacturer in accordance with the failure mode and effects analysis (FMEA).

Table D4.1 Emergency Shutdown and Alarm Settings(1)

Monitoring parameter	Alarm	Emergency Shutdown	
		Gas turbines used as main propulsion machinery	Gas turbines other than those used as main propulsion machinery
Turbine speed	H	X	X
Lubricating oil pressure	L(2)	X	X(3)
Failure of the lubricating oil system	○(4)	X	X
Lubricating oil pressure of reduction gear	L(2)	X	
Differential pressure across lubricating oil filter	H		
Lubricating oil temperature	H		
Oil fuel supply pressure	L		
Oil fuel temperature	H		
Cooling medium temperature	H		
Bearing temperature	H		
Flame and ignition failure	○	X	X
Automatic starting failure	○	X	X
Vibration	H(2)	X	X
Axial displacement of rotor	H	X	
Exhaust gas temperature at the turbine inlet	H(2)	X	X
Exhaust gas temperature at the turbine outlet	H(2)	X	X
Vacuum pressure at the compressor inlet	H(2)	X	
Loss of control system	○		

Notes:

(1) "H" and "L" mean "high" and "low". "○" means abnormal condition occurred.

(2) Alarms are to be activated at the suitable setting points prior to arriving the critical condition for the activation of shut-down devices in the case where such shutdown is required.

(3) Only in the case where forced lubrication is adopted.

(4) Alarms are to be audible and visual.

4.4 Associated Installations

4.4.1 Air Inlet Systems

Air inlet systems are to be so constructed and arranged that any intrusion of harmful particles and water into compressors can be minimized. In addition, means are to be provided to minimize the detrimental effects caused by any salt deposits in the suction air, and if necessary, by any icing of the air intake.

4.4.2 Exhaust Gas Arrangement, etc.*

1 The open ends of exhaust gas pipes are to be located so as to prevent exhaust gas from entering into the air inlet system.

2 Boilers and heat exchangers utilizing the exhaust heat of gas turbines are also to comply with the requirements specified in **Chapter 9** and **Chapter 10**.

3 Exhaust gas arrangements and other hot surfaces are to be water-cooled or sufficiently covered with thermal insulation so that surface temperature does not exceed 220℃. However, in cases where no fire is likely to occur, this requirement may be dispensed with.

4 Exhaust gas arrangements are also to comply with the requirements specified in **13.16**.

4.4.3 Starting Arrangements*

1 Starting devices are to be so arranged that the firing operation is discontinued and the main fuel valve is closed within a pre-determined time in cases where ignition fails. In addition, gas turbines are to be provided with automatic or interlocked means for the following **(1)** or **(2)** before ignition commences (on starting) or recommences so as to prevent abnormal combustion or ignition trouble.

(1) Clearing all parts of the main gas turbine of the accumulation of liquid fuel; or

(2) Purging gaseous fuel

2 Where compressed air is used for starting, the starting arrangement is to comply with **13.13**, in addition to the following **(1)** to **(3)**:

(1) In order to protect starting air mains against the effects of backfiring and internal explosion in the starting air pipes (including explosion arising from improper functioning of starting valves), means are to be provided in accordance with the following **(a)** to **(e)**:

(a) An isolation non-return valve or equivalent is to be fitted at the starting air supply connection to each gas turbine.

(b) A rupture disc or flame arrester is to be fitted in way of the supply inlet to the starting air manifold.

(c) In cases where an flame arrester is provided in accordance with **(b)** above, a rupture disc is to be fitted at an appropriate position on the starting air manifold as an emergency means for pressure relief.

(d) For rupture discs which cannot be readily replaced, a mechanism of blocking up the exhaust way is to be provided for the purpose of quick restart of the gas turbine. This blocking mechanism is to be fitted with a means of indicating whether it is blocking or not.

(e) An effective arrangement to prevent the accumulation of oils in the starting air manifold or to prevent the excessive temperature rise in the starting air manifold is to be provided.

(2) The arrangement for the air starting of main propulsion machinery is to be provided with at least two starting air reservoirs which may be used independently. The total capacity of the air reservoirs is to be sufficient to provide, without their being replenished, the number of consecutive starts of main propulsion machinery not less than the following **(a)** and **(b)**. Where the arrangements of the main propulsion machinery and shafting systems are other than those shown below, the required number of starts is to be as deemed appropriate by the Society. When other consumers such as auxiliary machinery starting systems, pneumatic piping systems for essential services (refer to **13.13.6(2)**), control systems, whistles, etc. are to be connected to starting air reservoirs, their air consumption is also to be taken into account.

(a) Ships other than electric propulsion ships

$Z = 6C$

where

Z: Total number of starts of gas turbines

C: Constant determined by the arrangement of gas turbines and shafting systems, where the following values are to be referred to as the standard

$C = 1.0$: Single screw ships, where one gas turbine is either coupled with the shaft directly or through reduction gears.

$C = 1.5$: Twin screw ships, where two gas turbines are either coupled with the shafts directly or through reduction gear, and for single screw ships, where two gas turbines are coupled with the shaft through declutchable coupling provided between gas turbines and reduction gear.

$C = 2.0$: Single screw ships, where two gas turbines are coupled with one shaft without any declutchable coupling between gas turbines and reduction gear.

(b) Electric propulsion ships

$Z = 6 + 3\,(k\text{-}1)$

where

Z: Total number of starts of gas turbines

k: Number of engines (In the case of more than three gas turbines, the value of k to be used need not exceed three.)

(3) The capacities of the reservoirs specified in **(2)** above are to be about the same.

3 Gas turbines which are arranged for electrical starting are to comply with the requirements specified in **Part H**, in addition to the following **(1)** to **(3)**:

(1) Two separate batteries are to be fitted to the starting arrangement for main propulsion machinery. The arrangement is to be such that the batteries cannot be connected in parallel, and each battery is to be capable of starting the main propulsion machinery under cold and ready-to-start conditions. The combined capacity of the batteries is to be sufficient (without recharging) to provide the number of consecutive starts specified in **-2** above within 30 *minutes*.

(2) Electric starting arrangements for gas turbines driving generators and auxiliary machinery are to have two separate batteries, but may be supplied by separate circuits from the batteries for main propulsion machinery. In the case of a single gas turbine, only one battery need be fitted. The capacity of each set of batteries is to be sufficient for at least three starts for each gas turbine.

(3) The starting batteries are to be used for starting and the gas turbine's own monitoring purposes only. Provisions are to be made to continuously maintain the stored energy at all times.

4 Gas turbines which are arranged for hydraulic starting are to comply with the requirements specified in **13.10**, in addition to the following **(1)** and **(2)**:

(1) Starting arrangements for main propulsion machinery are to be provided with two sets of hydraulic systems.

(2) The capacity of the hydraulic power pack is to be sufficient (without recharging) to provide the number of consecutive starts specified in **-2** above within 30 *minutes*.

4.4.4 Ignition Arrangements

1 Each ignition arrangement is to consist of two or more systems independent of each other.

2 Cables of an electric ignition device are to be arranged so that satisfactory electrical insulation is ensured and the cables are not likely to be damaged.

3 Ignition distributors are to be of an explosion-proof construction or are to be provided with proper shielding. No coils for any ignition device are to be situated in areas where explosive gases may accumulate.

4.4.5 Fuel Oil Arrangements

1 Sufficient consideration is to be given to the prevention of any clogging of fuel manifolds and fuel nozzles due to solids contained in the fuel and to the prevention of any corrosion of turbine blades and other parts due to corrosive substances such as salts.

2 The fuel control system is to comply with the following requirements:

(1) The fuel control system is to be capable of adjusting the fuel supply to the burners so as to maintain the exhaust gas temperature within the pre-determined range throughout the normal operation.

(2) The fuel control system is to be capable of ensuring stable combustion throughout the operation range where the fuel supply is adjustable.

(3) The fuel control system is to be capable of maintaining the minimum speed of the turbines without stopping the gas generator in the case of sudden load fluctuations.

(4) In dual-fuel applications, provision is to be made for automatic isolation of both primary and standby fuel supplies in the event of a fire.

3 The fuel oil arrangements are also to comply with the requirements in **13.9**, **Part D** and **4.2.2**, **Part R**.

4.4.6 Lubricating Oil Arrangements

1 Gas turbines used as main propulsion machinery are to be provided with an effective emergency supply of lubricating oil which comes into service automatically and has sufficient amount to ensure adequate lubrication until the turbine is brought to rest after a shutdown of the fuel oil supply in the event of a failure of the lubricating oil supplying system. For this purpose, a gravity tank or from an auxiliary lubricating oil pump driven by the turbine may be used.

2 An oil sampling valve is to be provided at a proper location.

3 Lubricating oil arrangements are also to comply with the requirements in **13.10**, **Part D** and **4.2.3, Part R**.

4.4.7 Automatic Temperature Controls

The gas turbine services specified in the following **(1)** to **(3)** are to be fitted with automatic temperature controls so as to maintain steady state conditions throughout the normal operating range of the main gas turbine.

(1) Lubricating oil supply

(2) Oil fuel supply (or automatic control of oil fuel viscosity as alternative)

(3) Exhaust gas

4.4.8 Cooling Arrangements

Gas turbines are to be provided with cooling arrangements as required, and arrangements are to be provided so that the design temperature is not exceeded.

4.5 Tests

4.5.1 Shop Tests*

1 For gas turbines and their accessories hydrostatic tests are to be carried out at pressures specified below.

(1) Casing: 1.5 times the maximum design pressure.

(2) Piping system: Pressures specified in section **12.6**.

2 For rotating assemblies of turbines and compressors, dynamic balancing tests are to be carried out after their assembly.

3 For turbine rotors, excess speed tests are to be carried out at 115% or greater of the maximum continuous rotational speed for at least 2 *minutes* after completion of manufacture. When the Society recognizes that the rotational speed does not exceed 115% of the maximum continuous rotational speed, tests may be carried out at 115%.

4 For gas turbines, shop trials are to be carried out, including the test of safety devices specified in **4.3** above, by procedures deemed appropriate by the Society. In this case, the Society may request tests regarding the starting characteristics and critical speeds of rotor shafts.

Chapter 5 POWER TRANSMISSION SYSTEMS

5.1 General

5.1.1 Scope

The requirements of this Chapter apply to power transmission systems which transmit power from main propulsion machinery and prime movers driving generators and auxiliaries (hereinafter referred to in this Chapter as all auxiliaries excluding auxiliary machinery for specific use etc.).

5.1.2 Drawings and Data

Drawings and data to be submitted are generally as follows:

(1) Drawings:
- (a) Sectional assembly
- (b) Gears
- (c) Gear shafts
- (d) Couplings
- (e) Construction of main parts such as clutches and flexible shafts

(2) Data:
- (a) Specifications for materials used in power transmission parts (chemical compositions, heat treatment methods, mechanical properties and their test methods)
- (b) Transmitted power and number of revolutions per minute for each pinion at maximum continuous output
- (c) Particulars of each gear (number of teeth, module, pitch circle diameters, pressure angles of teeth, helix angles, face widths, centre distances, tool tip radius, backlash, amount of profile shift, amount of profile and tooth trace modification, finishing method of tooth flank, expected finishing accuracy of gears)
- (d) Welding methods of principal components (including tests and inspection)
- (e) Necessary data for the strength calculation of principal components of the power transmission systems.

5.2 Materials and Construction

5.2.1 Materials

1 Materials used for the following components (hereinafter referred to as "the principal components of the power transmission system") are to comply with the requirements in **Part K**.

(1) Power transmission shafts(including power take-off (PTO) shafts) and gears

(2) Power transmission parts of couplings

(3) Power transmission parts of clutches

(4) Coupling bolts

2 The principal components of power transmission systems (excluding coupling bolts, clutch discs and the like) are to have been subjected to the non-destructive tests specified in **5.1.10** and **6.1.10, Part K**.

5.2.2 Welding

Where the principal components of power transmission systems are of welded construction, the requirements in **Chapter 11** are to be complied with.

5.2.3 General Construction of Gearings

1 Gears are to comply with the requirements in the following **(1)**, **(2)** and **(3)**.

(1) Where a gear rim is shrunk on the boss, the rim is to be thick enough to ensure sufficient strength and is to have enough shrinkage allowance against transmitted power. Where the shrinkage fit is made after tooth cutting, construction is to be such as to fully guarantee the accuracy of gearing, or the final tooth finishing is to be carried out after the shrinkage fit.

(2) Where gears are of welded construction, they are to have sufficient rigidity and are to be stress-relieved before tooth cutting.

(3) Gears are not to be of a harmful unbalanced weight.

2 Gear casings are to have sufficient rigidity and their construction is to be such as to allow inspection and maintenance to be preformed as easily as possible.

3 In cases where heavy articles are intended to be fitted onto the extended part of the pinion shaft, the construction of pinions is to be such that the whirling movement of the pinions and the deviation of the shaft centre may be minimized.

5.2.4 General Construction of Power Transmission Systems other than Gearings*

1 Power transmission systems other than the gearings are to be of constructions and materials that have been previously approved by the Society, function safely and reliably and having sufficient strength against transmitted power. Where rubber couplings are used, they are to be appropriate for their conditions of use for heating due to hysteresis.

2 The construction of electro-magnetic slip couplings is to conform to the requirements in **2.4, Part H** as well as to any other requirements deemed appropriate by the Society.

3 Where the clutch of power transmission systems for main propulsion is operated by a hydraulic or pneumatic system, a stand-by pump or compressor that is connected and ready for use at anytime or some other appropriate unit is to be provided in order to ensure that a ship can maintain its normal service condition.

4 Where rubber couplings are used, consideration is to be given to the heat emission of the rubber elements and they are to be constructed so that inspections can be preformed as easily as possible.

5.2.5 Lubricating Oil Arrangements

1 Lubricating oil arrangement is to comply with the requirements in **13.10**. Additionally, it is recommended to use strainers with magnets for gearings.

2 The lubricating oil arrangements of power transmission systems with the driving units above 37 kW are to be provided with alarm devices which give visible and audible alarms in the event of a failure of the supply of lubricating oil or an appreciable reduction in lubricating oil pressure.

5.3 Strength of Gears

5.3.1 Application*

The requirements in **5.3** apply to external tooth cylindrical gears having an involute tooth profile. All other gears are to be as deemed appropriate by the Society. In addition, enclosed gear strength calculations are to be in accordance with **Annex 5.3.1** "**Calculation of Strength of Enclosed Gears**".

5.3.2 General Requirements

1 The fillets between the roots of the teeth are to be as smooth and have as large of a radius as possible. It is recommended that the tooth tip and the both ends of the tooth trace are suitably chamfered.

2 Gears, which are subjected to a surface hardening process, are to have necessary flank hardness and depth of hardened zone.

5.3.3 Allowable Tangential Loads for Bending Strength*

The tangential loads on gear-teeth are to satisfy the following condition for bending strength at the root section of gear-teeth:

$$P_{MCR} \leq 9.81(K_1 S_b - K_2)K_3\left(4.85 - \frac{30.6}{Z}\right)m_n$$

where

P_{MCR} : Tangential load on gear-teeth at the maximum continuous output. Given by the following formula:

$$P_{MCR} = \frac{1.91H}{N_0 D_1 b} \times 10^6 \ (N/cm)$$

H : Output which the pinion shares at maximum continuous output (kW)

N_0 : Number of revolutions of the pinion at maximum continuous output (rpm)

D_1 : Pitch circle diameter of the pinion (cm)

b : Effective face width of the gears on the pitch circle of the shaft parallel section (cm)

Z : Number of teeth

m_n : Rectangular module of tooth

K_1 : External load magnification factor. Determined by the amount of fluctuating loads working on the gears and given by the following formula:

$$K_1 = \frac{1.10 P_{MCR}}{P_{MAX}}$$

P_{MAX} : Instantaneous maximum tangential load occurring within the service revolution range (*N/cm*).

Where, however, the value K_1 is unknown, the values in **Table D5.1** may be used.

K_2 : Internal load magnification value. This value depends on the accuracy of gears and their overlap ratio and can be derived either from the following formula or from **Fig. D5.1**.

$$K_2 = k_2 (Dn)^{0.8}$$

D : Pitch circle diameter of gears (*cm*)

n : Number of revolutions per minute of gears divided by 1,000.

k_2 : Value given in **Table D5.2.** In this case, ε_{sp} is the value derived from the following formula:

$$\varepsilon_{sp} = \frac{b_e \sin\beta_0}{0.1\pi m_n}$$

b_e : Face width (in the case of double-helical gears, the face width is that of a single side) (*cm*)

β_0 : Helical angle

K_3 : Load magnification coefficient due to flexibility. This value depends on the face width and pitch circle diameter and is given either by the following formula or by **Fig. D5.2.**

$$K_3 = 1 - k_3 \left(\frac{b_t}{D_1}\right)^3$$

b_t : Total face width of pinions (in case of double-helical gears, the central gap is included) (*cm*)

D_1 : Pitch circle diameter of the pinion (*cm*)

k_3 : Value given in **Table D5.3**

S_b : This value depends mainly on the material of gears and is given by the following formula. However, in the case of ahead idle gears and astern gears, the values of S_b are to be 0.7 times and 1.2 times respectively. In this case, S_b is not to exceed 25.

(1) In the case of gears, including bottom land, to which a surface hardening process was applied:

$$S_b = 0.83\sqrt{T}$$

(2) In the case of other gears:

$$S_b = \frac{\frac{T+Y}{49}}{1 + (0.0096T - 2.4)\left(\frac{0.04}{r_0} + 0.02\right)(0.023 m_n + 0.75)}$$

T : Specified tensile strength of gear material (*N/mm*^{2})

Y : Specified yield strength of gear material (*N/mm*^{2})

r_0 : Ratio of tool tip radius to module

Table D5.1 Values of K_1 [3][4]

Driving unit	Construction Kind of coupling	Use Gear for main propulsion	Gear for auxiliaries
Steam turbine Gas turbine Electric motor	Single-stage reduction gear Multiple-stage reduction gear	1.00 1.00[1], 1.10[2]	1.15 1.15
Reciprocating internal combustion engine	Hydraulic or electromagnetic coupling High elastic coupling Elastic coupling	1.00 0.90 0.80	1.15 1.05 0.95

Notes:

(1) Applicable only to gearing connected directly to the main propulsion shafting system.

(2) Applicable to gearing connected, through effective flexible couplings, to the propulsion shafting system.

(3) Where one pinion meshes with more than two wheels, 0.9 *times* these values may be used for the value of K_1.

(4) The value of K_1 for rigid couplings is to be approved by the Society.

Table D5.2 Values of k_2

Expected accuracy	$\varepsilon_{sp} \geq 1.25$	$\varepsilon_{sp} < 1.25$
Those correspond to finishing shaved or ground	0.044	0.088
Those correspond to finishing hobbed	0.11	0.22

Table D5.3 Values of k_3

	When one pinion meshes with one wheel	When two wheels mesh with one pinion in the positions forming a row
k_3	0.01	0.003

Fig. D5.1 Values of K_2

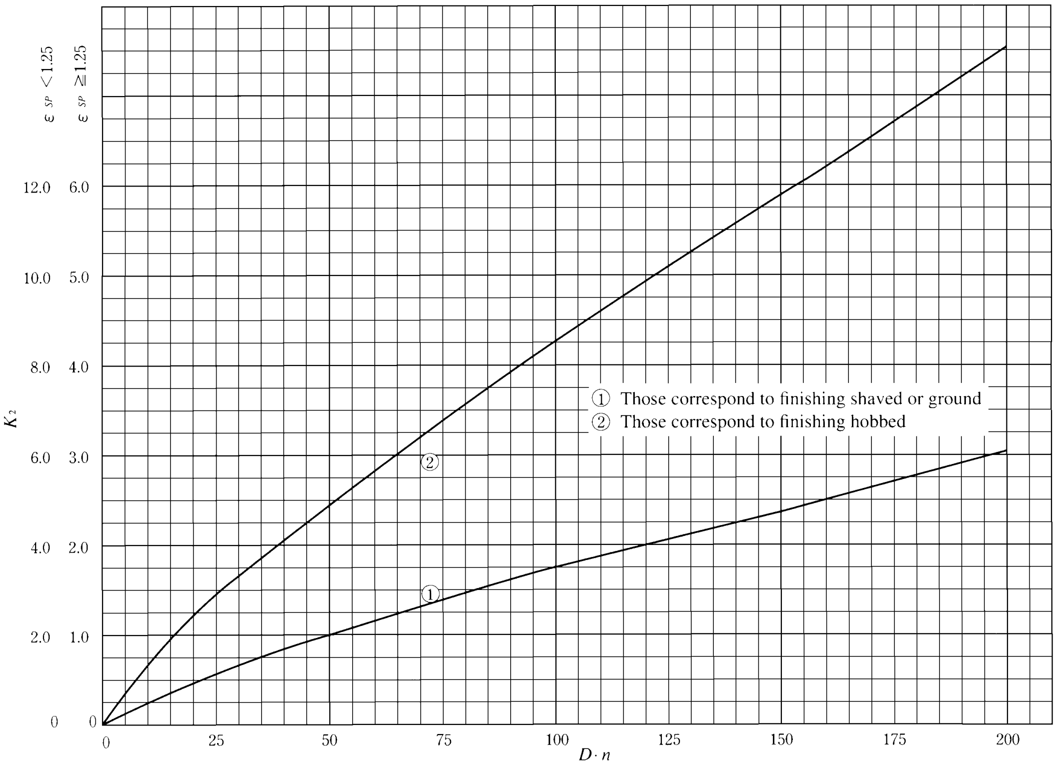
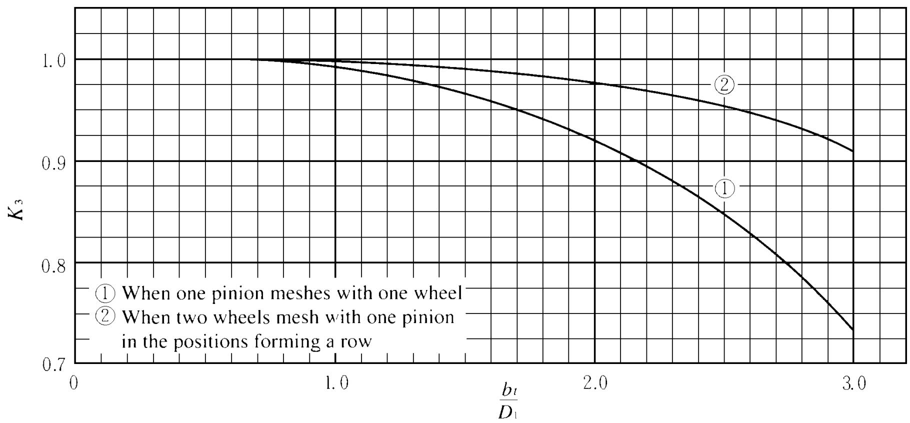


Fig. D5.2 Values of K_3



5.3.4 Tangential Loads for Surface Strength*

The tangential loads on gear-teeth are to satisfy the following condition for limiting tooth surface stress, but these do not apply to astern gears.

$$P_{MCR} \leq 9.81(K_1 S_S - K_2) K_3 K_4 \frac{i}{1+i} D_1$$

where

S_S : The value related mainly to the material of gears, given by the following formula:

(1) Combination of hardened gear

$$S_S = 2.23\sqrt{T_w}$$

(2) Combination of other gears

$$S_S = (0.005\frac{H_{BP}}{H_{BW}} + 0.007)T_w + 7.5$$

H_{BP} : Hardness of the tooth face of the pinion (Brinell hardness)

H_{BW}: Hardness of the tooth face of the wheel (Brinell hardness)

T_w : Specified tensile strength of wheel material (N/mm^2)

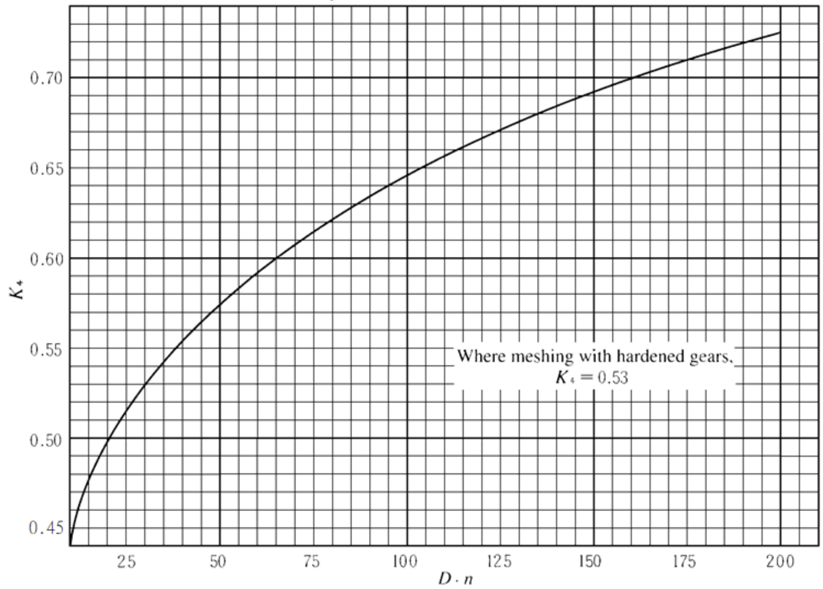
K_4 : Lubricating coefficient. This value depends on the pitch circle diameter and the number of revolutions per minute and is given either by the following formula or by **Fig. D5.3**. However, in the case of a combination of hardened gears, $K_4 = 0.53$

$$K_4 = 0.3(Dn)^{\frac{1}{6}}$$

i : Gear ratio (the number of teeth of the wheel divided by the number of teeth of the pinion)

Other symbols are same as in **5.3.3**.

Fig. D5.3 Values of K_4



5.3.5 Detailed Evaluation for Strength*

Special consideration will be given to the gearing devices, notwithstanding the requirements in **5.3.3** and **5.3.4**, provided that detailed data and calculations on their strength are submitted to the Society and considered appropriate. In addition, the wording "detailed data and calculations on their strength" means calculations based on **Annex 5.3.1** "**Calculation of Strength of Enclosed Gears**".

5.4 Gear Shafts and Flexible Shafts

5.4.1 Gear Shafts

1 The diameter of gear shafts is to comply with the following requirements specified in **(1)** to **(3)**:

(1) The diameter of a gear shaft by which power is transmitted is not to be less than the value given by the formula in **6.2.2**. In this case, H and R in the formula represent respectively the output and the number of revolutions per minute of the shaft at the maximum continuous rating.

(2) The diameter of the pinion shaft between the pinion shaft bearings is to have sufficient rigidity against the bending force generated by the meshing of gears.

(3) The diameter of the wheel shaft between the wheel shaft bearings is not to be less than 1.16 times the value specified in **(1)**, when one pinion is gearing, or two pinions which are arranged at an angle less than 120 degrees are gearing and not to be less than 1.10 times the value specified in **(1)** when two pinions, which are arranged at an angle more than 120 degrees, are gearing.

2 Special consideration will be given to the gear shaft, notwithstanding the requirements in **-1**, provided that detailed data and calculations on the strength are submitted to the Society and considered appropriate.

5.4.2 Flexible Shafts

The diameter of a flexible shaft is not to be less than the value given by the following formula:

$$d = 93\sqrt[3]{\frac{560H}{N_0(T+160)}}$$

where

d : Diameter of the flexible shaft (mm)

H : Output which the flexible shaft shares at maximum continuous output (kW)

N_0 : Number of revolutions of the flexible shaft at maximum continuous output (rpm)

T : Specified tensile strength of the shaft material (N/mm^2)

5.4.3 Couplings and Coupling Bolts

The dimensions of couplings and coupling bolts are to be of values not less than those obtained from the formula given in **6.2.12-1** in this Part. Furthermore, in cases where they support heavy materials in cantilever style, couplings and coupling bolts are to be designed so as to have sufficient strength to resist such weight. In addition, in the formula referred to above, d_o is the value of the shaft diameter that has been calculated according to each kind of shaft.

5.5 Tests

5.5.1 Shop Tests

1 For the parts subjected to surface hardening process, the measurement of the hardened depth is to be carried out on sample materials.

2 For parts subjected to a surface hardening process, hardness tests and non-destructive tests by suitable procedures are to be carried out.

3 For gears, accuracy tests to examine the machining accuracy of finish are to be carried out.

4 In the case of gears where the value given by the following formula exceeds 50, dynamic balancing tests are to be carried out.

$$\frac{DN_0}{1000}$$

where

D : Pitch circle diameter of gear (cm)

N_0 : Number of rotations of gear (rpm)

5 The contact marking of the teeth of all gearings is to be tested with a thin uniform coat of suitable paint under an appropriate load.

Chapter 6 SHAFTINGS

6.1 General

6.1.1 Scope

The requirements of this Chapter apply to propulsion shafting systems (excluding propellers) and shafting systems which transmit power from prime movers to drive generators and auxiliaries (hereinafter referred to in this Chapter as all auxiliaries excluding auxiliary machinery for specific use etc.). For torsional vibrations, the requirements in **Chapter 8** are to be complied with.

6.1.2 Drawings and Data

Drawings and data to be submitted are generally as follows:

(1) Drawings for approval (including specifications of material)

(a) Shafting arrangement

(b) Thrust shaft

(c) Intermediate shaft

(d) Stern tube shaft

(e) Propeller shaft

(f) Stern tube

(g) Stern tube bearing; this drawing may be included in the drawings and data specified in **(l)** in the case of propeller shafts Kind 1*C*.

(h) Stern tube sealing device; this drawing may be included in the drawings and data specified in **(l)** in the case of propeller shafts Kind 1*C*.

(i) Shaft bracket bearing

(j) Shaft couplings and coupling bolts

(k) Shafts which transmit power to generators or auxiliaries

(l) In the case of propeller shafts Kind 1*C*, four sets of drawings and data of the following **i)** to **viii)**:

i) Specifications for the devices and equipment required in **6.2.11**

ii) Tables of monitoring, control, and alarm items (including bearing temperatures, tank levels, seal air pressure, switchover of pumps, changeover of tanks (high or low) and changeover of spare seal rings), and their settings and indication methods

iii) Documents for countermeasures after alarms have been activated (including necessary operations such as valve handling after alarming)

iv) Drawings of stern tube bearings

v) Drawings of stern tube sealing devices

vi) Piping diagrams (including those related to the height of sensing positions for monitoring and controlling tank levels, systems for monitoring and controlling seal air pressure, and explanations of each symbol for sensors, switches, valves and other fittings and valve operations)

vii) Specification sheets for allowable ranges of pressure or tank levels for stern tube sealing devices determined by the manufacturer

viii) Shaft alignment calculation sheets in accordance with **Annex 6.2.13**.

(2) Data for reference

(a) Data for the calculations of shafting strength specified in this Chapter

(b) Data which is deemed necessary by the Society

6.2 Materials, Construction and Strength

6.2.1 Materials

1 Materials used for the following components (hereinafter referred to as "the principal components of shafting") are to be of steel forgings conforming to the requirements specified in **6.1 of Part K**; of stainless steel forgings conforming to the requirements specified in **6.2 of Part K**; of rolled stainless steel bars approved for shaft use conforming to the requirements specified in **3.5.1-2 of Part K** (hereinafter, stainless steel forgings and rolled stainless steel bars are to be referred to as "stainless steel forgings, etc."); or of material specially approved for shaft use by the Society under **1.1.1-3 of Part K**. Built-up type shaft couplings may be of steel castings that conform to the requirements in **Part K**.

(1) Thrust shafts
(2) Intermediate shafts
(3) Stern tube shafts
(4) Propeller shafts
(5) Shafts which transmit power to generators or auxiliaries
(6) Shaft couplings
(7) Coupling bolts

2 Depending on the type of material being used, the principal components of shafting, excluding the coupling bolts, are to have been subjected to the non-destructive tests specified either in **5.1.10**, **6.1.10** or **6.2.10 of Part K**.

3 The specified tensile strength of the shaft materials is generally to be between 400 and 760 N/mm^2 and to be between 500 and 760 N/mm^2 for shafts experiencing torsional vibration stress that exceeds 85 % of the value for τ_2 given in **8.2.2**.

Steel forgings with a specified tensile strength exceeding 760 N/mm^2 are not to be used for any shafts unless specially approved by the Society. For alloy steel castings, the value "760 N/mm^2" is to be read as "1100 N/mm^2".

6.2.2 Intermediate Shafts*

1 The diameter of the intermediate shafts made of steel forgings (excluding stainless steel forgings, etc.) is not to be less than the value given by the following formula:

$$d_0 = F_1 k_1 \cdot \sqrt[3]{\frac{H}{N_0}\left(\frac{560}{T_s + 160}\right)K}$$

where

d_0 : Required diameter of intermediate shaft (mm)

H : Maximum continuous output of engine (kW)

N_0 : Number of revolutions of intermediate shaft at maximum continuous output (rpm)

F_1 : Factor given in **Table D6.1**

k_1 : Factor given in **Table D6.2**

T_s : Specified tensile strength of intermediate shaft material (N/mm^2)

The upper limit of the value of T_s used for the calculation is to be 760 N/mm^2 for carbon steel forgings and 800 N/mm^2 for low alloy steel forgings. The upper limit of the value of T_s used for the calculation may be increased to 950 N/mm^2 when intermediate shafts are manufactured using steel forgings (excluding stainless steel forgings) which have specified minimum tensile strengths greater than 800 N/mm^2 and are in accordance with **Annex 6.2.2** "**Use of High-Strength Materials for Intermediate Shafts**".

K : Factor for hollow shaft and given by the following formula. In cases where $d_i \leq 0.4 d_a$, it may be considered that $K = 1$

$$K = \frac{1}{1 - \left(\frac{d_i}{d_a}\right)^4}$$

where

d_i : Inside diameter of hollow shaft (mm)

d_a : Outside diameter of hollow shaft (mm)

2 The diameter of the intermediate shaft of material other than specified in **-1** above is to be deemed appropriate by the Society.

Table D6.1 Values of F_1

In cases where steam turbines in or gas turbines are used as main propulsion machinery, or in the case of reciprocating internal combustion engines with slip type coupling(1) or electric propulsion	For all other reciprocating internal combustion engines than those noted in the left hand column
95	100

Note:

Slip type coupling signifies hydraulic coupling, electromagnetic coupling or the equivalent.

Table D6.2 Values of k_1

Shaft with integral flange coupling (1)	Shaft with flange coupling either shrink fit, push fit or cold fit (2)	Shaft with keyway (3)(4)	Shaft with transverse hole (5)	Shaft with longitudinal slot (6)	Shaft with splines (7)
1.0	1.0	1.1	1.1	1.2	1.15

Notes:

(1) The fillet radius at the base of the flange is not to be less 0.08 *times* the diameter of the shaft.

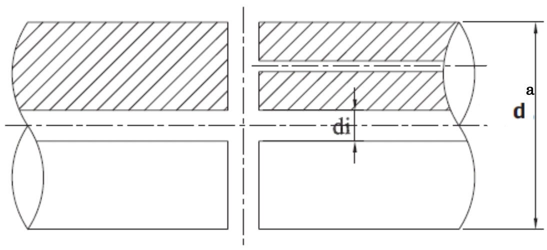
(2) In cases where shafts, during continuous operation, experience torsional vibration stress exceeding 85 % of τ_1 given in **8.2.2-1(1)**, an increase of 1 to 2 % in diameter to the fit diameter and a blending radius nearly equal to the change in diameter are to be provided.

(3) After a length of not less than 0.2 d_0 from the end of the keyway, the diameter of a shaft may be reduced progressively to the diameter calculated with k_1=1.0.

The fillet radius in the transverse section of keyway bottom is to be 0.0125 d_0 or more.

(4) Keyways are in general not to be used in installations with a barred speed range in accordance with **8.3**.

(5) The diameter of the hole is not to be more than 0.3 d_0. When a transverse hole intersects an eccentric axial hole (see below), the value is to be determined by the Society based on the submitted data in each case.



(6) The shape of the slot is to be in accordance with the following: any edge rounding other than by chamfering is to be avoided in principle; the number of slots is to be 1, 2 or 3 and they are to be arranged 360, 180 or 120 *degrees* apart from each other respectively.

(a) $l < 0.8d_a$

(b) $d_i < 0.7d_a$

(c) $0.15d_a < e \leqq 0.2d_a$

(d) $r \geqq e/2$

where

l : slot length

d_a : outside diameter of the hollow shaft

d_i : inside diameter of the hollow shaft

e : slot width

r : end rounding of the slot

(7) The shape of the spline is to conform to *JIS B* 1601 or the equivalent thereof.

6.2.3 Thrust Shafts

1 The diameter of the thrust shaft transmitting the torque of main propulsion machinery, and which is made of steel forgings (excluding stainless steel forgings, etc.), on both sides of the thrust collar, or in way of the axial bearing where roller bearings are used as thrust bearings, is not to be less than the value given by the following formula:

$$d_t = 1.1F_1\sqrt[3]{\frac{H}{N_0}\left(\frac{560}{T_s+160}\right)K}$$

where

d_t : Required diameter of thrust shaft (mm)

Other symbols used here are the same as those used in **6.2.2-1**.

2 In cases where the required diameter of the thrust shaft given by **-1 above** is larger than the diameter of the intermediate shaft, the diameter of the thrust shaft may be reduced gradually at either fwd or aft of the thrust block by multiplying the required diameter of the thrust shaft given by **-1 by** 0.91.

3 The fillet radius at the base of the thrust collar on both sides is not to be less 0.08 *times* the diameter of the shaft.

4 The diameter of the thrust shaft of material other than specified in **-1** above is to be deemed appropriate by the Society.

6.2.4 Propeller Shafts and Stern Tube Shafts*

1 The diameters of propeller shafts and stern tube shafts made of carbon steel forgings or low alloy steel forgings are not to be less than the value given by the following formula. However, in cases where the propeller shaft is Kind 2 or the stern tube shaft is Kind 2, the diameters are to be deemed appropriate by the Society.

$$d_s = 100k_2\sqrt[3]{\frac{H}{N_0}\left(\frac{560}{T_s+160}\right)K}$$

where

d_s : Required diameter of propeller shaft or stern tube shaft (mm)

k_2 : Factor concerning shaft design. Values given in **Table D6.3**

T_s : Specified tensile strength of shaft material (N/mm^2)

The upper limit of the value of T_s used in this calculation is to be 600 N/mm^2

Other symbols used here are the same as those used in **6.2.2-1**

2 The diameters of propeller shafts and stern tube shafts made of stainless steel forgings, etc. are not to be less than the value given by the following formula:

$$d_s = 100k_3\sqrt[3]{\frac{H}{N_0}}$$

k_3 : Factor concerning the shaft material and shaft portion, which is given in **Table D6.4**. Material other than that specified in the table is to be determined by the Society in each case.

Other symbols used here are the same as those used in **6.2.2-1**.

3 The shaft diameter may be reduced by either a smooth taper or a blending radius nearly equal to the change in diameter to the diameter calculated by the formula given in **6.2.2-1** at the portions located forward of the fore end of the fwd stern tube seal. In cases where shafts are manufactured using stainless steel, shaft diameters calculated as $T_s = 400$ are to be used.

4 The diameters of propeller shafts and stern tube shafts other than those prescribed in **-1** and **-2** are to be deemed appropriate by the Society.

Table D6.3 Values of k_2

	Application		k_2
1	The portion between the big end of the tapered part of propeller shaft (in cases where the propeller is fitted with a flange, the fore face of the flange) and the fore end of the aftermost stern tube bearing, or 2.5 d_S, whichever is greater	For a shaft carrying a keyless propeller, or where the propeller is attached to an integral flange	1.22
		For a shaft carrying a keyed propeller	1.26
2	Excluding the portion given in 1 above, the portion up to the fore end of the fwd stern tube seal in the direction of the bow		1.15(1)
3	Stern tube shaft		1.15(1)
4	The portion located forward of the fore end of the fwd stern tube seal		1.15

Notes:

(1) At the boundary, the shaft diameter is to be reduced with either a smooth taper or a blending radius nearly equal to the change in diameter.

Table D6.4 Values of k_3

	Application	*KSUSF* 316 *KSUS*316-*SU*	*KSUSF* 316*L* *KSUS*316*L*-*SU*
1	The portion between the big end of the tapered part of propeller shaft (in cases where the propeller is fitted with a flange) and the fore end of the aftermost stern tube bearing, or 2.5 d_S, whichever is greater	1.28	1.34
2	Excluding the portion given in 1 above, the portion up to the fore end of the fwd stern tube seal in the direction of the bow	1.16(1)	1.22(1)
3	The portion located forward of the fore end of the fwd stern tube seal	1.16	1.22

Notes:

(1) At the boundary, the shaft diameter is to be reduced with either a smooth taper or a blending radius nearly equal to the change in diameter.

6.2.5 Other Shafts

The diameter of shafts transmitting power to generators or essential auxiliary machinery is, in principle, to conform to the requirements in **6.2.2**.

6.2.6 Detailed Evaluation for Strength*

Special consideration will be given to the shaft diameters, notwithstanding the requirements in **6.2.2**, **6.2.3**, **6.2.4** and **6.2.5**, provided that the detailed data and calculations are submitted to the Society and considered appropriate.

6.2.7 Corrosion Protection of Propeller Shafts and Stern Tube Shafts*

1 Propeller shafts Kind 1 and stern tube shafts Kind 1 are to be effectively protected against corrosion by water_(sea water, outboard freshwater and inboard freshwater. The same is referred to hereinafter in this Chapter) with the means specified in the following **(1)** to **(3)**, as applicable.

(1) to effectively protect the propeller shafts and stern tube shafts against any contact with water by the means approved by the Society

(2) to use *KSUSF*316, *KSUSF*316*L*, *KSUS*316-*SU* or *KSUS*316*L*-*SU* specified in **Part K** for shafts with diameter not exceeding 200 *mm*

(3) to use corrosion resistant materials approved by the Society other than those specified in **(2)** above

2 Effective means are to be provided to prevent water from having access to the part between the aft end of propeller shaft sleeve or the aft end of the aftermost stern tube bearing and the propeller boss.

3 Spaces between the propeller cap or propeller boss and the propeller shaft are to be filled up with grease or provided with other effective means to protect the shaft against corrosion by water.

6.2.8 Propeller Shaft Sleeves and Stern Tube Shaft Sleeves

The sleeves to be fitted to a propeller shaft and a stern tube shaft are to comply with the requirements in the following **(1)** to **(3)**:

(1) The thickness of the sleeve is not to be less than the value given by the following formula:

$$t_1 = 0.03d_S + 7.5$$

$$t_2 = \frac{3}{4}t_1$$

where

t_1 : Thickness of sleeve in way of stern tube bearing or shaft bracket bearing in contact with the bearing face (*mm*)

t_2 : Thickness of sleeve of other parts than the above (*mm*)

d_S : Required diameter of propeller shaft given by the formula in **6.2.4** (*mm*)

(2) Sleeves are to be of bronze or equivalent thereof and to be free from porosity and other defects.

(3) Sleeves are to be fitted to the shafts by a method free from stress concentration such as shrinkage fit, etc.

6.2.9 Fixing of Propellers to Shafts

1 In cases where propellers are force fitted onto the propeller shaft, the fixing part is to be of sufficient strength against the torque to be transmitted.

2 In cases where a key is provided to the fixing part, ample fillets are to be provided at the corners of the keyway and key is to have a true fit in the keyway. The fore end of keyway on the propeller shaft is to be rounded smoothly in order to avoid any excessive stress concentration.

3 In cases where the propeller and propeller shaft flange are connected with bolts, the bolts and pins are to be of sufficient strength.

4 The thickness of the aft propeller shaft flange at the pitch circle is not to be less than 0.27 times the diameter of the intermediate shaft (calculated with k_1 = 1.0, K = 1.0 and T_S = 400) in **6.2.2**.

6.2.10 Stern Tube Bearings and Shaft Bracket Bearings*

1 The aftermost stern tube bearing or shaft bracket bearing which supports the weight of propeller is to comply with the following requirements **(1)** to **(3)**:

(1) In the case of oil lubricated bearings.

(a) In the case of white metal

i) The length of the bearing is not to be less than twice the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**. However, where the nominal bearing pressure (determined by the static bearing reaction calculation taking into account shaft and propeller weight which is deemed to be exerted solely on the aft bearing divided by the projected area of the shaft in way of the bearing, hereinafter defined the same way in this chapter) is not more than 0.8 *MPa* and special consideration is given on the construction and arrangement in accordance with provisions specified elsewhere, the length of the bearing may be fairly shorter than that specified above. However, the minimum length is to be not less than 1.5 *times* the actual diameter of the propeller shaft.

ii) The stern tube is to be always filled with oil. Adequate means are to be provided to measure the temperature of oil in the stern tube.

iii) In cases where a gravity tank supplying lubricating oil to the stern tube bearing is fitted, it is to be located above the load water line and provided with a low level alarm device. However, in cases where the lubricating system is designed to be used under the condition that the static oil pressure of the gravity tank is lower than the water pressure, the tank is not required to be above the load water line.

iv) The lubricating oil is to be cooled by submerging the stern tube in the water of the after peak tank or by some other suitable means.

(b) In the case of materials other than white metal

i) The materials, construction and arrangement are to be approved by the Society.

ii) For bearings of synthetic rubber, reinforced resin or plastics materials which are approved for use as oil lubricated stern tube bearings, the length of the bearing is to be not less than twice the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**. However, where nominal bearing pressure is not more than 0.6 *MPa* and bearings have a construction and arrangement in accordance with provisions specified elsewhere, the length of the bearing may be fairly shorter than that specified above. However, the minimum length is to be not less than 1.5

times the actual diameter of the propeller shaft.

iii) Notwithstanding the requirement given in **ii)**, the Society may allow use of bearings whose nominal bearing pressure is more than 0.6 *MPa* where the material has proven satisfactory testing and operating histories.

(2) In the case of water lubricated bearings

(a) The materials, construction and arrangement are to be approved by the Society.

(b) The length of the bearing is to be not less than 4 *times* the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**, or 3 *times* the actual diameter, whichever is greater. However, for bearings of synthetic materials, such as rubber or plastics, that are approved for use as water lubricated stern tube bearings and where special consideration is given to their construction and arrangement in accordance with provisions specified elsewhere, the length of the bearing may be fairly shorter than that specified above. However, minimum length is to be not less than twice the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**, or 1.5 *times* the actual diameter, whichever is greater.

(3) In the case of grease lubricated bearings

In cases where the actual diameter of the propeller shaft is not more than 100 *mm*, grease lubricated bearings may be used. The length of the bearing is to be not less than 4 *times* the required diameter of the propeller shaft given by the formulae in either **6.2.4-1** or **-2**.

2 Sealing devices, other than gland packing type water sealing devices, are to be approved by the Society with regards to materials, construction and arrangement.

6.2.11 Additional Requirements for Propeller Shaft Kind 1*C**

Means are to be provided to sufficiently ensure the integrity of the stern tube bearings in accordance with the following **(1)** to **(4)** requirements where the propeller shaft is intended to be a propeller shaft Kind 1*C*.

(1) One set of the drawings and data listed in **6.1.2(1)(l)** which has been approved and returned is to be kept on board.

(2) Stern tube bearings are to comply with the following **(a)** to **(c)**:

(a) Either of the following devices to measure stern tube metal temperature at the aft end bottom along with devices to record temperature is to be provided. In addition, audible and visible high temperature alarms (with a preset value of 55 °C or below) are to be provided in main control stations as specified in **18.1.2(3)** or **(4), Part D of the Rules**.

i) Two or more temperature sensors embedded in the metal; or

ii) An embedded temperature sensor, replaceable from inboard the ship, and a spare temperature sensor. In this case, replacement of such sensors according to procedures submitted beforehand is to be demonstrated.

(b) Devices are to be installed for automatically reducing the speed of main propulsion machinery or for issuing audible and visual alarms which requires operation to reduce the running speed of the main propulsion machinery in main control stations in cases where at least one of the sensors specified in **(a)** above detects temperatures higher than preset values. In cases where means are provided to reduce the speed of main propulsion machinery automatically, the override arrangements specified in **18.2.6-3, Part D** are to be provided for bridge control devices.

(c) The adhering strength of stern tube white metal lining is to be not less than 40 *MPa*.

(3) Stern tube sealing devices are to be of such a construction as to permit repairs or replacement without drawing out the propeller shaft not to impair the oil sealing properties and the durability. For this purpose, a two-piece type sealing device is to be used or a distance piece to shift the seal ring contact position is to be provided.

(4) Piping arrangements are to comply with the following **(a)** to **(d)**:

(a) Audible and visual alarms are to be provided in main control stations to show the lubricating oil pressure of the stern tube bearing and the sealing oil (or air) pressure between the seal rings (excluding the pressure of any oil between enclosed seals) are out of their allowable ranges. In cases where the upper limit of the pressure is controlled by means of over flow piping, alarms for the upper limit may be omitted.

(b) As for the lower limit of the allowable range specified in **(a)**, low level alarms in cases where oil tanks are provided; otherwise, non-flow alarms may be used.

(c) The lubricating oil for stern tube bearings is to be incessantly circulated. Two circulating pumps, arranged to be automatically switched over in the event of a pump stopping or a lower delivery pressure than the preset value, are to be provided. Audible and visual alarms, coming into action at the switchover, and a means to indicate which pump is working are to be provided in main control stations.

(d) In cases where the pressure of the lubricating oil for the stern tube bearing or the sealing oil between seal rings of the stern tube sealing device is changed to high or low according to the draught of the ship, means, such as a lamp linked to the valve change operation, to indicate which pressure is working are to be provided at main control stations.

6.2.12 Shaft Couplings and Coupling Bolts

1 The diameter of coupling bolts at the joining face of the couplings is not to be less than the value given by the following formula:

$$d_b = 0.65\alpha\sqrt{\frac{{d_0}^3(T_s + 160)}{nDT_b}}$$

where

d_b : Bolt diameter (mm)

d_0 : Diameter (mm) of intermediate shaft calculated with $k_1 = 1.0$ and $K = 1.0$ in 6.2.2

n : Number of bolts

D : Pitch circle diameter (mm)

T_S : Specified tensile strength of intermediate shaft material taken for the calculation in 6.2.2

T_b : Specified tensile strength of bolt material (N/mm^2), while generally $T_S \leq T_b \leq 1.7T_S$; and, the upper limit of the value of T_b used for the calculation is to be 1,000 N/mm^2

α : Coefficient concerning vibratory torque given by the following formula or to be taken as 1.0, whichever is greater. However, α =1.0 may be accepted for coupling bolts used for shafting systems which transmit power from prime movers to drive generators and auxiliaries.

$$\alpha = 0.95\sqrt[3]{\frac{Q_a}{Q_m}}$$

Q_a : Torsional vibratory torque acting on the joining face of the couplings rotating at resonant critical speed in all conditions (Nm)

Q_m : Nominal rated torque given by the following formula (Nm)

$$Q_m = 9549\frac{H}{N_0}$$

H : Maximum continuous output of engine (kW)

N_0 : Rate of revolutions of intermediate shaft at the maximum continuous output (rpm)

2 The thickness of the coupling flange at the pitch circle is not to be less than the required diameter of the coupling bolt calculated by the formula in -1 for the material having the same tensile strength as the corresponding shaft. However it is not to be less than 0.2 times the required diameter of the corresponding shaft.

3 The fillet radius at the base of the flange is not to be less than 0.08 times the diameter of the shaft, where the fillet is not to be recessed in way of nuts and bolt heads.

4 In cases where the shaft couplings are not integral with the shaft, the couplings are to have sufficient strength against the torque to be transmitted to the shaft and also the astern pull. In this case, consideration is to be taken so as not to cause an excessive, stress concentration.

6.2.13 Shaft Alignment

For the main propulsion shafting having an oil-lubricated propeller shaft of which diameter is not less than 400 mm, the shaft alignment calculation in accordance with **Annex 6.2.13** including bending moments, bearing loads and deflection curve of the shafting is to be submitted to the Society for approval.

6.3 Tests

6.3.1 Shop Tests

The following components are to be subjected to hydrostatic tests at pressures specified below

(1) Stern tubes:

: 0.2 MPa

(2) Propeller shaft sleeves and stern tube shaft sleeves:

: 0.1 MPa (tests are to be carried out before shrinkage fit)

6.3.2 Tests after Installation On Board*

1 The sealing devices specified in **6.2.10-2** are to be tested for leakage under lubricating oil or lubricating freshwater supply pressure after installation on board.

2 For the main propulsion shafting (excluding those of waterjet propulsion systems or azimuth thrusters), confirmation tests relating to shaft alignment are to be carried out in accordance with the requirements specified otherwise by the Society.

3 In the case of propeller shafts Kind 1*C*, the devices and equipment specified in **6.2.11** are to be tested in order to verify the performance of the system in accordance with the items in the table specified in **6.2.1(1)(l)(ii)**.

Chapter 7 PROPELLERS

7.1 General

7.1.1 Scope

The requirements in this Chapter apply to screw propellers.

7.1.2 Drawings and Data

Drawings and data to be submitted are generally as follows:

(1) Drawings
   (a) Propeller
   (b) Operating oil piping diagram of controllable pitch propeller indicating pipe materials, pipe sizes and service pressure
   (c) Blade fixing bolts of controllable pitch propeller

(2) Data
   (a) Particulars of propeller (maximum continuous output and number of maximum continuous revolutions per minute of main propulsion machinery, details of blade profile, diameter, pitch, developed area, propeller boss ratio, rake or rake angle, number of blades, mass, moment of inertia, material specifications, etc.)
   (b) Calculation sheet of propeller pull-up length (where it is proposed to fit keyless propellers)

7.1.3 Materials

1 The materials of propellers and the blade fixing bolts of controllable pitch propellers are to comply with the requirements in **Part K**.

2 Propellers are to have been subjected to non-destructive tests on their principal parts in accordance with **7.2.9, Part K of the Rules**.

7.2 Construction and Strength

7.2.1 Thickness of Blade*

1 The thickness of the propeller blades at a radius of 0.25 R and 0.6 R (where R is the radius of the propeller) for solid propellers and at a radius of 0.35 R and 0.6 R for controllable pitch propellers is not to be less than the values given by the following formula. The thickness of the highly skewed propeller blades is to conform with the provisions specified in **2** below.

$$t = \sqrt{\frac{K_1}{K_2}\frac{H}{ZN_0\ell}SW}$$

where

t : Thickness of blades (excluding the fillet of blade root) (cm)

H : Maximum continuous output of main propulsion machinery (kW)

Z : Number of blades

N_0 : Number of maximum continuous revolutions (rpm) divided by 100

ℓ : Width of blade at radius in question (cm)

K_1 : Coefficient of the radius in question given by the following formula:

$$K_1 = \frac{30.3}{\sqrt{1 + k_1\left(\frac{P'}{D}\right)^2}}\left(k_2\frac{D}{P} + k_3\frac{P'}{D}\right)$$

D : Diameter of propeller (m)

k_1, k_2, k_3 : Values given in **Table D7.1**

P' : Pitch at radius in question (m)

P : Pitch at radius of 0.7 R (m)

K_2 : Coefficient given by the following formula:

$$K_2 = K - \left(k_4 \frac{E}{t_0} + k_5\right) \frac{D^2 N_0^2}{1000}$$

k_4, k_5 : Values given in **Table D7.1**

E : Rake at the tip of the blade (Measuring from face side base line and taking positive value for backward rake) (cm)

t_0 : Imaginary thickness of blade at propeller shaft centreline (t_0 may be obtained by drawing the each side line which connects the blade tip thickness with the thickness at 0.25 R (or 0.35 R for controllable pitch propeller), in the projection of the blade section along the maximum blade thickness line.) (cm)

K : Value depending upon the type of the propeller material given in **Table D7.2**

S : Coefficient concerning the increase in stress during times of bad weather. Where S>1.0 or S<0.8, the value of S is to be taken as 1.0 or 0.8 respectively.

$$S = 0.095 \left(\frac{D_S}{d_S}\right) + 0.677$$

D_S : Depth of ship for strength computation (*See* **2.1.7, Part A**)

d_S : Load draught (*See* **2.1.12, Part A**)

W : Coefficient concerning alternate stress, given by the following formula or to be taken as 2.80, whichever is greater.

$$W = 1 + 1.724 \left(\frac{A_2 A_3 + A_4 A_1 P'/D}{A_3 + A_4 P'/D}\right)$$

$$A_1 = \frac{\Delta w}{w + C_1}$$

$$A_2 = \frac{\Delta w}{w + C_2}$$

$$A_3 = \frac{(C_1 + 1)(C_2 + w)}{C_3 (C_2 + 1)(C_1 + w)}$$

$$A_4 = \begin{cases} 3.52(0.25R) \\ 2.41(0.35R) \\ 1.26(0.6R) \end{cases}$$

$$C_1 = \frac{D}{0.95P} \left\{\frac{P}{D}\left(1.3 - \frac{2a_e}{Z}\right) + 0.22\right\} - 1$$

$$C_2 = \frac{D}{0.95P} \left(1.1 \frac{P}{D} - \frac{1.19 a_e}{Z} + 0.2\right) - 1$$

$$C_3 = 0.122 \frac{P}{D} + 0.0236$$

a_e : Expanded area ratio of propeller

w : Nominal mean wake in the propeller disc

Δw : Peak to peak value of wake fluctuation in the propeller disk at a radius of 0.7 R. The values of w and Δw are to be calculated by using the following formulae, except in the case of multi-screw ships or when expressly approved by the Society.

$$\Delta w = 7.32 \left\{1.56 - 0.04 \left(\frac{B}{D} + 4\right) \sqrt{\frac{B}{d_S}} - C_b\right\} w$$

$$w = 0.625 \left\{0.04 \left(\frac{B}{D} + 4\right) \sqrt{\frac{B}{d_S}} + C_b\right\} - 0.527$$

B : Breadth of ship (m)

C_b : Block coefficient of ship

Table D7.1 Values of k_1, k_2, k_3, k_4 and k_5

Radial position	k_1	k_2	k_3	k_4	k_5
0.25R	1.62	0.386	0.239	1.92	1.71
0.35R	0.827	0.308	0.131	1.79	1.56
0.6R	0.281	0.113	0.022	1.24	1.09

Table D7.2 Values of K

Material		K
Copper alloy castings	$KHBsC1$	1.15
	$KHBsC2$	
	$KAlBC3$	1.3
	$KAlBC4$	1.15
Stainless steel forgings for propellers	$KSCSP1$, $KSCSP2$, $KSCSP3$	1.0
	$KSCSP4$	0.9

Notes:

(1) For the blades of materials different from those specified in the above table, the value of K is to be as deemed appropriate by the Society.

(2) For propellers having a diameter of 2.5 *metres* or less, the value of K may be taken as the value in the above table multiplied by the following factor:

$2-0.4D$ for $2.5 \geq D > 2.0$

1.2 for $2.0 \geq D$

2 The thicknesses of highly skewed propeller blades, depending upon the skew angle (i.e. the angle, on the expanded blade drawing, between the line connecting the centre of the propeller shaft with the point at the blade tip on the centreline of blade width and the tangential line drawn from the centre of the propeller shaft to the centreline of blade width (*See* **Fig. D7.1**)) is to comply with either of the following **(1)** or **(2)**:

(1) In cases where the skew angle exceeds 25 degrees but is 60 degrees or less

(a) The blade thicknesses at radii $0.25R$ ($0.35R$ for a controllable pitch propeller) and $0.6R$ are not to be less than the values obtained from multiplying those values calculated by the formulae in **1** above by coefficient A given below:

$$A = 1 + B\frac{\theta - 25^{\circ}}{60^{\circ}}$$

where

θ : skew angle (*degrees*)

B : 0.2 at $0.25R$ (or $0.35R$ for a controllable pitch propeller) and 0.6 at $0.6R$

(b) Blade thickness t_x at any radius between $0.6R$ and $0.9R$ is not to be less than the value determined by the following formula. Moreover, this thickness is to provide sufficient strength against loads imparted during reversing manoeuvres, etc.

$$t_x = 0.003D + \frac{(1-x)(t_{0.6}-0.003D)}{0.4} \quad (mm)$$

where

D : diameter of propeller (*mm*)

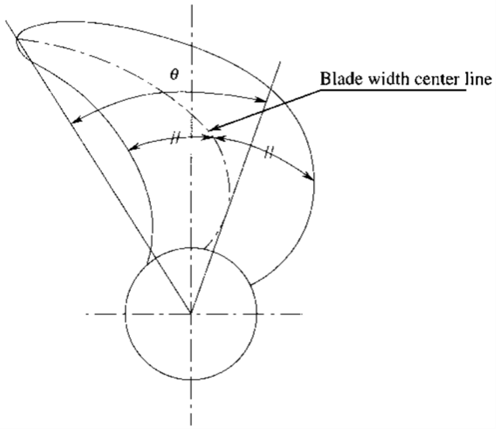
x : ratio of the radius (equals $2r/D$, where r is the radius (*mm*))

$t_{0.6}$: blade thickness at $0.6R$ as required in **(a)** above (*mm*)

(2) In cases where the skew angle exceeds 60 degrees

Based upon the precise calculation sheet of propeller strength submitted by the manufacturer or designer, blade thickness is to be determined by the Society on a case by case basis.

Fig. D7.1 Definition of Skew Angle



3 The fillet radius between the root of a blade and the boss of the propeller, on the pressure side at the maximum blade thickness part, is to be not less than the value of R_0 given by the following formula:

$$R_0 = t_r + \frac{(e - r_B)(t_0 - t_r)}{e}$$

R_0 : Required radius of the fillet (cm)

t_r : Required thickness of blades at a radius of 0.25 R (or 0.35 R for a controllable pitch propeller) specified in **-1** (cm)

t_0 : Same as that used in **-1**

r_B : Boss ratio of propeller

e : 0.25 (or 0.35 for a controllable pitch propeller)

4 Special consideration will be given to the thickness of the blades or the radius of the fillet, notwithstanding the requirements in **-1** to **-3** above, provided that detailed data and calculations are submitted to the Society and considered appropriate.

7.2.2 Controllable Pitch Propellers*

1 The thickness of the controllable pitch propeller blade and the fillet radius between the root of a blade and the boss of the propeller is to be in accordance with the requirements specified in **7.2.1**.

2 The diameter of blade fixing bolts of controllable pitch propellers is not to be less than the value calculated by the following formula. However, in cases where documents deemed appropriate by the Society are submitted and it can be demonstrated that the blade fixing bolts satisfy the strength requirements specified in the Rules, this requirement may be dispensed with.

$$d = 0.55\sqrt{\frac{1}{\sigma_a n}\left(\frac{AK_3}{L} + F_c\right)}$$

where

d : Required diameter of blade fixing bolt (mm) (*See* **Fig. D7.1**)

A : Value given by the following formula, where H, N_0 and Z are the same as those specified in **7.2.1**:

$$A = 3.0 \times 10^4 \frac{H}{N_0 Z}$$

K_3 : Value given by the following formula:

$$K_3 = \left\{\left(\frac{D}{P}\right)^2 \times (0.622 - 0.9x_0)^2 + (0.318 - 0.499x_0)^2\right\}^{\frac{1}{2}}$$

x_0 : Ratio of the radius from centreline of the propeller shaft to the boundary between the "blade flange and pitch control gear" and the propeller radius (*See* **Fig. D7.2**). Where x_0>0.3, the ratio is to be taken as 0.3.

L : Mean value of L_1 and L_2 (cm)

where L_1 and L_2 are the lengths of lines constructed from the centre of the bolts located on the edge of each side that are perpendicular to the line passing through the rotating centre of the flange at a pitch angle of β. (*See* **Fig. D7.3**)

F_c : Centrifugal force (N) of propeller blade given by the following formula:

$$F_c = 1.10 \times mR'N_0^2$$

m : Mass of one blade (kg)

R' : Distance between the centre of gravity of the blade and the centre line of the propeller shaft (cm)

n : Number of bolts on the face side of blade

σ_a : Allowable stress of bolt material given by the following formula (N/mm^2):

$$\sigma_a = 34.7 \times \left(\frac{\sigma_B + 160}{600}\right)$$

σ_B : Specified Tensile strength of bolt material (N/mm^2)

where σ_B >800 N/mm^2, it is to be taken as 800 N/mm^2.

Other symbols are the same as those given in the formula of **7.2.1-1**.

3 For blade fixing bolts, corrosion-resistant materials are to be used, or special means precluding their direct contact with sea water are to be provided.

4 The thickness of the flange for fitting the blade to the pitch control gear (the thickness as measured from the seat of fixing bolt or nut to the boundary face between the flange and the pitch control gear) is to be not less than the value calculated by the following formula:

$$t_f = 0.9d$$

where

t_f : Thickness of flange (mm) (*See* **Fig. D7.2**)

d : Required diameter of bolt calculated by the formula specified in **-2** (mm)

5 Blade fixing bolts are to be fitted tightly into the pitch control gear and provided with effective means for locking.

6 In cases where recesses for bolts are provided on the fillet at the root of the blades, the design blade section determined by the requirements for blade thickness in **7.2.1** is not to be reduced for the recess.

7 The face of the flange of the blade is to be fitted tightly to the face of pitch control gear and the circumferential clearance of the edge of flange is to be minimized.

8 In cases where pitch control gears are operated by hydraulic oil pump, a stand-by oil pump that is connected and ready for use at anytime or some other suitable device is to be provided in order to ensure that the ship can maintain its normal service condition in the event of a failure of the oil pump.

9 The operating oil piping arrangement is to comply with the requirements in **13.10**.

Fig. D7.2 Measuring Method of Blade Fixing Bolt Dimensions

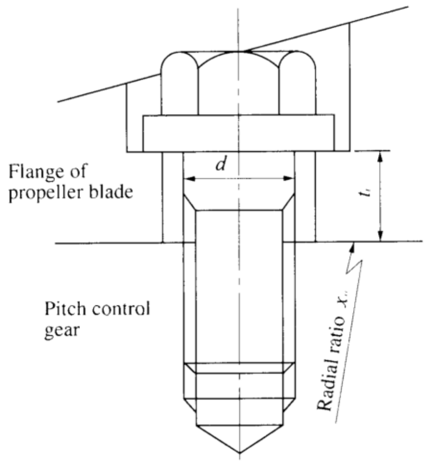
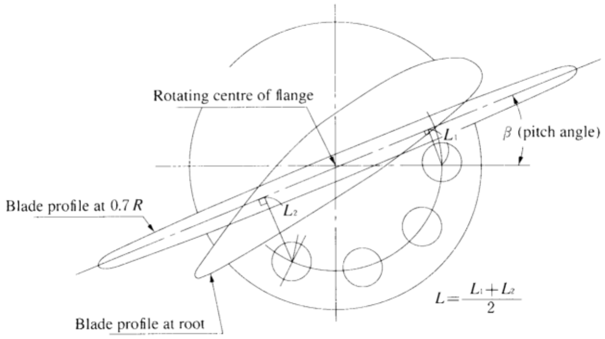


Fig. D7.3 Determination of L



7.2.3 Blade Fitting of Built-up Propeller

The blade fixing bolts and the flanges for fitting the blade of built-up propellers are to so designed as to comply with the requirements concerning to those for controllable pitch propellers specified in **7.2.2**.

7.3 Force Fitting of Propellers

7.3.1 Pull-up Length*

1 In cases where a propeller is force fitted onto a propeller shaft without the use of a key, the pull-up length is to be in accordance with the following **(1)** to **(3)**:

(1) Pull-up length by force fitting is to be within the range of the lower and upper limits as given by the following formulae. However, a taper is not to be more than 1/15, and special consideration is required in cases where propellers are force fitted onto propeller shafts through sleeves.

$$L_1 = PK_E + K_C(C_b - C_0)$$

$$L_2 = K_E K_W \frac{({K_{R1}}^2 - 1)}{\sqrt{(3{K_{R1}}^4 + 1)}} + K_c(C_b - C_0)$$

$$L_3 = 19.6K_E({K_{R1}}^2 - 1) + K_c(C_b - C_0)$$

L_1 : Lower limit of pull-up length against slipping at the reference temperature 35 °C (mm)

L_2 : Upper limit of pull-up length against detrimental deformations at the reference temperature 0 °C (mm) (in cases other than the case of L_3 shown below)

L_3 : Upper limit of pull-up length against detrimental deformations at the reference temperature 0 °C (mm) (in cases where the material of boss is manganese bronze casting and K_{R1}<1.89)

K_W : Value given by the following formula. For cast iron, the value is not to exceed 30 % of the nominal tensile strength.

$$K_W = 0.7\sigma_{0.2}$$

$\sigma_{0.2}$: Value of 0.2 % proof stress of propeller boss material as specified in **Table D7.3** (N/mm^2)

K_{R1} : Rate of R_1 to R_0 (R_1/R_0)

K_{R2} : Rate of R_2 to R_0 (R_2/R_0)

R_0 : Radius of the propeller shaft at the midpoint of taper in the axial direction (mm)

R_1 : Radius of propeller boss at the determinant point of the propeller boss ratio (mm)

R_2 : Inner radius at the section corresponding to R_0 in the case of a hollow propeller shaft (mm). For solid propeller shafts, the value is to be 0.

C_b : Temperature of propeller boss at time of fitting propeller (°C)

C_0 : Reference temperature: 35°C for L_1 (at which the space between boss and shaft tends to be loose), 0°C for L_2 and L_3 (at which the space between boss and shaft tends to shrink)

P : Value of minimum required surface pressure given by the following formula (N/mm_2):

$$P = \frac{qT}{SB}\left\{-q\tan\alpha + \sqrt{\mu^2 + B\left(\frac{F_V}{T}\right)^2}\right\}$$

q : Safety factor not to be less than 2.8 against friction slip at the reference temperature 35 °C

S : Contact area between propeller shaft and propeller boss on the drawing (mm^2)

α : Half angle of the taper at the propeller shaft cone part (*rad*)

B : Value given by the following formula:

$$B = \mu^2 - q^2 tan^2\alpha$$

μ : Coefficient of friction, equal to 0.13

T : Thrust force given by the following formula (*N*);

$$T = 1.76 \times 10^3 (H/V_s)$$

V_s : Ship speed at maximum continuous output (*kt*)

F_V : Tangential force acting on contact surface given by the following formula (*N*);

$$F_V = \frac{9.55cH}{N_0 R_0} \times 10^4$$

c : Value given by one of following

i) For steam turbine or gas turbines used as main propulsion machinery, geared reciprocating internal combustion engine drives and electric drives, and for direct reciprocating internal combustion engine drives with hydraulic, electromagnetic or high elasticity couplings

1.0

ii) For direct reciprocating internal combustion engine drives (except in the case of **i)** above)

1.2 or the value given by the following formula, whichever is greater. However, where a detailed report on the maximum torque acting on the fitted portion of the propeller under all operating conditions including transient conditions has been submitted to the satisfaction of the Society, it may comply with the provisions specified otherwise.

$$c = (0.194\ln D + 0.255)\left\{\left(\frac{N_C}{N_0}\right)^2 + 1.047\frac{Q_v N_0}{H} \times 10^{-2}\right\}$$

Q_V: Torsional vibratory torque acting on the fitted portion of the propeller at a rotational speed of resonant critical within the range of above 25 % of the number of maximum continuous revolutions (*N-m*)

H, N_0, D: Same as those specified in **7.2.1-1**, However, D is to be taken as 2.6 *m* for D <2.6 *m* and as 10.2 *m* for D >10.2 *m*.

N_c : Number of revolutions (*rpm*) at resonant critical divided by 100

K_E : Value given by the following formula (mm^3/N):

$$K_E = \frac{R_0}{tan\alpha}\left\{\frac{1}{E_b}\left(\frac{K_{R1}^2 + 1}{K_{R1}^2 - 1} + \nu_b\right) + \frac{1}{E_s}\left(\frac{1 + K_{R2}^2}{1 - K_{R2}^2} - \nu_s\right)\right\}$$

ν_b : Poisson's ratio for propeller boss material as specified in **Table D7.4**

ν_s : Poisson's ratio for propeller shaft material as specified in **Table D7.5**

E_b : Modulus of elasticity of propeller boss material as specified in **Table D7.4** (N/mm^2)

E_s : Modulus of elasticity of propeller shaft material as specified in **Table D7.5** (N/mm^2)

K_C : Value given by the following formula (*mm* /°C):

$$K_C = \left\{(\lambda_b - \lambda_S) + \frac{(C_b - C_S)}{(C_b - C_0)}\lambda_S\right\}\left\{\ell_0 - \frac{R_0}{tan\alpha}\right\}$$

C_S : Temperature of propeller shaft at time of fitting propeller (°C).

λ_b : Coefficient of linear thermal expansion of propeller boss material as specified in **Table D7.4** (*mm/mm*°C)

λ_S : Coefficient of linear thermal expansion of propeller shaft material as specified in **Table D7.5** (*mm/mm*°C)

ℓ_0 : Half length of the tapered part in the propeller boss hole in the axial direction (*mm*)

(2) Prior to final pull-up according to **(1)** above, the contact area between the mating surfaces is to be checked. Non-contact bands extending circumferentially around the boss or over the full length of the boss are not acceptable.

(3) After final pull-up according to **(1)** above, the propeller is to be secured by a nut on the propeller shaft. The nut is to be secured to the shaft.

Table D7.3 0.2 % proof stress of propeller boss material

Propeller boss material	0.2 % proof stress (N/mm^2)
*KHBsC*1	175
*KHBsC*2	
*KAlBC*3	245
*KAlBC*4	275

Note:
For materials different from those specified in the above table, the value is to be as deemed appropriate by the Society.

Table D7.4 Poisson's ratio, modulus of elasticity and coefficient of linear thermal expansion of propeller boss material

Material	Poisson's ratio	Modulus of elasticity (N/mm^2)	Coefficient of linear thermal expansion (mm/mm°C)
*KHBsC*1	0.33	1.08×10^5	17.5×10^{-6}
*KHBsC*2			
*KAlBC*3		1.18×10^5	
*KAlBC*4			
Cast iron	0.26	0.98×10^5	12.0×10^{-6}
Cast steel	0.29	2.06×10^5	

Note:
For materials different from those specified in the above table, the value is to be as deemed appropriate by the Society.

Table D7.5 Poisson's ratio, modulus of elasticity and coefficient of linear thermal expansion of propeller shaft material

Material	Poisson's ratio	Modulus of elasticity (N/mm^2)	Coefficient of linear thermal expansion (mm/mm°C)
Forged steel	0.29	2.06×10^5	12.0×10^{-6}

Note:
For materials different from those specified in the above table, the value is to be as deemed appropriate by the Society.

2 In cases where propeller is force fitted on the propeller shaft with the use of a key, the strength of the fitted part is to be such that it is sufficient for the torque to be transmitted.

7.3.2 Propeller Boss*

1 In cases where a propeller is force fitted onto a propeller shaft, the edge at the fore end of the tapered hole of the propeller boss is to be appropriately rounded off.

2 Propeller boss is not to be heated locally to a high temperature at the time of forcing on or drawing out.

7.4 Tests

7.4.1 Shop Tests*

Propellers are to be subjected to static balancing tests.

7.4.2 Tests after Installation On Board*

When a propeller is force-fitted onto a propeller shaft, irrespective whether it is done with or without a key, a force-fitting test is to be carried out to measure and record the pull-up length. This test may be carried out as a Shop Test.

Chapter 8 TORSIONAL VIBRATION OF SHAFTINGS

8.1 General

8.1.1 Scope

1 The requirements of this Chapter apply to power transmission systems for propulsion and propulsion shafting systems (except propellers), shafting systems for transmitting power from main propulsion machinery to generators, crankshafts of reciprocating internal combustion engines used as main propulsion machinery and shafting systems of generating plants using reciprocating internal combustion engines.

2 The requirements of this Chapter apply mutatis mutandis to the shafting systems of auxiliaries (hereinafter referred to in this Chapter as all auxiliaries excluding auxiliary machinery for specific use etc.) driven by reciprocating internal combustion engines.

8.1.2 Data to be Submitted

1 Torsional vibration calculation sheets covering the following items are to be submitted for approval:

(1) Natural frequency calculation tables for one node and two nodes vibration, and also for more nodes vibrations if necessary

(2) Calculation results of the torsional vibration stress at each resonant critical within a speed range up to 120% of the number of maximum continuous revolutions; and, in cases of reciprocating internal combustion engines, those of the torsional vibration stress for the flank appearing in the speed range from 90 to 120% caused by a resonance of the first major order (i.e., the *n* th or n/2 th order where *n* denotes the number of cylinders of reciprocating internal combustion engines) having its critical speed above 120% of the number of maximum continuous revolutions.

(3) Arrangement of crank throws and firing order (in cases of shafting systems driven by reciprocating internal combustion engines)

(4) For propulsion shafting systems intended to be continuously operated under one cylinder misfiring (i.e., no injection but with compression) condition of reciprocating internal combustion engines, calculation results of the torsional vibration stress with any one cylinder misfiring giving rise to the highest torsional vibration stress.

2 Notwithstanding the requirements specified in **-1**, submission of torsional vibration calculation sheets may be omitted in the following cases provided that approval of the Society is obtained:

(1) In cases where the shafting system is of the same type as previously approved one.

(2) In cases where there is a slight alternation in the specifications of the vibration system, and the frequency and torsional vibration stress can be deduced with satisfactory accuracy on the basis of the previous results of calculations or measurements.

(3) In cases where the shafting system is for a generating set which has an engine power of less than 110 *kW*.

8.1.3 Measurements

1 For the shafting systems where the submission of torsional vibration calculation sheets is required, measurements to confirm the correctness of the estimated value are to be carried out. However, where the submission of calculation sheets is omitted according to the requirement in **8.1.2-2**; and, the Society considers that there is no critical vibration within the service speed range, the measurement of torsional vibration may be omitted.

2 In cases where the barred speed ranges specified in **8.3.1** are marked for reciprocating internal combustion engines used as main propulsion machinery, the following **(1)** and **(2)** are to be confirmed and recorded.

(1) Passing time as well as the ship draft and speed of passing through the barred speed range (accelerating and decelerating). In the case of a controllable pitch propeller, the pitch is also to be confirmed and recorded.

(2) Running condition of engines at both the upper and lower borders of the barred speed range. In this case, the oscillation range of fuel index (fuel injection quantity (fuel rack position)) is normally to be less than 5% of the effective stroke (maximum fuel injection quantity (possible fuel rack range)). Alternatively, in the case of engines which do not have means to confirm fuel index, an oscillation range of speed less than 5% of maximum continuous revolution may be confirmed and recorded.

8.2 Allowable Limit

8.2.1 Crankshafts

The torsional vibration stresses on the crankshafts of reciprocating internal combustion engines of ships in which the reciprocating internal combustion engines are used as main propulsion machinery (excluding electric propulsion ships) are to be in accordance with the following requirements **(1)** through **(4)**:

(1) For continuous operation, when the number of revolutions is within the range of 80% to 100% of the number of maximum continuous revolutions, the torsional vibration stresses are not to exceed τ_1 given in following:

(a) For 4-stroke cycle in-line engines or 4-stroke cycle Vee type engines with firing intervals of 45 degrees or 60 degrees, the value of τ_1 is given by the following formula:

$$\tau_1 = 45 - 24\lambda^2$$

(b) For 2-stroke cycle engines or 4-stroke cycle Vee type engines other than shown in **(a)** above, the value of τ_1 is given by the following formula:

$$\tau_1 = 45 - 29\lambda^2$$

τ_1 : Allowable limit of torsional vibration stresses for the range of $0.8 < \lambda \leq 1.0$ (N/mm^2)

λ : Ratio of the number of revolutions to the number of maximum continuous revolutions

(2) When the number of revolutions is within the range of 80% and below the number of maximum continuous revolutions, the torsional vibration stresses are not to exceed τ_2 given below. Furthermore, in cases where the stresses exceed the value calculated by the formula of τ_1 in **(1)**, the barred speed ranges specified in **8.3** are to be imposed.

$$\tau_2 = 2\tau_1$$

τ_2 : Allowable limit of torsional vibration stresses for the range of $\lambda \leq 0.8$ (N/mm^2)

λ : Ratio of the number of revolutions to the number of maximum continuous revolutions

(3) When the number of revolutions is within the range of the number of maximum continuous revolutions to 115%, the torsional vibration stresses are not to exceed τ_3 given in the following:

(a) For 4-stroke cycle in-line engines or 4-stroke cycle Vee type engines with firing intervals of 45 degrees or 60 degrees, the value of τ_3 is given by the following formula:

$$\tau_3 = 21 + 237(\lambda - 0.8)\sqrt{\lambda - 1} \ (1 < \lambda \leq 1.15)$$

(b) For 2-stroke cycle engines or 4-stroke cycle Vee type engines other than shown in **(a)** above, the value of τ_3 is given by the following formula:

$$\tau_3 = 16 + 237(\lambda - 0.8)\sqrt{\lambda - 1} \ (1 < \lambda \leq 1.15)$$

τ_3 : Allowable limit of torsional vibration stresses for the range of $1.0 < \lambda \leq 1.15$ (N/mm^2)

λ : Ratio of the number of revolutions to the number of maximum continuous revolutions

(4) In cases where the tensile strength of the shaft material exceeds 440 N/mm^2, or its yield strength exceeds 225 N/mm^2, the values of τ_1, τ_2 and τ_3 given in **(1)**, **(2)** and **(3)** may be increased by multiplying the factor f_m given in the following formula:

(a) For τ_1 and τ_3

$$f_m = 1 + \frac{2}{3}\left(\frac{T_s}{440} - 1\right)$$

(b) For τ_2

$$f_m = \frac{Y}{225}$$

where

f_m : Correction factor for allowable limit of torsional vibration stress concerning the shaft material

T_s : Specified tensile strength of the shaft material (N/mm^2). However, the value of T_s for calculating f_m is not to exceed 760 N/mm^2 for carbon steel forgings, or 1080 N/mm^2 for low alloy steel forgings.

Y : Specified yield strength of the shaft material (N/mm^2)

8.2.2 Intermediate Shafts, Thrust Shafts, Propeller Shafts and Stern Tube Shafts*

1 For ships in which the reciprocating internal combustion engines are used as main propulsion machinery (excluding electric propulsion ships), the torsional vibration stresses acting on the intermediate shafts, thrust shaft, propeller shafts and stern tube shafts made of steel forgings (excluding stainless steel, etc.) are to be in accordance with the following requirements **(1)** and **(2)**. However, those shafts classified as either propeller shafts Kind 2 or stern tube shafts Kind 2 are to be deemed appropriate by the Society.

(1) For continuous operation, when the number of revolutions is within the range of 80% to 105% of the number of maximum continuous revolutions, the torsional vibration stresses are not to exceed τ_1 given in the following formulae:

$$\tau_1 = \frac{T_s + 160}{18} C_K C_D (3 - 2\lambda^2)(\lambda \leq 0.9)$$

$$\tau_1 = 1.38 \frac{T_s + 160}{18} C_K C_D (0.9 < \lambda)$$

where

τ_1 : Allowable limit of torsional vibration stresses for the range of $0.8 < \lambda \leq 1.05$ (N/mm^2)

λ : Ratio of the number of revolutions to the number of maximum continuous revolutions

T_s : Specified tensile strength of shaft material (N/mm^2)

However, the value of T_s for using in the formulae is not to exceed 800 N/mm^2 (600N/mm^2 for carbon steels in general) in intermediate shafts and thrust shafts, and 600 N/mm^2 in propeller shafts and stern tube shafts. The upper limit of the value of T_s used for the calculation may be increased to 950 N/mm^2 where the intermediate shafts are manufactured using steel forgings (excluding stainless steel forgings, etc.) which have specified tensile strengths greater than 800 N/mm^2 and are in accordance with the requirements of **Annex 6.2.2** "**Use of High-Strength Materials for Intermediate Shafts**". Where propeller shafts and stern tube shafts are made of the approved corrosion resistant materials or other materials having no effective means against corrosion by sea water, the value of T_s for use in the formulae is to be as deemed appropriate by the Society.

C_K : Coefficient concerning to the type and shape of the shaft, given in **Table D8.1**.

C_D : Coefficient concerning to the shaft size and determined by the following formula:

$$C_D = 0.35 + 0.93 d^{-0.2}$$

d = Diameter of the shaft (mm)

(2) When the number of revolutions is within the range of 80% and below the number of maximum continuous revolutions, the torsional vibration stress (including those in one cylinder misfiring conditions if intended to be continuously operated under such conditions) are not to exceed τ_2 given below. Furthermore, in cases where the stresses exceed the value calculated by the formula of τ_1 for the range of $\lambda \leq 0.9$ in **(1)**, the barred speed ranges specified in **8.3** are to be imposed.

$$\tau_2 = 1.7\tau_1/\sqrt{C_K}$$

where

τ_2 : Allowable limit of torsional vibration stresses for the range of $\lambda \leq 0.8$ (N/mm^2)

Other symbols used here are the same as in **(1)**

2 For ships in which the reciprocating internal combustion engines are used as main propulsion machinery (excluding electric propulsion ships), the torsional vibration stresses on the propeller shafts and stern tube shafts made of stainless steel forgings, etc. are to be in accordance with the following requirements **(1)** and **(2)**.

(1) For continuous operation, when the number of revolutions is within the range of 80 % to 105 % of the number of maximum continuous revolutions, the torsional vibration stresses are not to exceed τ_1 given in the following formulae:

$$\tau_1 = A - B\lambda^2 (\lambda \leq 0.9)$$

$$\tau_1 = C\ (0.9 < \lambda)$$

τ_1 : Allowable limit of torsional vibration stresses for the range of $0.8 < \lambda \leq 1.05$ (N/mm^2)

λ : Ratio of the number of revolutions to the number of maximum continuous revolutions

A, B, C: Values determined by the materials used, given in **Table D8.2**. For the materials other than specified in the Table, however, the values are to be deemed appropriate by the Society.

(2) When the number of revolutions is within the range of 80 % and below the number of maximum continuous revolutions, the torsional vibration stress (including those in one cylinder misfiring conditions if intended to be continuously operated under such conditions) are not to exceed τ_2 given below. Furthermore, in cases where the stresses exceed the value calculated by the formula of τ_1 for the range of $\lambda \leq 0.9$ in **(1)**, the barred speed ranges specified in **8.3** are to be imposed.

$$\tau_2 = 2.3\tau_1$$

τ_2 : Allowable limit of torsional vibration stresses for the range of $\lambda \leq 0.8$ (N/mm^2)

Other symbols used here are the same as in **(1)**.

3 The allowable limits of torsional vibration stresses on the shafts made of materials other than specified in **-1** and **-2**, and the allowable limits of torsional vibration stresses on the intermediate shafts, thrust shafts, propeller shafts and stern tube shafts for ships in which steam turbines or, gas turbines are used as main propulsion machinery, for electric propulsion ships, or for ships in which reciprocating internal combustion engines are used as main propulsion machinery which have electromagnetic slip couplings between main propulsion machinery and main propulsion systems are to be deemed appropriate by the Society.

Table D8.1 Values of C_K (4)

Intermediate shaft with						Thrust shaft		Propeller shaft and stern tube shaft	
integral flange coupling	flange couplings either shrink fit, push fit or cold fit	keyway, tapered connection	Keyway, cylindrical connection	transverse hole	longitudinal slot (1)	on both sides of thrust collar	in way of part subjected to axial load of roller bearing	near the big end of the tapered part of propeller shaft (2)	excluding the portion given in the left column (3)
1.0	1.0	0.6	0.45	0.50	0.30	0.85	0.85	0.55	0.80

Notes:

(1) For intermediate shafts with longitudinal slots, values of C_K may be determined using the following formulae:

$$C_K = 1.45/scf$$

$$scf = \alpha_{t(hole)} + 0.80 \frac{(l-e)/d_a}{\sqrt{\left(1-\frac{d_i}{d_a}\right)\frac{e}{d_a}}}$$

where

scf : Stress concentration factor at the end of slots defined as the ratio between the maximum local principal stress and $\sqrt{3}$ times the nominal torsional stress determined for the hollow shafts without slots (values obtained through Finite Element Calculation may be used as well)

l : Slot length

e : Slot width

d_i : Inside diameter of the hollow shaft at the slot

d_a : Outside diameter of the hollow shaft

$\alpha_{t(hole)}$: Stress concentration factor of radial holes (in this context, e = hole diameter) determined by the following formula (an approximate value of 2.3 may be used as well)

$$\alpha_{t(hole)} = 2.3 - 3\frac{e}{d_a} + 15\left(\frac{e}{d_a}\right)^2 + 10\left(\frac{e}{d_a}\right)^2\left(\frac{d_i}{d_a}\right)^2$$

(2) The portion between the big end of the tapered part of the propeller shaft (in cases where the propeller is fitted with a flange, the fore face of the flange) and the fore end of the aftermost stern tube bearing, or 2.5 d_s, whichever is greater. In this case d_s is the required diameter of the propeller shaft or stern tube shaft.

(3) The portion in the direction of the bow up to the fore end of the fwd stern tube seal.

(4) Any value of C_K other than those above is to be determined by the Society based on the submitted data in each case.

Table D8.2 Values of A, B and C

	A	B	C
KSUSF316 *KSUS316-SU*	40.7	30.6	15.9
KSUSF316L *KSUS316L-SU*	37.6	28.3	14.7

8.2.3 Shafting System of Generating Plants

1 The torsional vibration stresses on the crankshafts of reciprocating internal combustion engines used for generating plants (hereinafter referred to in this Chapter as including propulsion generating plants used for electric propulsion ships) are to be in accordance with the following requirements **(1)** and **(2)**:

(1) When the number of revolutions is within the range of 90 % to 110 % of the number of maximum continuous revolutions, the torsional vibration stresses are not to exceed τ_1 given in the following:

(a) For 4-stroke cycle in-line engines or 4-stroke cycle Vee type engines with firing intervals of 45 degrees or 60 degrees, the value of τ_1 is given by the following formula:

$$\tau_1 = 21N/mm^2$$

(b) For 2-stroke cycle engines and 4-stroke cycle Vee type engines other than shown in **(a)** above, the value of τ_1 is given by the following formula:

$$\tau_1 = 16N/mm^2$$

(2) When the number of revolutions is within the range of 90% and below the number of maximum continuous revolutions, the torsional vibration stresses are not to exceed τ_2 given below. Furthermore, in cases where the stresses exceed the value of τ_1 given in **(1)**, the barred speed ranges specified in **8.3** are to be imposed.

$$\tau_2 = 90N/mm^2$$

2 The torsional vibration stresses on the generator shafts of generating plants using reciprocating internal combustion engine are to be in accordance with the following requirements **(1)** and **(2)**:

(1) When the number of revolutions is within the range of 90 % to 110 % of the number of maximum continuous revolutions, the torsional vibration stresses are not to exceed τ_1 given in the following:

$$\tau_1 = 31N/mm^2$$

(2) When the number of revolutions is within the range of 90 % and below the number of maximum continuous revolutions, the torsional vibration stresses are not to exceed τ_2 given below. Furthermore, in cases where the stresses exceed the value of τ_1 given in **(1)**, the barred speed ranges specified in **8.3** are to be imposed.

$$\tau_2 = 118N/mm^2$$

3 In cases where the tensile strength of the shaft material exceeds 440 N/mm^2, or its yield strength exceed 225 N/mm^2, the values of τ_1 and τ_2 given in **-1** and **-2** may be increased by multiplying the factor f_m given in **8.2.1(4)**.

8.2.4 Power Transmission Systems*

1 The torsional vibration torques on the power transmission systems are to be in accordance with the following requirements **(1)** and **(2)**:

(1) Within the range of the allowable limits for τ_1 specified in **8.2.1**, **8.2.2** and **8.2.3**, the amplitudes of the torsional vibration torques are not to exceed the mean of the transmitting torque of the systems.

(2) Within any range other than that specified in **(1)**, the barred speed ranges are to be imposed in cases where the amplitudes of the torsional vibration torques exceed the mean transmitting torque.

2 The torsional vibration stresses on the gear shafts are to comply with the requirements for the intermediate shafts specified in **8.2.2**.

3 The allowable limits of the torsional vibration torques, stresses or amplitudes for the power transmission systems (including shaft couplings) other than gearings are to comply with the provisions specified elsewhere.

8.2.5 Avoidance of Major Criticals

The major criticals of one node vibration (e.g. the *n* th and *n* /2 th order for 4-stroke cycle and the *n* th order for 2-stroke cycle where *n* denotes the number of cylinders) in in-line engines are not to exist, except when approval of the Society is specifically obtained, within the following speed ranges:

For main propulsion shafting system $0.8 \le \lambda \le 1.1$

For shafting system of generating plants $0.9 \le \lambda \le 1.1$

where

λ : Ratio of the number of revolutions at the major critical to the number of maximum continuous revolutions

8.2.6 Detailed Evaluation for Strength*

Special consideration will be given to allowable limits of torsional vibration stresses that do not comply with the requirements in **8.2.1**, **8.2.2** and **8.2.3**, provided that detailed data and calculations are submitted to the Society and considered appropriate.

8.3 Barred Speed Range

8.3.1 Barred Speed Range for Avoiding Continuous Operation*

1 In cases where the torsional vibration stresses exceed the allowable limit τ_1 specified in **8.2**, barred speed ranges are to be marked with red zones on the engine tachometers and these ranges are to be passed through as quickly as possible. In this case, barred speed ranges are to be imposed in accordance with the following:

(1) The barred speed ranges are to be imposed between the following speed limits.

$$\frac{16N_c}{18-\lambda} \leq N_0 \leq \frac{(18-\lambda)N_c}{16}$$

where

N_0 : The number of revolutions to be barred (*rpm*)

N_c : The number of revolutions at the resonant critical (*rpm*)

λ : Ratio of the number of revolutions at the resonant critical to the number of maximum continuous revolutions

(2) For controllable pitch propellers, both full and zero pitch conditions are to be considered.

(3) The tachometer tolerance is to be considered.

(4) The engines are to be stable in operation at each end of barred speed ranges.

(5) Restricted speed ranges in one cylinder misfiring conditions are to enable safe navigation even where the ship is provided with only one propulsion engine.

2 In cases where the range in which the stresses exceed the allowable limit τ_1 specified in **8.2** is verified by measurements, such range may be taken as the barred speed range for avoiding continuous operation, notwithstanding the required range specified in **-1**.

3 For engines for which clearing the barred speed range for avoiding continuous operation specified in **8.3.1-1** and **-2** above is not readily available, transferring of the resonant points of torsional vibrations and other necessary measures are to be taken.

Chapter 9 BOILERS, ETC. AND INCINERATORS

9.1 General

9.1.1 Scope

1 The requirements in this Chapter apply to the following.

(1) Boilers (excluding the following **(a)** and **(b)**)

(a) Steam boilers with a design pressure not exceeding $0.1 MPa$ and heating surface not exceeding $1 m^2$

(b) Hot water boilers with a design pressure not exceeding $0.1 MPa$ and heating surface not exceeding $8 m^2$

(2) Thermal oil heaters

(3) Incinerators

2 The requirements in **9.11** may be applied to the boilers referred to in the preceding **-1(1)** with a design pressure not exceeding 0.35 MPa (hereinafter referred to as "small boilers").

9.1.2 Terminology

Terms used in this Part are defined as follows:

(1) "Boilers" are plants which generate steam or hot water by means of flame, combustion gases or other hot gases and include superheaters, reheaters, economizers and exhaust gas economizers, etc.

(2) "Main boiler" means boilers which supply steam to steam turbines used for the main propulsion of ships.

(3) "Essential auxiliary boilers" are boilers which supply steam necessary for the operation of auxiliary machinery essential for main propulsion, auxiliary machinery for manoeuvring and safety as well as for generators.

(4) "Exhaust gas boilers" are boilers which generates steam or hot water using only exhaust gases from reciprocating internal combustion engines, have independent steam spaces or hot wells and have outlets for steam or hot water.

(5) "Exhaust gas economizers" are those equipment which generates steam or hot water using only exhaust gases from reciprocating internal combustion engines and do not have independent steam spaces or hot wells.

(6) "Heating surfaces of boilers" are those areas calculated on combustion gas side surfaces where one side is exposed to combustion gas and the other side to water. Unless specified otherwise, the heating surfaces of superheaters, reheaters, economizers or exhaust gas economizers are excluded.

(7) "Approved working pressures of boilers" and "nominal pressure of boilers with built-in superheaters" are as defined in **2.1.21** and **2.1.22, Part A**.

(8) "Design pressures" are those pressure used in the calculations made to determine the scantlings of each component and are the maximum permissible working pressure of a component. Design pressures of boiler drums are not to be less than the approved working pressure of their respective boilers.

(9) "Fittings" are items directly attached (i.e. welded) to boilers (e.g. nozzles) as well as items not directly attached but connected to the boilers (e.g. valve boxes (including safety valves) and water level gauges) that receive pressure.

(10) "End plates" means the plates that cover both ends of the shell.

(11) "Tube plates" means end plates to which smoke tubes are attached in the case of smoke tube boilers, or end plates to which water tubes are attached in the case of water tube boilers.

9.1.3 Drawings and Data to be Submitted*

Drawings and data to be submitted are generally as follows:

(1) Drawings (with materials and scantlings)

(a) General arrangement of the boiler

(b) Details of shells and headers (including the internal fittings)

(c) Details of the seats for boiler fittings and nozzles

(d) Arrangement and details of the boiler tubes

(e) Arrangement and details of the tubes for the superheater and reheater

(f) Details of the internal desuperheater

(g) Arrangement and details of the tubes for the economizer or exhaust gas economizer
(h) Details of the air preheater
(i) Arrangement and details of the boiler fittings
(j) Arrangement of the safety valves (including principal particulars)
(k) Other drawings considered necessary by the Society

(2) Data
(a) Particulars of the boiler (design pressure, design temperature, maximum evaporation, heating surface, etc.)
(b) Welding specifications (including welding procedures, welding consumables and welding conditions)
(c) Operating instructions (for shell type exhaust gas economizers only)
(d) Other data considered necessary by the Society

9.2 Materials and Welding

9.2.1 Materials*

1 The materials used for the construction of the pressure parts of boilers are to comply with the requirements in **3.2**, **3.7**, **4.1**, **4.2**, **4.4**, **5.1**, **5.4**, or **6.1, Part K** according to respective conditions of service, and to be tested in accordance with the requirements in **Chapter 1** and **Chapter 2** of this Part. However, materials other than those given in any of the above may be used, provided that the material specifications are submitted and approved by the Society.

2 Notwithstanding the requirements given in **-1**, boiler fittings such as valves and nozzles, etc., whose dimensions and conditions of service have been approved by the Society, may use materials specified as appropriate in standards recognized by the Society.

9.2.2 Service Limitation of Materials Used for Fittings

For service limitation of materials to be used for the fittings, the requirements in **9.9.1** are to be complied with.

9.2.3 Heat Treatment of Steel Plates

In cases where a heat treatment, such as hot forming or stress relieving, is to be carried out on steel plates during the manufacturing process of boilers, the manufacturer of the boiler is to make this intention known when placing the order for the materials. In this case, the manufacturer of the steel plates is expected to follow what is specified in **3.2.4, Part K**.

9.2.4 Non-destructive Tests for Cast Steels*

The cast steel materials used for a boiler drum exposed to internal pressure are to be subjected to a radiographic test and a magnetic particle test in order to confirm that they are free from any detrimental defects.

9.2.5 Welding

Welding workmanship of the boiler is to comply with the requirements in **Chapter 11**.

9.3 Design Requirements

9.3.1 Symbols

Unless expressly specified otherwise, the symbols used in this Chapter are as follows:

f : Allowable stress (N/mm^2) conforming to the requirements in **9.4.1** or **12.2.1**.

T_r : Required thickness (mm) calculated by the design pressure. In addition, "allowable pressure" is the pressure obtained by replacing required thickness with actual thickness in any of the formula.

P : Design pressure (MPa)

J : Minimum value of the efficiency specified in **9.4.2**

R : Internal radius of drum (mm)

9.3.2 Design Pressure of Economizers and Exhaust Gas Economizers

1 The design pressure of an economizer is not to be less than the maximum working pressure of the economizer that is determined on the basis of the maximum working pressure of the feed pump.

2 The design pressure of an exhaust gas economizer is not to be less than the maximum working pressure of the exhaust gas economizer that is determined on the basis of the maximum working pressure of the boiler water circulating pump.

9.3.3 Considerations for Structural Strength

1 In cases where the effects from any additional stresses (e.g., such as local stress concentration, repeated loads and thermal stress, etc.) are significant, suitable measures, such as increasing the thickness, are to be taken if necessary.

2 The fixed parts of the flue tubes for vertical boilers are to be designed so that the deformation of the shape of the flue tube induced by the thermal expansion of the hemispherical furnace is not excessively restricted.

3 Sufficient consideration is to be given to the following **(1)** and **(2)** to prevent the overheating of the water tubes for any boilers having a combustion chamber with a high calorific capacity:

(1) Water is to be sufficiently circulated throughout the boiler by water tubes, and

(2) In order to prevent any scale from adhering to the sides of the boiler, proper means, such as a water softener, etc., are to be provided.

9.3.4 Boilers of Unusual Shape*

1 In cases where it is not practicable or reasonable to calculate the strength or to reinforce of the pressure receiving part of the boiler according to the requirements in **9.5** to **9.7** because the part is of an unusual shape, calculation results by another detailed method or analysis results as deemed appropriate by the Society are to be used after receiving the approval of the Society. Based on the results of such calculation or analysis, the part may be considered to be in compliance with the requirements in **9.5** to **9.7**.

2 In cases where it is not appropriate to design the pressure receiving part of the boiler according to the requirements in **9.5** to **9.7** because the part is of an unusual shape, strains or deformations are to be measured under a suitable load after receiving the approval of the Society. Based on the results of these measurements, the part may be considered to be in compliance with the requirements in **9.5** to **9.7**.

9.3.5 Considerations for Installing

1 Boilers are to be so installed as to minimize the effects of the following loads or external forces:

(1) Ship motions or any vibrations caused by machinery installations

(2) External forces caused by the piping or any other supports fitted onto the boiler

(3) Thermal expansions due to temperature fluctuation

2 Boilers are to be installed so that they are clear of any bulkheads as far as practicable.

3 Shell type exhaust gas economizers are to be installed so that the tube plate to shell connection can be inspected easily.

9.3.6 Protections against Flame

In cases where part of the boiler drum and the tube header construction will be exposed to flames or high temperature gas, proper thermal insulation or some other suitable means is to be provided. In addition, for shell type exhaust gas economizers, the insulation at the circumference of the tube end plate is to be detachable so that an ultrasonic examination of the tube plate to shell connection can be carried out.

9.3.7 Consideration for Soot Fire*

Consideration is to be given to prevent exhaust gas boilers and exhaust gas economizers, from being damaged by a soot fire.

9.4 Allowable Stress and Efficiency

9.4.1 Allowable Stress*

1 The allowable stress for each of the materials used for boilers is to be determined in accordance with the following. In this case, the material temperature used to evaluate the allowable stress of boilers is determined by increasing the designed maximum temperature of the internal fluid by the temperature increment at heating surface that is given in **Table D9.1**. However, the minimum temperature of the materials is not to be less than 250°C.

(1) Excluding all cast steels, the allowable stress (f) of carbon steel (including carbon manganese steel, hereinafter referred to as the same in this Chapter) and low alloy steels is not to be greater than value obtained from the following formulae, whichever is the smallest. However, the values given in **Table D9.2** for the allowable stress for each material temperature may also be used instead those from the formulae.

$$f_1 = \frac{R_{20}}{2.7}, f_2 = \frac{E_t}{1.6}, f_3 = \frac{S_R}{1.6}, f_4 = \frac{S_C}{1.0}$$

where

R_{20} : Specified tensile strength of the applicable steel at room temperature (N/mm^2).

E_t : Yield point of the applicable steel at material temperature (or 0.2% proof stress) (N/mm^2).

S_R : Average stress of the applicable steel to produce a rupture in 100,000 *hours* at material temperature. However, when the average stress of the width of dispersion exceeds the average value by ±20%, the value is 1.25 times the minimum stress that is required to produce rupture in 100,000 *hours* at material temperature (N/mm^2).

S_C : Average stress to produce a 1% elongation (creep) in the applicable steel in 100,000 *hours* at material temperature (N/mm^2).

(2) The allowable stress of electric resistance welded steel pipes is to be 85% of the values given in **Table D9.2**.

(3) The allowable stress of cast steels is to be 80% of the value obtained by the formula in **(1)** or the value given in **Table D9.2**. Cast steel exceeding 50*mm* in thickness is not to be used unless specially approved by the Society.

(4) The stress values of materials other than those specified in **(1)** and **(3)** will be considered in each case by the Society taking account of the mechanical properties of the materials.

Table D9.1 Temperature Increment to Internal Fluid Temperature for Material Temperature at Heating Surface

Heating surface, in general	Heated by contact	25°C
	Heated by radiation	50°C
Heating surface of superheater	Heated by contact	35°C
	Heated by radiation	50°C
Heating surface of economizer and exhaust gas economizer		25°C

Table D9.2 Value of Allowable Stress

Kind of material (grade)	Allowable stress (f) N/mm^2											
	250°C or below	300 °C	350 °C	375 °C	400 °C	425 °C	450 °C	475 °C	500 °C	525 °C	550 °C	575 °C
Rolled steel plate for boilers												
*KP*42	110	104	103	96	88	76	57	39	-	-	-	-
*KP*46	122	117	113	106	95	80	58	39	-	-	-	-
*KP*49	124	122	121	114	102	84	58	39	-	-	-	-
*KPA*46	122	117	113	113	113	108	101	90	69	48	-	-
*KPA*49	124	122	121	121	121	117	106	91	69	48	-	-
Steel headers												
*KBH*1	105	104	103	97	88	76	57	39	-	-	-	-
*KBH*2	117	115	113	106	95	80	58	39	-	-	-	-
*KBH*3	102	99	96	96	96	93	91	87	67	-	-	-
*KBH*4	106	104	103	103	103	102	98	92	74	-	-	-
*KBH*5	106	104	103	103	103	102	98	92	81	64	-	-
*KBH*6	106	104	103	103	103	102	98	92	81	64	-	-
Steel tubes for boilers												
*KSTB*33	86	84	81	78	74	66	-	-	-	-	-	-
*KSTB*35	88	87	86	82	76	76	53	-				
*KSTB*42	113	104	103	97	88	94	57	-	-	-	-	-
*KSTB*12	102	99	96	96	96	102	91	87	69	-	-	-
*KSTB*22	106	104	103	103	103	102	98	92	81	64	44	-
*KSTB*23	106	104	103	103	103	102	98	92	81	64	47	34
*KSTB*24	106	104	103	103	103	102	98	92	81	64	48	36
Forged steel (*see* **Part K**)	1/4 of the specified tensile strength of the material (where used at 350℃ or below)											
Cast steel (*see* **Part K**)	1/5 of the specified tensile strength of the material (where used at 350℃ or below)											

Note:

In cases where the material temperature is between those given in the Table, the value of allowable stress is to be determined by interpolation.

9.4.2 Efficiencies of Joints and Ligaments

1 The efficiency of joints is to be as follows:

(1) Seamless shells: 1.00

(2) Welded shells

(a) Double-welded butt joints: 1.00

(b) Other cases: 0.90

2 The efficiency of ligaments is to be as follows:

(1) The efficiency of a longitudinal ligament (hereinafter referred to as "longitudinal efficiency") along the row of tube holes of a shell plate having a row parallel or nearly parallel to the shell axis, or of a shell or tube plate having several parallel rows that are sufficiently apart from each other, is to be determined by the following formulae:

(a) In cases where the pitch of the tube holes is uniform:

$$J_1 = \frac{p-d}{p}$$

where

J_1 : Efficiency of the ligament

p : Pitch of the tube holes (mm)

d : Diameter of the tube holes (mm)

(b) In cases where the pitch of the tube holes is irregular:

$$J_2 = \frac{L - nd}{L}$$

where

J_2 : Efficiency of the ligament

d : As specified in **(a)**

L : Total length between the centres corresponding to n consecutive ligaments (mm)

n : Number of tube holes in the length L

(2) The efficiency of a circumferential ligament (hereinafter referred to as "circumferential efficiency") at the part of the tube holes drilled in the circumferential direction of the shell is to be calculated in a similar manner to that specified in **(1)**, and is not to be less than 50% of the longitudinal efficiency. In this case, the pitch of the tube holes in the circumferential direction is to be measured either on the flat plate before rolling or along the median line of plate thickness after rolling.

(3) The efficiency of a ligament at the part of the tube holes drilled in a diagonal direction to the shell is to be determined by the following formula:

(a) In cases where the tube holes are drilled in a diagonal direction to the shell as shown in **Fig. D9.1** and **Fig. D9.2**: The efficiency obtained from the following formula or the longitudinal efficiency, whichever is smaller, is to be taken as the efficiency of the ligament at the part of the tube holes.

$$J_3 = \frac{2}{A + B + \sqrt{(A - B)^2 + 4C^2}}$$

where

J_3 : Efficiency of the ligament

$$A = \frac{\cos^2\alpha + 1}{2(1 - \frac{d\cos\alpha}{a})}$$

$$B = \frac{1}{2}(1 - \frac{d\cos\alpha}{a})(\sin^2\alpha + 1)$$

$$C = \frac{\sin\alpha \cos\alpha}{2(1 - \frac{d\cos\alpha}{a})}$$

$$\cos\alpha = \frac{1}{\sqrt{1 + \frac{b^2}{a^2}}}$$

$$\sin\alpha = \frac{1}{\sqrt{1 + \frac{a^2}{b^2}}}$$

α : As shown in **Fig. D9.1**, **Fig. D9.2** and **Fig. D9.3**

a, b : As shown in **Fig. D9.1**, **Fig. D9.2** and **Fig. D9.3** (mm)

d : Diameter of the tube holes (mm)

(b) In **(a)**, where the tube holes are arranged in a regular staggered spacing as shown in **Fig. D9.3**:
The efficiency obtained from the formula in **(a)**, twice the circumferential efficiency or the longitudinal efficiency, whichever is the smallest, is to be taken as the efficiency of the ligament at the part of the tube holes.

Note: The efficiencies of the ligament obtained from **(a)** and **(b)** shown in **Fig. D9.4** and **Fig. D9.5** are from taking the ratio $\frac{b}{a}$ on the abscissa and ratio $\frac{2a-d}{2a}$ as the parameter.

(4) The efficiency of a ligament per unit length, where the tube holes are irregularly arranged along the longitudinal direction of the shell, is to be the smaller of the two smallest values calculated by the following **(a)** or **(b)**. However, the efficiency need not be smaller than the minimum efficiency calculated by taking L_1 as the distance between the centres of tubes on both ends of the tube rows within a length equal to the inside diameter of the shell (the distance to the centre of the adjacent tube hole, in cases where there is only one tube hole within a length equal to the inside diameter of the shell).

(a) For a length L_1 equal to the inside diameter of the shell (1,520 mm maximum)

$$J_4 = \frac{a + b + c + \cdots\cdots}{L_1}$$

(b) For a length L_2 equal to the inside radius of the shell (760 mm maximum)

$$J_5 = \frac{a + b + c + \cdots\cdots}{L_2} \times 1.25$$

where

J_4 and J_5 : Efficiency of the ligament

a, b, c : Distances between the tube holes arranged along the longitudinal direction of shell. If they are arranged in a diagonal direction, the distances are to be the length projected on the longitudinal direction multiplied by the efficiency obtained from **(3)**.

9.5 Calculations of Required Dimensions of Each Member

9.5.1 Restrictions to Thickness of Each Member

1 The thickness of shell plates and end plates are not to be less than 6mm. The thickness of a formed end plate, except for a full hemispherical end plate, is not to be less than the thickness (calculated by using an efficiency value equal to 1.00) of the shell to which the end plate is attached.

2 The thickness of tube plates and flat plates is not to be less than 10mm for the tube plates and 6mm for the flat plates.

3 The thickness of nozzles welded to drum shells and connected with mountings, etc. is not to be less than either the value 2.5mm added to 1/25 of the outside diameter of the nozzle or the value calculated by the formula given in **9.7.4.** However, this value need not be more than the thickness of the drum onto which the nozzle is welded.

4 The thickness of furnace plates is not to be less than 5mm or more than 22mm.

9.5.2 Required Thickness of Cylindrical Shell Plates subjected to Internal Pressure

The required thickness of cylindrical shell plates subjected to internal pressure is to be calculated by the following formula. However, in the case of cylindrical shell plates having openings for which reinforcement is required, the openings are to be reinforced in accordance with the requirements in **9.6.3**.

$$T_r = \frac{PR}{fJ - 0.5P} + 1$$

Fig. D9.1 Spacing of Holes on a Diagonal Line

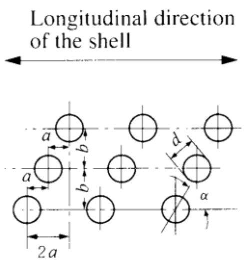


Fig. D9.2 Saw Tooth Pattern of Holes

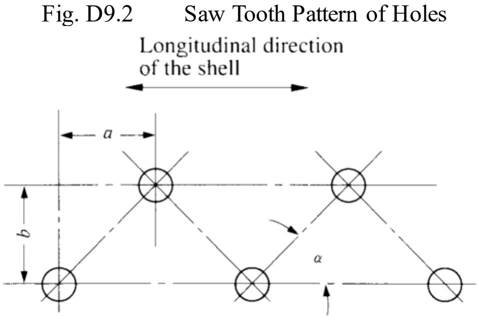


Fig. D9.3 Regular Staggered Pattern of Holes

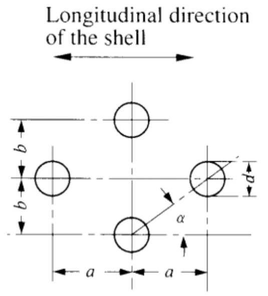


Fig. D9.4 The Efficiency of Ligament at the Part of the Tube Holes drilled in a Circumferential Direction

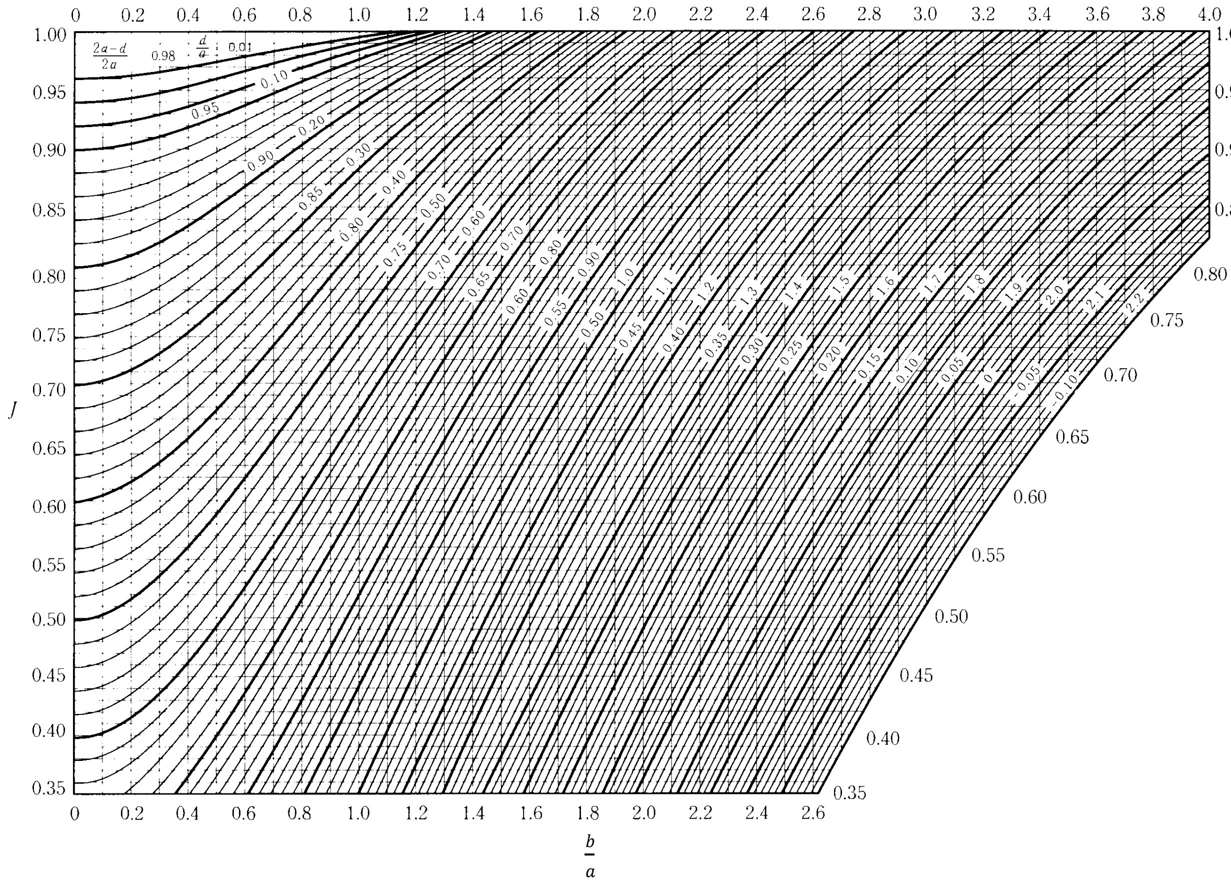
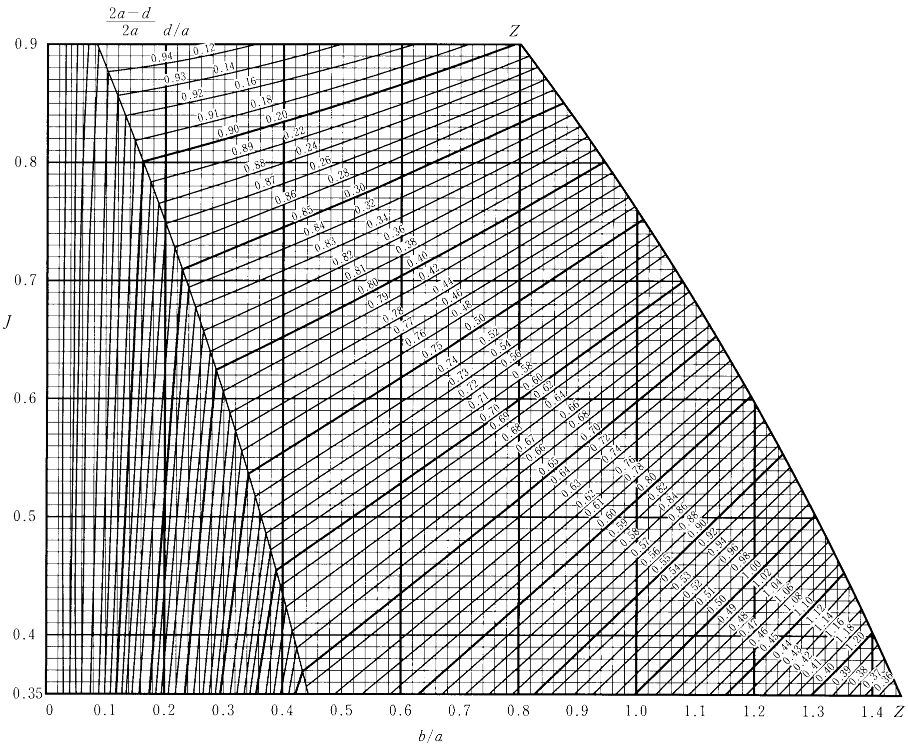


Fig. D9.5 The Efficiency of Ligament at the Part of the Tube Holes drilled in a Diagonal Direction



Note:

In cases where a point falls in the field on the right side of the intersection Z-Z, the longitudinal efficiency is to be deemed as the efficiency at the part of the tube holes

9.5.3 Required Thickness of Formed End Plates Subjected to Pressure on Concave Side without Stays or Other Supports

1 The required thickness of end plate having no opening is to be calculated by the following formula:

(1) Dished and hemispherical end plates

$$T_r = \frac{PR_1W}{2fJ - 0.5P} + 1$$

where

$W = \frac{1}{4}\left(3 + \sqrt{\frac{R_1}{r}}\right)$ for a dished end plate

W= 1 for a hemispherical end plate

R_1 : Inside crown radius

To be less than the outside diameter of the end plate

r : Inside knuckle radius

Not to be less than 6% of the outside diameter of the skirt of the end plate or 3 times the actual thickness of the end plate, whichever is greater

(2) Semi-ellipsoidal end plates (in cases where half of the inside minor axis of the end plate is not less than 1/4 of the inside major axis of the end plate)

$$T_r = \frac{PR}{fJ - 0.25P} + 1$$

2 The required thickness of end plates having openings is to comply with the following requirements in **(1)**, **(2)** or **(3)**:

(1) In cases where no reinforcement for openings is necessary according to the requirements in **9.6.2**, or the openings are reinforced in accordance with the requirements in **9.6.3-3** to **-5**, the required thickness is to be calculated by the formula specified in **-1** above.

(2) In cases where an end plate has a flanged-in manhole or an access opening with a maximum diameter exceeding 150mm and the flanged-in reinforcement complies with the requirement in **9.6.3-7**, the thickness is to be calculated as follows:

(a) Dished or hemispherical end plates

The thickness is to be increased by not less than 15% (if the calculated value is less than 3 mm, the value is to be taken as 3mm) of the required thickness calculated by the formula specified in **-1(1)**. In this case, where the inside crown radius of the end plate is smaller than 0.80 times the inside diameter of the shell, the value of the inside crown radius in the formula is to be 0.80 times the inside diameter of the shell. In calculating the thickness of an end plate having two manholes in accordance with **(a)**, the distance between the two manholes is not to be less than 1/4 of the outside diameter of the end plate.

(b) Semi-ellipsoidal end plates

The requirements specified in **-1(1)** are to be applied. However, in this case R_1 is to be 0.80 times the inside diameter of shell and W is to be 1.77.

(3) The required thickness, where the openings are not reinforced in accordance with the requirements in **(1)** or **(2)**, is to be calculated by the following formula. However, the thickness is not to be less than the value obtained by the formula given in **-1**.

$$T_r = \frac{PD_0}{2f}K + 1$$

where

D_0 : Outside diameter of the end plate (mm)

K : As shown in **Fig. D9.6**, however, this is only applicable to the end plates complying with the following conditions:

Hemispherical end plates:

$0.003D_0 \leq T_e \leq 0.16D_0$

Semi-ellipsoidal end plates:

$0.003D_0 \leq T_e \leq 0.08D_0$

$H \geq 0.18D_0$

Dished end plates:

$0.003D_0 \leq T_e \leq 0.08D_0$

$r \geq 0.1D_0$

$r \geq 3T_e$

$R_1 \leq D_0$

$H \geq 0.18D_0$

or $0.01D_0 \leq T_e \leq 0.03D_0$

$r \geq 0.06D_0$

$H = 0.18D_0$

or $0.02D_0 \leq T_e \leq 0.03D_0$

$r \geq 0.06D_0$

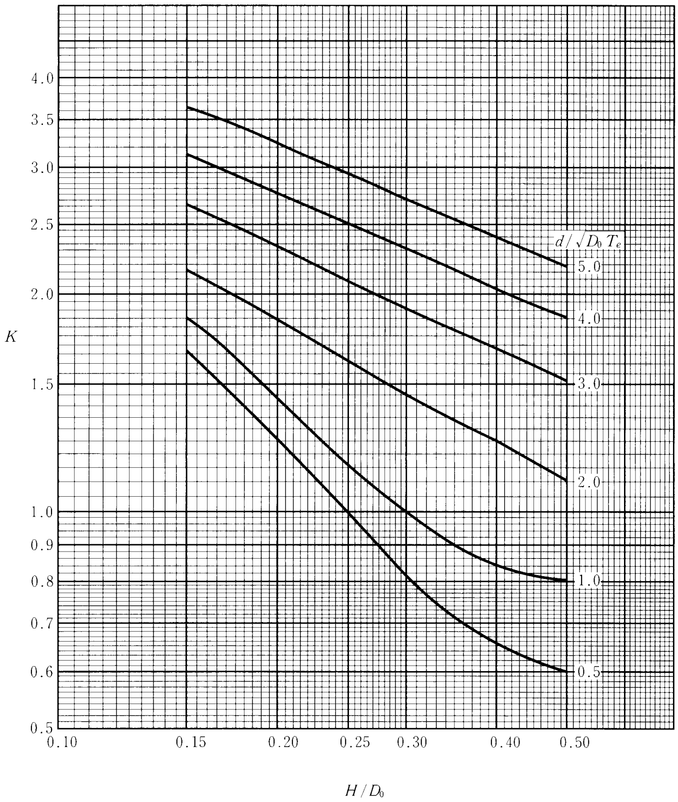
$0.18D_0 \leq H \leq 0.22D_0$

T_e : Actual thickness of the end plate (mm)

H : Depth of the end plate measured on its external surface from the plane of junction of the dished part with the cylindrical part (mm)

R_1 and r : As specified in **-1(1)**

Fig. D9.6 Value of K



Notes:

d : Diameter of the opening (mm)

H : Depth of the end plate measured on its external surface from the plane of junction of the dished part with the cylindrical part (mm)

D_0 : Outside diameter of the end plate (mm)

9.5.4 Required Thickness of Formed End Plates subjected to Pressure on Convex Side

The required thickness of formed end plates subjected to pressure on their convex sides is not to be less than the thickness calculated on the assumption that their concave sides are subjected to a pressure at least 1.67 times the design pressure.

9.5.5 Required Thickness of Flat End Plates and Cover Plates, etc., without Stays or Other Supports

1 In cases where the flat end plates and cover plates without stays or other supports are welded to the shell plates, the required thickness is to be calculated by the following formulae:

(1) Circular plates

$$T_r = C_1 d\sqrt{\frac{P}{f}} + 1$$

(2) Non-circular plates

$$T_r = C_1 C_2 d\sqrt{\frac{P}{f}} + 1$$

where

C_1 : Constant shown in **Fig. D9.11**

$C_2 = \sqrt{3.4 - 2.4\frac{d}{D'}}$, but need not be over 1.6.

d : Diameter shown in **Fig. D9.11** (for circular plates), or the minimum length (for non-circular plates) (mm)

D' : Long span of non-circular end plates or covers measured perpendicular to the short span (mm)

2 In cases where the flat cover plates without stays are bolted to the shell plate, the required thickness is to be calculated by the following formulae:

(1) In cases where full face gaskets are used;

For circular plates

$$T_r = d\sqrt{\frac{C_3P}{f}} + 1$$

For non-circular plates

$$T_r = d\sqrt{\frac{C_3C_4P}{f}} + 1$$

(2) In cases where moment due to gasket reaction is to be taken into account;

For circular plates

$$T_r = d\sqrt{\frac{C_3P}{f} + \frac{1.78Wh_g}{fd^3}} + 1$$

For non-circular plates

$$T_r = d\sqrt{\frac{C_3C_4P}{f} + \frac{6Wh_g}{fLd^2}} + 1$$

where

C_3 : Constant determined by bolting methods as shown in Fig. D9.12

C_4 : $3.4 - 2.4\frac{d}{D'}$, but need not be over 2.5.

d : Diameter shown in Fig. D9.12 (for circular plates), or minimum length (for non-circular plates) (mm)

D' : Long span of non-circular end plates or covers measured perpendicular to the short span (mm)

W : Mean load (N) of bolt loads necessary for the watertightness and allowable load of the bolt actually used

L : Total length of the circle passing through bolt centers (mm)

h_g : Arm length of moment due to the gasket reaction shown in Fig. D9.12 (mm)

9.5.6 Required Thickness of Flat Plates with Stays or Other Supports*

1 The required thickness of flat plates, except tube nests supported by stays or stay tubes, is to be calculated by the following formula. In cases where gusset plates are used as supports instead of stays or stay tubes, they are to comply with standards deemed appropriate by the Society.

$$T_r = C_5S\sqrt{\frac{P}{f}} + 1$$

where

C_5 : Constant determined by the fixing methods of the stays or stay tubes as given in Table D9.3. In cases where various fixing methods are used, the value C_5 is to be the mean of the constants for the respective methods.

S : In cases where the stays or stay tubes are arranged regularly, "S" is to be calculated by the following formula:

$S = \sqrt{a^2 + b^2}\,(mm)$

a : Horizontal pitch of stays or stay tubes (mm)

b : Vertical pitch of stays or stay tubes (mm).

In cases where stays or stay tubes are arranged irregularly, "S" is the diameter (mm) of the maximum circle drawn to pass through at least three supported points, but not including any supported point in the circle. However, in cases where the maximum circle drawn passes through only two supported points and there are no supported points located within the circle, the diameter (mm) of the maximum circle may be used as "S".

2 The position and constant C_5 of the supported point at the welding part between the plain end and the curved flange or shell, furnace, etc. are as follows

(1) The commencement of the curvature of the flange is to be regarded as the point of support. Where, however, the inner radius of the curvature is greater than 2.5 times the thickness of the plate, the points located at a distance of 3.5 times the thickness of the plate from the outer surface of the flange may be considered as a commencement of the curvature. In this case, the value of constant C_5 is to be 0.39 where the plates are exposed to flames or 0.36 where the plates are not exposed to flames.

(2) The inside of the welding part between the plain ends and the shell, furnaces, etc. are to be regarded as points of support. In this case, the value of constant C_5 is to be 0.47 where the plates are exposed to flames or 0.43 where the plates are not exposed to

flames.

3 The required thickness of the tube nests of the boiler tube plate supported by stay tubes is to be calculated by the following formula:

$$T_r = C_6 p \sqrt{\frac{P}{f}} + 1$$

where

C_6 : Constant determined by the fixing method of the stay tubes as given in **Table D9.4**

p : In cases where the stay tubes are arranged regularly, the mean pitch of the stay tubes is obtained by dividing the sum of the four sides of a quadrilateral formed by four supports (mm). In cases where the stay tubes are arranged irregularly, "S" (mm) is the diameter of a maximum circle drawn to pass through at least three supported points, but not including any supported point in the circle and $S/\sqrt{2}$ is to be used in lieu of "p"

4 The required thickness of the tube plates of vertical boilers having horizontal smoke tubes which form smoke tube nests is to be calculated by the formula in **-3** or by the following formula, whichever is greater:

$$T_r = \frac{PDp}{1.97f(P - d_s)} + 1$$

where

D : Twice the distance from the centre of the outer row of tube holes of the tube plate to the axis of the shell (mm)

p : Vertical pitch of the tubes (mm)

d_s : Diameter of the tube holes in the tube plate (mm)

5 The required thickness of the back tube plates in a cylindrical boiler with a wet combustion chamber is to be calculated by the formula in **-3** or by the following formula, whichever is greater:

$$T_r = \frac{PWH}{183(H - d_i)}$$

where

H : Horizontal pitch of the smoke tube (mm)

d_i : Inside diameter of the ordinary smoke tube (mm)

W : Depth of the upper part of the combustion chamber (mm)

6 As for the scantlings of the stayed top plate and the stayed side plate of the combustion chamber of a cylindrical boiler, the distance between the row of stays nearest to the tube plate or the back plate and the commencement line of curvature of the tube plate or the back plate is not to be greater than "a" determined by the formula in **-1**, substituting the actual thickness for the required thickness.

Table D9.3 Value of Constant C5

Fixing method of stay or stay tube		In cases where the plates are not exposed to flames	In cases where the plates are exposed to flames
(1)	In cases where the stays are inserted into the plate as (5) *A* in **Fig. D9.11**	0.35	0.38
(2)	In cases where the stays are inserted into the plate as (5) *B* in **Fig. D9.11**	0.37	0.40
(3)	In cases where the stays are inserted into the plate as (5) *C* in **Fig. D9.11**	0.41	0.44
(4)	In cases where the stays are inserted into the plate as (5) *D* in **Fig. D9.11**	0.50	0.53
(5)	In cases where the stay tubes are inserted into the plate as (6) *A* in **Fig. D9.11**	0.42	0.45
(6)	In cases where the stay tubes are inserted into the plate as (6) *B* in **Fig. D9.11**	0.49	0.52
(7)	In cases where the stay tubes are inserted into the plate as (6) *C* in **Fig. D9.11**	0.49	0.52

Table D9.4 Values of Constant C_6

Fixing method of stay or stay tubes	In cases where the plates are not exposed to flames	In cases where the plates are exposed to flames
In cases where the stay tubes are inserted into the plate as (6) A in **Fig. D9.11**	0.51	0.54
In cases where the stay tubes are inserted into the plate as (6) B in **Fig. D9.11**	0.57	0.61
In cases where the stay tubes are inserted into the plate as (6) C in **Fig. D9.11**	0.57	0.61

9.5.7 Required Thickness of Corrugated Furnaces

The required thickness of a corrugated furnace is to be calculated by the following formula:

$$T_r = \frac{PD}{C} + 1$$

where

D : Minimum outside diameter at the corrugated part of the furnace (mm)

C : Constant given in **Table D9.5**

Table D9.5 Value of Constant C

Type of furnace	C
Morrison, Deighton and similar furnaces	107
Leeds forge bulb furnace	104

9.5.8 Required Thickness of Plain Cylindrical Furnaces*

The required thickness of a plain cylindrical furnace or a cylindrical bottom, which is not reinforced by stays or any other means, and the smoke uptake of a combustion chamber are to be calculated by the following formulae, whichever is greater:

$$T_r = \sqrt{\frac{PD(L+610)}{10500}} + 1$$

$$T_r = \frac{1}{325}\left(\frac{PD}{0.35} + L\right) + 1$$

where

D : External diameter of the furnace or the combustion chamber bottom (mm)

L : Length of the furnace or depth of the combustion chamber bottom (mm)

The length of a furnace is measured from the commencement of curvature in cases where the furnace plates are flanged and jointed to other plates, reinforcing rings, etc.

9.5.9 Required Thickness of Hemispherical Furnaces without Stays or Other Supports*

The required thickness of a hemispherical furnace without any stays or other supports is to be calculated by the following formula:

$$T_r = \frac{PR_f}{62} + 1$$

where

R_f : Outer radius of the curvature of the furnace (mm)

9.5.10 Required Thickness of Ogee Rings of Vertical Boilers

The required thickness of any ogee rings, which sustain the whole vertical load of the furnace, connecting the furnace bottom of a vertical boiler to the shell is to be calculated by the following formula:

$$T_r = \sqrt{\frac{PD(D-d)}{1010}} + 1$$

where

D : Inside diameter of the shell (mm)

d : External diameter of the lower part of the furnace where it joins the ogee ring (mm)

9.5.11 Required Thickness of Furnace Foundation Ring Plates of Vertical Boilers

The required thickness of a furnace foundation ring plate (refer to **Fig. D9.11(d)**(4)E) connecting the furnace bottom of a vertical boiler to the shell is to be calculated by the following formula:

$$T_r = 1.28\sqrt{DP}$$

where

D : Inside diameter of the shell (mm)

9.5.12 Required Diameter of Stays*

1 The required diameter of a stay is to be calculated by the following formula:

$$d = C\sqrt{PA} + 3$$

where

d : Required diameter of the stay (mm)

A : Net area supported by one stay (mm^2)

C : 0.13

2 In applying the formula in **-1** to diagonal stays, C in the formula is to be replaced by C_1 given by the following formula:

$$C_1 = 0.13\sqrt{\frac{L}{H}}$$

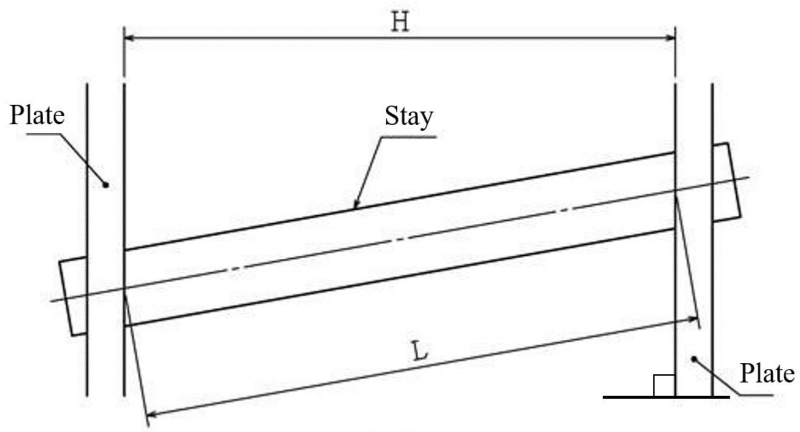
where

L : Length of the diagonal stay (mm) (refer to **Fig. D9.7**)

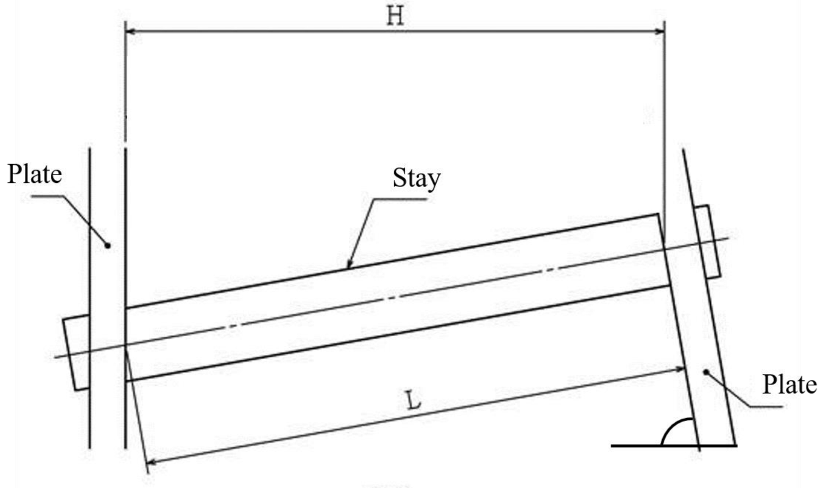
H : Equivalent length of the stays perpendicular to the support surface (mm) (refer to **Fig. D9.7**)

Fig. D9.7 Corresponding Parts of L and H

(In cases where the ends of the plates are at right angles to the installation part)



(In cases where the ends of the plates are not at right angles to the installation part)



9.5.13 Required Dimensions of Stay Tubes

The required dimensions of stay tubes supporting the tube plates are to be calculated by the following formula. However, the thickness of the stay tubes is not to be less than 6 *mm* for those in the bounding rows of tube nests or less than 4.5 *mm* for all others.

$$a = \frac{PA}{51.7}$$

where

a : Minimum net sectional area of one stay tube (mm^2)

A : Net area supported by one stay tube (mm^2)

9.5.14 Required Thickness of Girders Supporting Top Plates of Combustion Chambers and their Distance to Side Plates

1 The required thickness of steel girders supporting the top plates of combustion chambers is to be calculated by the following formula:

$$T_r = \frac{DLP\,(L-p)}{Cd^2S}$$

where

T_r : Required thickness of the girders (*mm*). However, in the case of double plate construction, the sum of the thickness of each plate

d : Depth of the girders at their centre (*mm*)

L : Width of the combustion chamber measured along inner upper part (*mm*)

p : Pitch of the stays supporting the girder (*mm*)

D : Pitch of the girders (*mm*)

S : Specified tensile strength of the material used for the girders (N/mm^2)

C : Constant given in **Table D9.6**

2 The distance between the inner surface of the side plate and the centre of the supporting beam nearest to it is not to be more than the pitch D in cases where the outer radius of the knuckle, used to connect the top plate of the combustion chamber of a boiler to a side plate, is less than 1/2 of the pitch D of the supporting beam obtained from the formula in **-1**, after substituting the actual thickness of girders of a boiler into the formula. And, where the outer radius of the knuckle is larger than $D/2$, the width of the flat surface measured from the centre of the supporting beam to the starting point of the knuckle is not to be more than $D/2$.

Table D9.6 Value of Constant C

When the number of stays (n) in each girder is odd	$\frac{0.253n}{n+1}$
When the number of stays (n) in each girder is even	$\frac{0.253(n+1)}{n+2}$

9.5.15 Required Thickness of Cylindrical Headers

The required thickness of any cylindrical headers is to be calculated by the formula in **9.5.2**. However, in cases where the thickness of the header exceeds 1/2 of the inside radius of the header and the material temperature is 375°C or below, the required thickness is to be calculated by the following formula:

$$T_r = R\left(\sqrt{\frac{fJ+P}{fJ-P}} - 1\right) + 1$$

9.5.16 Required Thickness of Square Headers

The required thickness of any square headers made of forged or welded steel plates is to be calculated by the following formula:

(1) In cases where the holes are not arranged in succession:

$$T_r = \frac{Pl_2}{4f}\left(1 + \sqrt{1 + 4f\frac{l_1^2}{Pl_2^2}}\right) + 1.5$$

(2) In cases where the holes are arranged in succession:

$$T_r = \frac{Pl_2}{4f}\left(1 + \sqrt{1 + \frac{8fl_1^2}{(1+J)Pl_2^2}}\right) + 1.5$$

where

l_1 : Inside breadth measured between the supports of any flat surfaces needed for strength calculation (mm)

l_2 : Inside breadth of another side adjacent to l_1 (mm)

9.6 Manholes, Other Openings for Nozzles, etc. and their Reinforcements

9.6.1 Manholes, Cleaning Holes and Inspection Holes*

1 Boilers are to be provided with manholes or cleaning holes of sufficient size at suitable positions, so that they permit easy access for the inspection and the maintenance. However, in cases where it is impractical to provide manholes or cleaning holes due to construction or dimension concerns, two or more inspection holes provided at positions suitable for internal inspection will be accepted as a substitute for them.

2 The construction of all manholes or cleaning holes is to comply with the following requirements in **(1)** to **(3)**:

(1) The minor axis of any oval manhole provided on the shell plate is to be parallel to the longitudinal direction of the drum.

(2) Internal type manhole covers are to be provided with a spigot which has a clearance of not more than 1.5 mm all-round.

(3) Covers are to have sufficient strength and be constructed so that the repetition of covering and uncovering does not to impair safety. In cases where covers are bolted shut, they are to be of such construction so that the breakage of a bolt will not cause any danger.

3 The inspection holes of headers are to be machine-finished so that all inspection hole covers can be effectively fitted.

4 In cases where flange openings on boiler drums are used as inspection holes, the pipes to be connected are to be ones that can be easily removed.

9.6.2 Reinforcement of Openings

In cases where manholes, other openings for nozzles, etc. are provided in the shell, the openings are to be reinforced. However, this reinforcement may be omitted for any of the following single openings:

(1) Openings having a maximum diameter (in threaded openings, the diameter of the root) of not more than 60mm or more than 1/4 of the inside diameter of the shell.

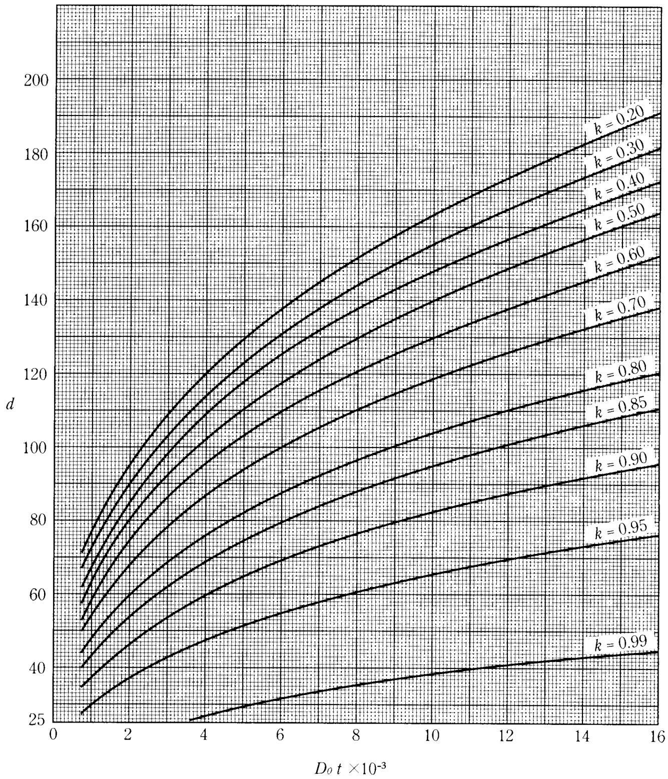
(2) Openings provided on the shell plate having a maximum diameter not exceeding the value given in **Fig. D9.8**. In this case, unreinforced openings are not to exceed 200mm in diameter.

(3) Openings provided on the end plate where no reinforcement is required due to the increased thickness of the end plates in compliance with the requirements in **9.5.3-2(3)**.

(4) Openings provided on the end plate or cover plate where the thickness of the end plate or cover plate is increased in accordance with the requirements in **9.6.3-3(2)**.

Fig. D9.8 Maximum Diameter of Openings provided on the Shell for which Reinforcement may be Omitted

(A)



Notes:

d : Maximum diameter of openings (mm) which are not required to be reinforced, in this case the maximum diameter of the oval opening means the mean of major and minor axes.

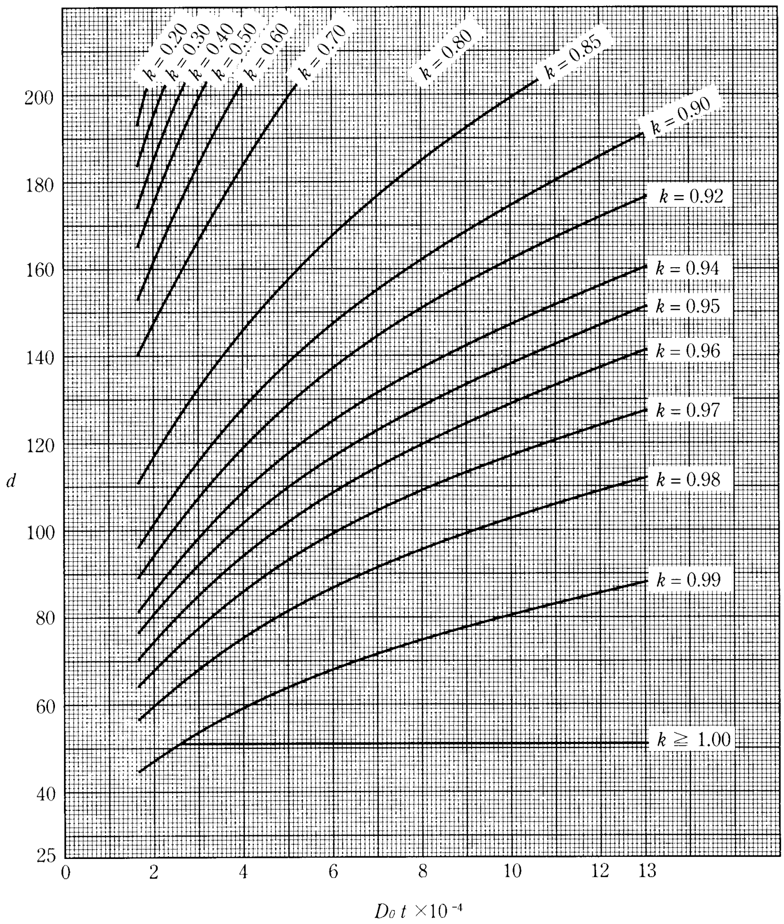
D_0 : Outside diameter of the shell (mm)

t : Actual thickness of the shell plate (mm)

$$k = \frac{PD_0}{1.82ft}$$

Fig. D9.8 Maximum Diameter of Openings Provided on the Shell for which Reinforcement may be Omitted (continued)

(B)



Notes:

d : Maximum diameter of openings (mm) which are not required to be reinforced, in this case the maximum diameter of the oval opening means the mean of major and minor axes.

D_0 : Outside diameter of the shell (mm)

t : Actual thickness of the shell plate (mm)

$$k = \frac{PD_0}{1.82ft}$$

9.6.3 Reinforcing Procedures of Openings

1 The meanings of the symbols used in **9.6.3** are as follows:

a : Area of the shell or end plate available for reinforcement (mm^2)

A_0 : Required cross sectional area of the reinforcement (mm^2)

d_1 : Diameter of the opening in the cross section where reinforcement is intended (mm)

d_0 : Maximum diameter of the finished opening in the longitudinal cross section of the shell plate or in the cross section of the end plate (mm)

h : Depth of the flange measured along the major axis of opening from the outer surface of end plate (mm)

t_n : Actual thickness of the nozzle (mm)

t_{nr} : Required thickness of the nozzle (mm)

T : Actual thickness of the shell plate or end plate (mm)

T_0 : Required thickness of the shell plate or of the blank end plate (mm) calculated by assuming an efficiency of 1.00. However, where the opening and its reinforcement are entirely within the spherical portion of a dished end plate, T_0 is the thickness required for a hemispherical end plate having the equal radius to the spherical portion of the end plate. In addition, where the opening and its reinforcement of a semi-ellipsoidal end plate and are located entirely within a circle on the end plate with the diameter of the circle taking 80% of the inside diameter of the shell, T_0 is the thickness required for a hemispherical end plate of a radius equal to 90% of the inside diameter of the shell.

2 For openings in shell plates and formed end plates, reinforcement is to be provided in such a manner that the area of its cross section through the centre of the opening and normal to the surface of the opening is not less than that calculated by the following formula:

$$A_0 = d_0 T_0$$

3 In cases where flat end plates or cover plates specified in **9.5.5** have openings, they are to comply with the following:

(1) In cases where flat end plates or cover plates have openings with a diameter not exceeding one-half of the diameter for the circular plates or the minimum length (d shown in **Fig. D9.11** and **Fig. D9.12**) for non-circular plates, the end plates or cover plates are to have a total cross sectional area of reinforcement not less than that calculated by the following formula:

$$A_0 = 0.5 d_0 T_0$$

(2) In cases where flat end plates or cover plates have openings with a diameter exceeding one-half of the diameter for the circular plates or the minimum length (d shown in **Fig. D9.11** and **Fig. D9.12**) for non-circular plates, the thickness of end plates or cover plates is to be 1.5 times the required thickness specified in **9.5.5** except for the corrosion allowance.

4 Reinforcement is to be provided within its effective limit. The effective limit of reinforcement is the range on a vertical plane to the wall containing the centre of the opening that is enclosed by two lines along the wall and also by two lines parallel to the axis of the opening. The lengths of the four lines are as follows: (*See* **Fig. D9.9**)

(1) The length of lines measured along the wall is to be measured, in both directions from the centre of the opening and is to be equal to the greater of the following:

(a) The diameter of the finished opening in the cross section (mm)

(b) The radius of the finished opening in the cross section plus the thickness of the wall plus the thickness of the nozzle wall (mm)

(2) The length of the lines measured parallel to the axis of the opening from each surface of the wall is to be equal to the smaller of the following (mm)

(a) 2.5 times the thickness of the wall (mm)

(b) 2.5 times the thickness of the nozzle wall plus the thickness of any added reinforcement exclusive of any welded metal

5 Any part of the shell, end plate or nozzle that exceeds its required thickness as calculated according to the requirements in **9.5** as well as any deposit metal for welding may be considered as part of the reinforcement, provided that it lies within the effective limit of reinforcement. In this case, the area of the shell or end plate available for reinforcement is to be the area calculated by the following formulae, whichever is greater.

$$a = d_1(T - T_0)$$

$$a = 2(T - T_0)(T + t_n)$$

6 In cases where the allowable stress of the reinforcing material differs from that of the material used for the shell, correction is to be made by the following formula:

$$K_R = \frac{f_R}{f_S}$$

where

K_R : Coefficient to be multiplied with the area of reinforcement. This is not to exceed 1.0.

f_S : Allowable stress of the material used for the shell (N/mm^2)

f_R : Allowable stress of the reinforcement (N/mm^2)

7 Openings in the end plate may be reinforced by flanged-in. In this case, the depth of the flange is not to be less than the value calculated by the following formula:

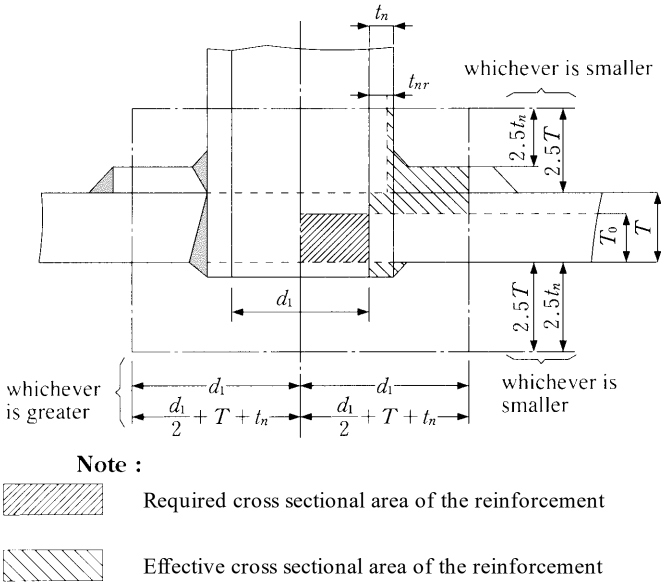
In cases where the thickness of the plate is not greater than 38mm;

$$h = 3T_0$$

In cases where the thickness of the plate is greater than 38mm;

$$h = T_0 + 76$$

Fig. D9.9 Effective Limit of Reinforcement



9.7 Tubes

9.7.1 Fitting of Tubes

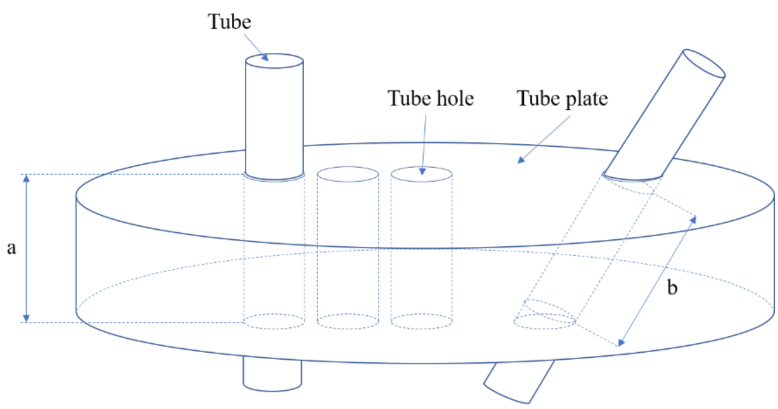
1 Tubes are to be attached to the tube plate by expanding or another suitable method and the tubes are to project through the neck or belt of the parallel seating by not less than 6*mm*, except for those attached by welding. In cases where the tube end is fitted by welding, consideration is to be given for preventing the deformation (thermal ratchet effect) of tubes due to tube-to-tube differentials in thermal expansion.

2 In cases where water tubes are secured against being pulled out by means of bellmouthing only, the included angle of belling is to be not less than 30 *degrees*.

3 Tube holes are to be formed so that the tubes can be effectively tightened inside them. Where the tubes are practically normal to the tube plates, the parallel seating of the holes is not to be less than 10*mm* in depth(a). Where the tubes are not normal to the tube plate, the depth of the holes perpendicular(b) to the tube plate is not to be less than 10*mm* for tubes not exceeding 60*mm* in outside diameter, and not to be less than 13*mm* for tubes exceeding 60*mm* in outside diameter. (refer to **Fig. D9.10**)

4 In horizontal smoke tube type vertical boilers, each alternate smoke tube in the outer vertical rows of tubes is to be a stay tube.

Fig. D9.10 Depth of Seating Tubes



9.7.2 Minimum Thickness of Tubes

The thickness of tubes used for boilers is not to be less than 2mm for any tubes with an outside diameter less than 30mm, or 2.5mm for any tubes with an outside diameter of 30mm or more.

9.7.3 Required Thickness of Smoke Tubes

The required thickness of smoke tubes is to be calculated by the following formula:

$$T_r = \frac{Pd}{70} + 2$$

where

d : Outside diameter of the smoke tube (mm)

9.7.4 Required Thickness of Tubes subjected to Internal Pressure

The required thickness of tubes (evaporating tubes, water wall tubes, downcomers, superheater tubes, economizer tubes and exhaust gas economizer tubes, etc.) subjected to internal pressure is to be calculated by the following formula:

$$T_r = \frac{Pd}{2f + P} + 1.5$$

where

d : Outside diameter of the tube (mm)

9.8 Joints and Connection of Each Member

9.8.1 Welded Joints

1 The dimensions and shapes of edge preparation are to be such that satisfactory penetration is obtainable without failure. The welded joint is to be so designed as not to be subjected to excessive bending stress. Where the construction is such that bending stress is concentrated at the root of the welded joints due to deformation caused by bending, single welded butt joints of fillet welded joints are to be avoided.

2 In cases where plates of unequal thickness are jointed by butt welding, the thicker plate is to be reduced in thickness to a taper of a distance not less than 4 times the offset so that the two plates are of equal thickness at the portion of the weld. In this case, the taper may be made only on one side for circumferential joints of shells. However, for longitudinal joints, as a rule, the taper is to be made on both sides so that the centre lines of both plates may coincide. In cases where the reduction in thickness is made on one side of the longitudinal joints, the distance between the centre line of the weld and the origin of the taper is not to be less than the thickness of the thinner plate.

3 The circumferential joints and the longitudinal joints of shells are to be double welded butt joints, or be single-welded butt joints that have been approved by the Society.

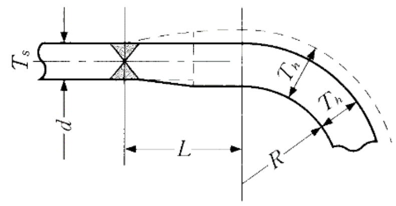
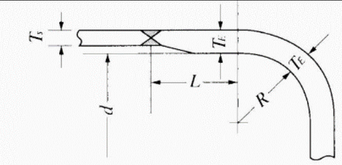
9.8.2 Shapes of Joints and Connections

The shapes of welded joints and connections are to be as shown in **Fig. D9.11,** or be of an equivalent shape that has been approved by the Society.

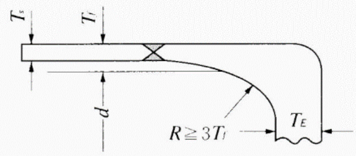
9.8.3 Construction of Bolted Cover Plates

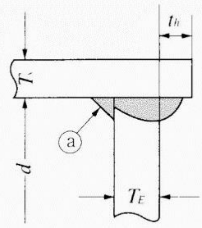
The construction of unstayed flat cover plates bolted to shells is to be as shown in **Fig. D9.12** or be of an equivalent construction that has been approved by the Society.

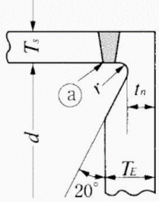
Fig. D9.11 Examples of Welded Joints Approved for Each Case

<table>
<tr><th>Welding part</th><th>Symbol</th><th>Welding mode and value of constant C_1</th><th>Remarks</th></tr>
<tr><td>(1) Welding joint between formed end plate and shell</td><td>A</td><td>
</td><td>L $\geq 3T_h$, but need not be more than 38 mm
Where $T_h \leq 1.25T_s$, the above-mentioned value may be reduced.</td></tr>
<tr><td rowspan="4">(2) Welding joint between flat end plate or cover plate and shell</td><td>A</td><td>

In case L is not restricted,
C_1 = 0.50 (circular or non-circular)
$R \geqq 3T_E$

In case $L \geqq (1.1 - 0.8 \times \frac{T_S^2}{T_E^2})\sqrt{dT_E}$
C_1 = 0.39 (circular only).</td><td></td></tr>
<tr><td>B</td><td>

C_1 = 0.50 (circular or noncircular)</td><td>$T_f \geq 2T_s$</td></tr>
<tr><td>C</td><td>

C_1 = 0.70
(circular or noncircular)</td><td>(1) $T_s \geq 1.25T_{ro}$
(2) $t_h \geq T_s$
(3) Where the welding of part ⓐ, is considered difficult, the backing strip is to be used or the welding process, which ensures a good penetration to the root, is to be employed.</td></tr>
<tr><td>D</td><td>

C_1 = 0.55 (circular)
C_1 = 0.70 (noncircular)</td><td>(1) $r \geq 0.2T_E$, but not less than 5 mm
(2) $t_n \geq 1.25T_{ro}$
(3) In welding the part ⓐ, such a welding process as to have a good penetration to the root, is to be employed.
(4) End plates or cover plates are to be made of forged steel</td></tr>
</table>

Fig. D9.11 Examples of Welded Joints Approved for Each Case (continued)

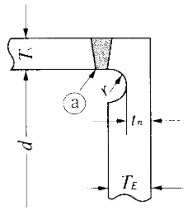
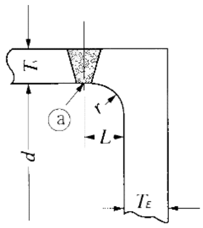
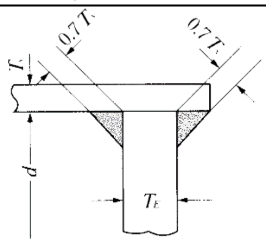
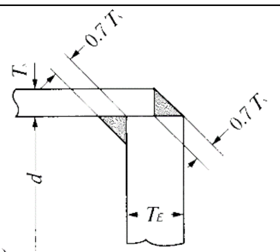
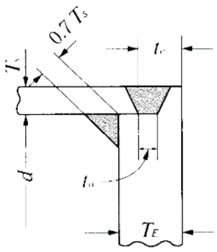
Welding part	Symbol	Welding mode and value of constant C_1	Remarks
(2) Welding joint between flat end plate or cover plate and shell	*E*	 $C_1 = 0.55$ (circular) $C_1 = 0.70$ (noncircular)	(1) $r \geq 0.2T_E$, but not less than 5 mm (2) $t_n \geq 1.25T_{ro}$ (3) In welding the part ⓐ, such a welding process as to have a good penetration to the root, is to be employed. (4) End plates or cover plates are to be made of forged steel.
	F	 $C_1 = 0.55$ (circular) $C_1 = 0.70$ (noncircular)	(1) $r \geq 0.3T_E$ (2) $L \geq T_E$ (3) For the part ⓐ, the same is required as above. (4) End plates or cover plates are to be made of forged steel.
	G	 $C_1 = 0.55$ (circular) $C_1 = 0.70$ (noncircular)	$T_s \geq 1.25T_{ro}$
	H	 $C_1 = 0.55$ (circular) $C_1 = 0.70$ (noncircular)	$T_s \geq 1.25T_{ro}$
	I	 $C_1 = 0.55$ (circular only)	(1) $T_s \geq 1.25T_{ro}$ (2) $t_a \geq T_s$, but need not be over 6.5 mm. (3) t_e is not be less than $2T_{ro}$ or $1.25T_s$, whichever is the greater.

Fig. D9.11 Examples of Welded Joints Approved for Each Case (continued)

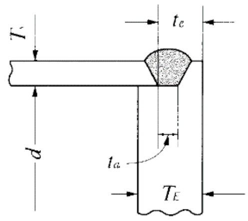
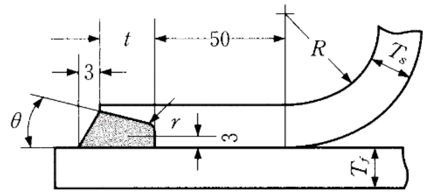
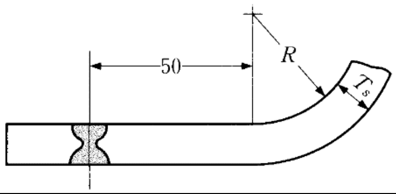
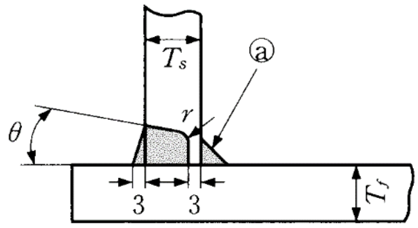
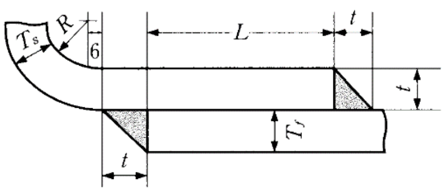
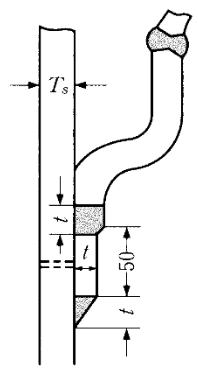
Welding part	Symbol	Welding mode and value of constant C_1	Remarks
(2) Welding joint between flat end plate or cover plate and shell	J	 C_1 = 0.70 (circular or noncircular)	(1) Tube headers only. (2) $T_s \geq 1.25T_{ro}$ (circular only) (3) $t_a \geq T_s$, but need not be over 6.5 mm (4) t_e is not be less than $2T_{ro}$ or $1.25T_s$, whichever is the greater.
(3) Welding joint between furnace and shell plate or end plate	A		(1) To be applied to welding joint on the front side of boiler. (2) $t \geq T_s - 3$ (3) θ ranges between 10°and 20° inclusive. (4) $10 \geq r \geq 5$
	B		
	C		(1) To be applied to welding joint on the front side of boiler. (2) The part ⓐ is to be of light fillet weld（throat thickness 4～6 mm）。 (3) θ ranges between 10°and 20° inclusive. (4) $10 \geq r \geq 5$
	D		(1) To be applied to welding joint on the front side of boiler. (2) $t \geq T_f$ (3) $L \geq 2T_s$
(4) Welding joint between ogee ring and shell plate	A		$t \geq T_s$

Fig. D9.11 Examples of Welded Joints Approved for Each Case (continued)

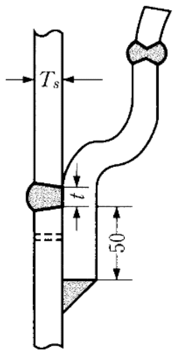
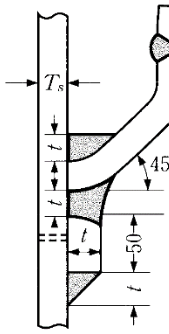
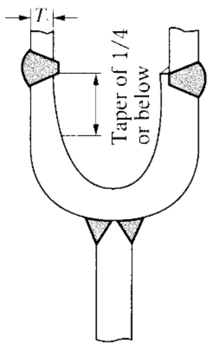
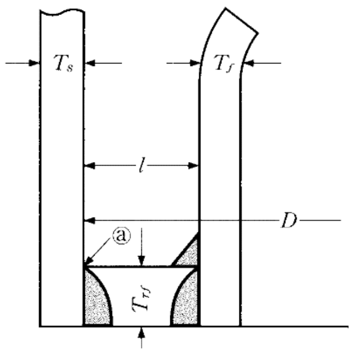
Welding part	Symbol	Welding mode and value of constant C_1	Remarks
(4) Welding joint between ogee ring and shell plate	*B*		$t \geq T_s$
	C		$t \geq T_s$
	D		$t \geq T_s$
	E		(1) If $D \leq 750,\ l \geq 50$. If $D > 750,\ l \geq 60$. (2) In welding the part ⓐ, such a welding process as to have a good penetration to the root, is to be employed.

Fig. D9.11 Examples of Welded Joints Approved for Each Case (continued)

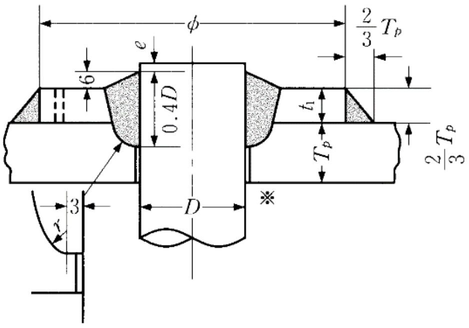
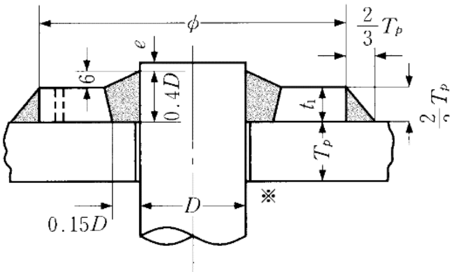
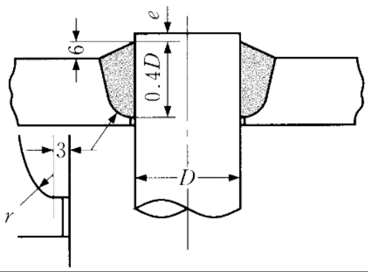
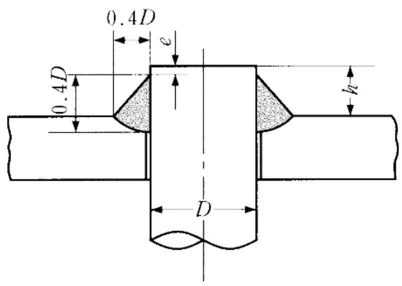
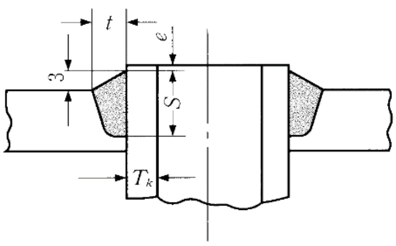
Welding part	Symbol	Welding mode and value of constant C_1	Remarks
(5) Welding joint between stay and tube plate or end plate	*A*		(1) $\emptyset \geq \frac{2}{3}P$ (P means the pitch of stays, hereinafter the same being referred) (2) $t_1 \geq \frac{2}{3}T_p$ (3) The part marked by ※ is to be applied with light fillet welding (root thickness, 4～6 mm) or caulking from the side of plate for filling the gap. (4) On the fire side, to be $e \leq 1.5$
	B		(1) $\frac{2}{3}P > \emptyset \geq 3.5D$ (2) $t_1 \geq \frac{2}{3}T_p$ (3) The part marked by ※ is to be same as above. (4) On the fire side, to be $e \leq 1.5$
	C		On the side exposed to flame, $e \leq 1.5$
	D		On the side exposed to flame, $h \leq 10$ and $e \leq 1.5$
(6) Welding joint between stay tube or tube and tube plate or end plate	*A*		(1) $t \geq T_k$ (2) $S \geq 2t$ (3) On the side exposed to flame, $e \leq 1.5$

Fig. D9.11 Examples of Welded Joints Approved for Each Case (continued)

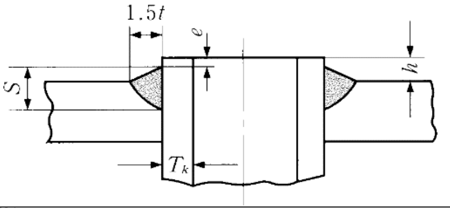
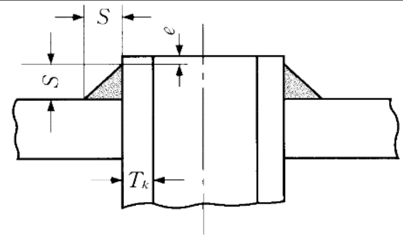
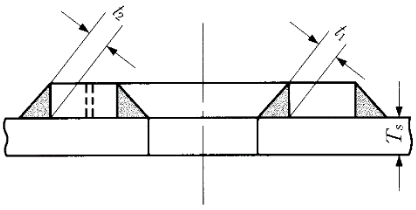
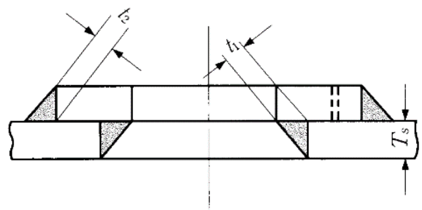
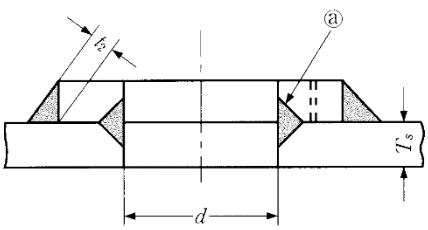
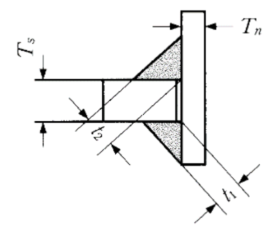
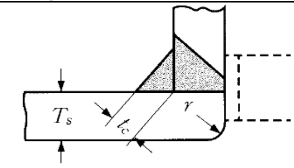
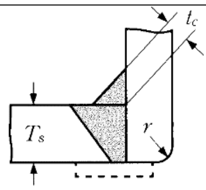
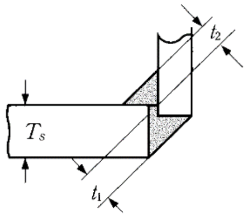
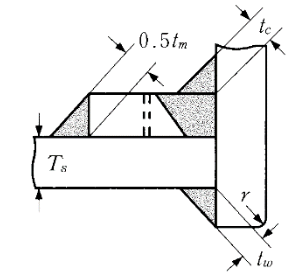
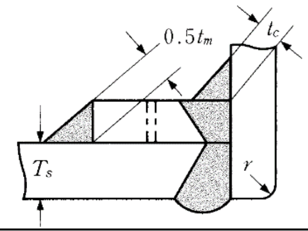
Welding part	Symbol	Welding mode and value of constant C_1	Remarks
(6) Welding joint between stay tube or tube and tube plate or end plate	*B*		(1) $t \geq T_k$ (2) $S \geq 1.5t$ or $t+3$ On the side exposed to flame, $h \leq 10$ and $e \leq 1.5$
	C		(1) $S \geq T_k + 3$ (2) To conduct tube expansion either before or after welding. (3) On the side exposed to flame, $e \leq 1.5$
(7) Welding joint between seat or reinforcement ring and shell plate or end plate	*A*		(1) $t_1 + t_2 \geq 1.25t_m$ (2) t_1, $t_2 \geq \frac{1}{3}t_m$ but the minimum is 6.5 *mm*
	B		
	C		(1) To be applicable only for the case of $d < 60$. (2) $t_2 \geq 0.7t_m$ (3) The part ⓐ is to be welded for stopping leakage.
(8) Welding joint between nozzle and shell plate or end plate	*A*		(1) $t_c \geq 6.5$ or $0.7t_m$, whichever is the smaller (2) $t_1 + t_2 \geq 1.25t_m$ (3) t_1, t_2 : 6.5 *mm* or $0.7t_m$ whichever is the smaller.

Fig. D9.11 Examples of Welded Joints Approved for Each Case (continued)

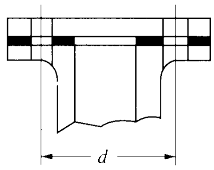
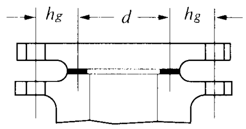
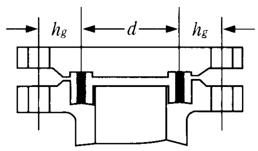
Welding port	Symbol	Welding mode and value of constant C_1	Remarks
(8) Welding joint between nozzle and shell plate or end plate	*B*		(1) $t_c \geq 6.5$ or $0.7t_m$, whichever is the smaller (2) $t_1 + t_2 \geq 1.25t_m$ (3) t_1, t_2 : 6.5 *mm* or $0.7t_m$ whichever is the smaller.
	C		
	D		
	E		(1) $t_c \geq 6.5$ or $0.7t_m$, whichever is the smaller (2) $t_1 + t_2 \geq 1.25t_m$ (3) t_1, t_2 : 6.5 *mm* or $0.7t_m$ whichever is the smaller. (4) $t_w \geq 0.7t_m$
	F		

Notes:

1. Constant C_1 is the value used for the formula in **9.5.5**.
2. The dimensions of welded parts are their minimum values.
3. The unit of all values in the figures is in *mm*.
4. The definitions of the symbols used in the figures are as Follows (units: *mm*):

T_s : Actual thickness of the shell plate
T_h : Actual thickness of the formed end plate
T_E : Actual thickness of the flat end plate or cover plate
T_{ro} : Required thickness of the seamless shell
T_p : Actual thickness of the tube plate or flat end plate (formed end plate)
T_{rf} : Required thickness of the furnace foundation ring plate
T_k : Actual thickness of the stay tube or tube
T_n : Actual thickness of the nozzle
t_m : Smaller value of the thickness of plates to be welded, but the maximum value is 20 *mm*.

Fig. D9.12 Examples of Bolting Covers and End Plates

Joining method	Shape and dimensions	C_3
Bolted with full face gasket		0.25
Bolted	 	0.3

9.9 Fittings, etc.

9.9.1 Materials of Fittings*

1 The material of the nozzles, flanges or distance pieces that attach directly to a boiler drum (including tube headers) is to be of steel which is suitable for the working temperatures.

2 Except for those specified in **-1**, the material of valve boxes or other fittings which are connected to a boiler and are subjected to its pressure is to be of steel which is suitable for the working temperature. However, the following cases are excluded:

(1) Copper alloy castings may be used in cases where the maximum working temperature does not exceed 210°C.

(2) Grey cast iron may be used in cases where the maximum working temperature does not exceed 220°C and the approved design pressure does not exceed 1*MPa*, except for blow-off valves.

(3) Special cast iron made by approved manufacturers may be used in cases where the maximum working temperature does not exceed 350°C and the approved design pressure does not exceed 2.5*MPa*.

9.9.2 Construction of the Fittings*

1 Fittings such as valves, flanges as well as bolts, nuts, gaskets, etc. are to have a construction and dimensions conform to the recognized standards of the Society. In addition, they are to conform to the service conditions specified in such standards.

2 Manual stop valves are to be provided with an indicator to show whether it is open or closed, except for rising-stem type valves.

3 Fittings are to be attached to boiler drums with flanged joints or by welding. However, in cases where the thickness of the drum is over 12*mm* or in cases where a seat for screwing is fitted onto the drum, fittings of 32*mm* or under in nominal diameter may be attached to the boiler by screwing.

4 In cases where boiler fittings are secured by studs, the stud holes are not to penetrate the whole thickness of the shell, and the depth of threaded part is not to be less than the diameter of the studs.

9.9.3 Safety Valves and Relief Valves*

1 Each boiler is to be provided with at least two spring loaded safety valves. However, only one safety valve will be accepted for the following types of boilers:

(1) Boilers with heating surface of less than $10m^2$.

(2) Boilers with an approved design pressure of not more than 1*MPa*, provided that they are equipped with a pressure controlling device and a device which cuts off the fuel supply automatically at a pressure not exceeding the approved design pressure.

(3) Exhaust gas boilers fitted with the relief valves specified in **-11.**

2 Safety valves that are attached with a spring pilot valve may be used in lieu of spring loaded safety valves.

3 The seat diameter of safety valves is not to be less than $25mm$, unless specifically approved.

4 Safety valves are to start releasing steam automatically at a set pressure in accordance with the requirements in **-14** and are to be capable of discharging the total evaporative capacity of the boiler under the maximum designed operating condition without raising steam pressure to 10% or more above the approved working pressure of the boiler.

5 The total area of safety valves in consideration of the maximum designed evaporation of the boiler is not to be less than the required area which is calculated under each steam condition and for each type of safety valves specified below. However, safety valves of boilers having a superheater are to comply with the requirements in **-7**, **-8** and **-9**. Furthermore, for any boiler with an exhaust gas economizer which is so designed that it may be additionally heated while in use, the required area of the safety valves is to be calculated after the maximum evaporation of the boiler is added with the evaporation of the exhaust gas economizer.

(1) For saturated steam

(a) For low lift valves $(\frac{D}{24} \le L < \frac{D}{15})$:

$$A = \frac{W}{K_1(1.03P + 0.1)} \times 10^{-2}$$

(b) For high lift valves $(\frac{D}{15} \le L < \frac{D}{7})$:

$$A = \frac{W}{K_2(1.03P + 0.1)} \times 10^{-2}$$

(c) For full lift valves $(\frac{D}{7} \le L)$:

$$A = \frac{W}{K_3(1.03P + 0.1)} \times 10^{-2}$$

(d) For full bore valves (Diameter of the seat is 1.15 times or more the diameter of the throat):

$$A' = \frac{W}{K_4(1.03P + 0.1)} \times 10^{-2}$$

where

D : Seat diameter of a safety valve (mm)

L : Lift of a safety valve (mm)

A : Required seat area of a safety valve (mm^2)

A' : Required nozzle throat area of a safety valve (mm^2)

W : Maximum designed evaporation capacity of a boiler (g/h)

P : Set pressure of a safety valve (MPa)

K_1 : 4.8

K_2 : 10.0

K_3 : 20.0

K_4 : 30.0

However, if the tests and examinations designated by the Society, such as a discharge capacity test and a measurement of lift have been carried out on each prototype under conditions equivalent to those of actual operation, the values of K_2, K_3 or K_4 may be increased to the values approved by the Society on the basis of these results.

(2) For superheated steam

$$A_S = \frac{A}{\sqrt{V_H / V_S}}$$

where

A_S : Required seat area of a safety valve (mm^2)

A : As specified in **(1)**

V_H : Specific volume of saturated steam (mm^3/g)

V_S : Specific volume of superheated steam (mm^3/g)

6 The area of steam passages of safety valve is to be of the following value for each type of safety valves.

(1) For steam passages of low lift safety valves, the minimum area at the chest inlet and at the outlet is not to be less than 0.5 times

and 1.1 times the required valve seat area respectively.

(2) For steam passages of high lift safety valves, the minimum area at the chest inlet is not to be less than the required valve seat area at the chest inlet; and not less than 2 times the required valve seat area at the outlet.

(3) For steam passages of full lift safety valves, the minimum area at the chest inlet and outlet is not to be less than 1.1 times and 2 times the steam passage area respectively when the valve is lifted to 1/7 of the valve seat diameter.

(4) For steam passages at the valve seats of full bore safety valves, the minimum area is not to be less than 1.05 times the area at the throat, when the valve is open. Furthermore, for steam passages at the valve inlet and the nozzle, the minimum area is not to be less than 1.7 times the area at the throat. And, for steam passages at the outlet, the minimum area is not to be less than 2 times the area at the valve seat when the valve is open.

7 In cases where a boiler is provided with a superheater, at least one safety valve is to be fitted at the outlet of the superheater.

8 The discharge capacity of any safety valve attached to a superheater is to be such that the superheater is not damaged when the main steam supply is shut down in an emergency as well as when the boiler is being operated under the stress of maximum continuous output. In cases where this purpose is not fulfilled, means are to be provided to automatically shut off or to control the fuel supply to the boiler in an emergency in order to prevent the superheater from damage.

9 In cases where there is no intervening device between a superheater and a boiler, the area of the superheater safety valve may be included in the total area of the safety valves of the boiler. However, the total area of any safety valves fitted to the evaporating parts of the boiler is not to be less than 0.75 times the required area calculated by the formulae in -5.

10 Safety valves are to be fitted at the inlets and the outlets of independent reheaters or independent superheaters respectively and the total discharge capacity of these valves is not to be less than the maximum passing steam quantity. The total discharge capacity of the safety valves fitted at outlets is to be not less than the quantity necessary to keep the steam temperature of the independent reheater or independent superheater from not exceeding its designed value. However, for independent superheaters which are connected directly to the boiler and designed with the same approved design pressure as that of the boiler drum, one safety valve that is capable of discharging the quantity of steam necessary to keep the steam temperature of the independent superheater from exceeding its designed value may be fitted at its outlet.

11 In cases where economizers and exhaust gas economizers (including the heating element of the exhaust gas boiler) are equipped with an intervening valve between the boiler and the economizer or exhaust gas economizer, they are to be provided with at least one relief valve capable of discharging a quantity not less than that calculated from the maximum absorbable energy. However, shell type exhaust gas economizers which have a total heating surface of $50m^2$ or more are to be provided with at least two relief valves.

12 The construction of safety valves and relief valves is to comply with the following requirements:

(1) Safety valves and Relief valves are to be so constructed that the spring and the valve are housed in a cage so that they can not be overloaded intentionally from outside, and that in case of spring failure they will not come out of their cage.

(2) Safety valves and Relief valves are to be fitted to boiler shells, headers, or outlet connections of a superheater by flanged or welded joints. The valve chests of these safety valves are not to be also used as the valve chests for other valves. However, safety valves of superheaters may also be fitted by attaching them to the flanges of any distance pieces welded to the outlet connection.

(3) Safety valves and Relief valves are to be provided with easing gears and their handles are to be so arranged that they can be operated from an accessible place that is free from danger.

(4) The housings of safety valves, relief valves or waste steam pipes are to be fitted with drainage arrangements from their lowest part, directed so that any drainage will continuously fall to a position clear of the boiler or exhaust gas economizer where it will not pose any threat to either personnel or machinery. No valves or cocks are to be fitted in these drainage arrangements.

13 Waste steam pipes for safety valves and relief valves are to comply with the following requirements:

(1) Waste steam pipes for safety valves and relief valves are to be of such construction that back pressure does not interfere with operation of the valves. Waste steam pipes are to have an inside diameter that is not to be less than the diameter of the valve outlet and the design pressure of these pipes is to be 1/4 or more than the set pressure of the valves.

(2) In cases where a common waste steam pipe is provided for two or more safety valves or relief valves, its cross sectional area is not to be less than the aggregate area of the steam passages of each safety valve or relief valve. However, waste steam pipes of boiler safety valves are to be separated from any pipe lines likely to contain a large amount of drainage such as steam blow-off pipes to the atmosphere or waste steam pipes of relief valves for exhaust gas economizers.

14 Safety valves or relief valves are to be set in accordance with the following requirements **(1)** to **(5)** after their installation on board the ship:

(1) Safety valves are to be set to blow-off steam automatically at a pressure not greater than 1.03 times the approved working pressure of the boiler.

(2) Superheater safety valves are to be set to blow-off steam automatically at a pressure not greater than the value obtained by subtracting the set pressure of the safety valve(s) on the boiler drum by the value of 0.035*MPa* plus the steam pressure drop in the superheater at the normal load. However, in cases where this pressure exceeds 1.03 times the nominal pressure of the boiler, at least one safety valve is to be set to blow-off steam at a pressure not greater than 1.03 times the nominal pressure, and the other valve(s) at a pressure not greater than 1.05 times the nominal pressure.

(3) The blow-off pressure of a safety valve at the outlet of a superheater is to be set lower than that at the inlet.

(4) The blow-off pressure of relief valves provided on an economizer or an exhaust gas economizer is to be set at a pressure not greater than the respective approved design pressure.

(5) Safety valves or relief valves are to function satisfactorily while blowing-off at the set pressure specified by the respective requirements **(1)** to **(4)**.

15 In cases where the calculated discharge capacity of the safety valves do not comply with the requirements in **-5** on account of the reduction of the approved working pressure of the boiler, it may be accepted, provided that an accumulation test deemed appropriate by the Society has been carried out and it has been confirmed that the pressure in the boiler drum has not exceeded 110% of the approved working pressure.

9.9.4 Steam Connections

1 A stop valve is to be fitted directly to the boiler drum at each steam outlet.

2 In cases where the steam from more than two boilers is led to one common steam pipe, the stop valve to be provided on each steam outlet as required in **-1** is to be a screw-down non-return type valve and one additional stop valve is to be provided on each steam pipe between the non-return valve and the point of steam pipe connection.

3 In ships provided with two or more main boilers or essential auxiliary boilers, steam lines are to be led in such a way that an uninterrupted steam supply to the auxiliary machinery for manoeuvring and the safety can be ensured even in cases when one of the boilers fails.

9.9.5 Feed Water Systems

1 A stop valve is to be fitted to the feed water connection and a screw-down non-return valve is to be provided at a point as close to the stop valve as practicable. An approved feed regulator may, however, be installed between the screw-down non-return valve and the stop valve.

2 Notwithstanding the requirement in **-1**, in cases where the boiler has an economizer which is recognized to be an integral part of the boiler, a feed water stop valve may be provided at the economizer inlet. In such cases, a screw-down non-return valve is to be provided at a point as close to the stop valve as practicable.

3 The part of the boiler drum where feed water is fed into is to be provided with sleeves or other suitable devices so that extreme thermal stress does not occur due to the direct contact of the cold feed water with the drum. This requirement also applies to the desuperheater in the boiler drum, if installed, where superheated steam pipes penetrate through the drum. Furthermore, feed water is to be fed into the drum so that it does not come into direct contact with any of the high temperature heating surfaces of the boiler drum.

9.9.6 Blow-off Systems

1 Each boiler is to be provided with a blow-off valve fitted directly to its drum so that boiler water may be discharged from the bottom of its water space. The nominal diameter of this blow-off valve is not to be less than 25*mm* and not to be more than 65*mm*. However, boilers with heating surface of $10m^2$ or less, may have a blow-off valve with a nominal diameter of 20*mm*.

2 In cases where blow-off pipes are exposed to the flue, they are to be protected by thermal insulation materials and be so arranged that they may be readily inspected.

3 The design pressure of blow-off piping is not to be less than 1.25 times the design pressure of the boiler drum.

4 Blow-off valves are to be of such construction that they are free from any deposits of scales and other sediments.

5 In cases where the blow-off pipes of two or more boilers are connected to one common discharge, a screw-down non-return valve is to be provided in each pipe line from each boiler.

9.9.7 Burning Systems*

1 Fuel Burners

(1) Fuel burners are to be so arranged that they cannot be withdrawn without shutting off the fuel supply to those burners.

(2) For top firing boilers, in order to absorb vibrations, flexible joints approved by the Society are to be provided at the connections between the boiler and the fuel supply pipe.

2 Draught Fans

Boilers are to be provided with draught fans of sufficient capacity to attain the designed maximum steam evaporation of the boiler and to ensure stable combustion in the boiler within its service range. In the case of failure of any one draught fan, an alternative means is to be provided to ensure normal navigation as well as a continuous flow of heat to any cargo that requires heating.

9.9.8 Water Level Indicators

1 Each boiler is to be provided with at least two independent water level indicators, one of which is to be a glass water gauge and the other which is to comply with either of the following requirements:

(1) A glass water gauge that is located at a position where the water level may be readily sighted.

(2) A remote water level indicator

2 For forced circulation or once-through boilers, where water level indication is difficult according to the requirements in **-1**, a suitable level detector and a low water level safety device which is comprised of two detectors so designed as to prevent the overheating of any part of the boiler by lack of water supply are to be provided.

3 In cases where the water space in the boiler is long in the transverse direction of the ship or excessive differences in water level may occur, the water level indicators specified in **-1** are to be so arranged as to indicate the water levels at both ends of the water space.

4 The lowest visible part of a glass water gauge is to be not less than 50*mm* above the lowest critical water level. The visible range of a remote level indicator is to cover all ranges related to the water level control in the boiler.

5 Construction of water level indicators is to comply with the following requirements:

(1) Construction of glass water gauges is to be of the built-up rectangular-section box type (double-plate-type) specified in the recognized standards or the equivalent approved by the Society.

(2) In cases where the water gauge is placed outside the boiler, a stop valve (or cock) is to be fitted on the top and bottom of the gauge respectively. In addition, an effective draining device is to be provided.

(3) In cases where the water gauge or the water column is connected by a pipe to the boiler drum, a stop valve is to be fitted to the boiler drum.

(4) Stop valves (or cocks) for water gauges and connection pipes to boiler drums are to be of a shape that is free from any deposits of scale and other sediments from the boiler water.

(5) The water column to which the water gauge(s) are attached is to be strongly supported so that it may maintain its correct position. The inside diameter of the water column is not to be less than 45*mm* and a draining hole of sufficient size is to be provided at the bottom of the column.

(6) Connection pipes to boiler drums are to be 15*A* or over in nominal diameter for the water gauge, and 25*A* or over in nominal diameter for the water column.

(7) In cases where connection pipes from the water column to the boiler penetrate the uptake, they are to be enclosed all the way through the uptake. Furthermore, an air passage of not less than 50*mm* is to be provided around the pipes.

9.9.9 Pressure and Temperature Measuring Devices

1 Each boiler is to be provided with a set of pressure measuring devices at both the boiler drum and the superheater outlet respectively. Pressure indicators are to be provided in the monitoring station.

2 Pressure indicators are to be such that they have a scale of 1.5 times or over the set pressure of the safety valves. Both the approved working pressure of the drum and the nominal pressure of the superheater are to be specially marked on the pressure gauges.

3 Pressure measuring and indicating devices are to be operation while the boiler is in operation.

4 At the steam outlets of superheaters and reheaters, temperature measuring devices are to be provided.

9.9.10 Safety Devices and Alarm Devices

1 Fuel oil shut-off devices

Each boiler is to be fitted with a safety device which is capable of automatically shutting off the fuel supply to all burners in any of the following cases:

Alarms which indicate the action of the safety device are to be in accordance with **18.2.6-2**.

(1) When automatic ignition fails.

(2) When the flame vanishes (in this case, the fuel oil supply is to be shut-off within 4 *seconds* after the flame has been extinguished).

(3) When the water level falls.

(4) When the combustion air supply stops.

(5) When the fuel oil supply pressure to the oil burners falls at times of pressure atomization, or when the steam pressure to the burners falls at times of steam atomization.

(6) All other cases considered necessary by the Society.

2 Alarm devices

(1) Each boiler is to be provided with an alarm device which operates when the water level in the drum falls.

(2) In addition to the above, main boilers are to be provided with alarm devices which operate in the following cases:

(a) When there is a reduction in the combustion air supply, or when a draught fan stops.

(b) When the fuel oil supply pressure to the burner falls at times of pressure atomization, or when the steam pressure to burner falls at times of steam atomization.

(c) When the water level in the boiler drum reaches a high level.

(d) If a superheater is provided, when the steam temperature at the superheater outlet rises.

(e) When the exhaust gas temperature at the outlet of a gas type air preheater or economizer rises.

(3) For auxiliary boilers supplying steam to the turbines driving the main generators, alarm devices which operate when the water level in the boiler drum reaches a high level are to be provided in addition to those alarm devices given in **(1)**.

3 Water level detectors

The water level detectors of the devices specified in **-1(3)** are to be separate from those of the feed regulating system and the remote water level indicator specified in **9.9.8-1(2)**.

9.9.11 Monitoring of Boiler Water

1 Each boiler is to be provided with a boiler water sampling connection in a convenient location, but its sampling valve is not to be connected to the water column for the water gauge.

2 Boilers are to be provided with means such as water analyzer or other suitable devices to supervise and control the quality of the feed water and the boiler water.

9.9.12 Drainage Arrangements of Superheaters and Reheaters

Superheaters and reheaters are to be provided with effective drainage systems and means for preventing damages arising from any thermal stresses or thermal shocks caused by drains.

9.10 Tests

9.10.1 Shop Tests*

1 Tests for welds are to conform to the requirements specified in **Chapter 11**.

2 Boilers are to be subjected to hydrostatic tests at a pressure of 1.5 times the design pressure for boilers and at a pressure of 2 times the design pressure for boiler fittings that are not directly welded to boilers.

9.10.2 Tests after Installation On Board

Popping tests for the safety valves and function tests for the safety devices and alarm devices of boilers are to be carried out after the boiler has been installed on board.

9.11 Construction etc. of Small Size Boilers

9.11.1 General

Notwithstanding the requirements in 9.2 to 9.10, the requirements in 9.11 may be applied to boilers with a design pressure that does not exceed 0.35*MPa*.

9.11.2 Materials, Construction, Strength and Accessories of Small Boilers

1 The materials, construction and strength and accessories of small boilers are to comply with the requirements in recognized standard.

2 Small boilers are to be provided with safety valves or pressure relief piping of sufficient capacity.

3 Small boilers are to be provided with the following safety devices:

(1) Prepurging system for preventing the explosion of furnace gas.

(2) Fuel oil shut-off system which activates in cases of flame vanishing, automatic ignition failure or draught fan stoppage.

(3) Fuel oil shut-off system which activates when the pressure is not exceeding the approved working pressure of the boiler.

(4) Fuel oil shut-off system for preventing any overheating due to a lack of water supply.

9.11.3 Tests

1 Shop Tests

The pressure parts are to be subjected to hydrostatic tests at a pressure 2 times the design pressure or at 0.2*MPa*, whichever is greater.

2 Tests after Installation on Board

Function tests for the safety devices of small boilers that are specified, in 9.11.2-3 above are to be carried out after the boiler has been installed on board.

9.12 Construction of Thermal Oil Heaters

9.12.1 General

Thermal oil heaters heated by flame or combustion gas are to comply with the relevant requirements specified in 9.1 through 9.10 (in this case, the term "boiler" is to be read as "thermal oil heater") as well as the requirements in 9.12.

9.12.2 Safety Devices, etc. for Thermal Oil Heaters Heated by Flame

1 Temperature regulators are to be provided to control the temperature of the thermal oil within the predetermined range.

2 The master valve of the expansion tank is to always be kept open and the burning system is to be interlocked in such a way that it does not start when the master valve is closed.

3 Safety valves or pressure relief pipes of sufficient capacity are to be provided.

4 Discharge pipes from the safety valve of the pressure relief pipe specified in -3 are to have their open ends in the thermal oil tank that is of sufficient capacity.

5 The following safety devices are to be provided:

(1) Prepurging system for preventing the explosion of the furnace gas.

(2) Fuel oil shut-off systems which operate in the following cases:

  (a) When there is an abnormal increase in the thermal oil temperature.

  (b) When the flow rate of the thermal oil falls or when the pressure difference of the thermal oil between the inlet and outlet of the heater falls.

  (c) When there is an abnormal fall in the thermal oil level of the expansion tank.

9.12.3 Safety Devices, etc. for Thermal Oil Heaters Directly Heated by the Exhaust Gas of Engines*

1 Safety devices etc. are to comply with the requirements in 9.12.2-1, -3 and -4.

2 The master valve of an expansion tank is to normally be kept open and an interlocking device that prevents exhaust gas from entering into the heater when the master valve is closed is to be provided.

3 A shut-down device for exhaust gas is to be provided at the exhaust gas inlet of a thermal oil heater. In addition, it is to be so arranged that the engine can be operable even when the supply of the exhaust gas to the heater is shutdown.

4 Means are to be provided to prevent the leakage of any oil from thermal oil heaters and to prevent water used for fire fighting or others from flowing into the exhaust gas duct of the engine.

5 Stop valves are to be provided at the inlet and outlet of the thermal oil heater.

6 An audible-visual alarm is to be provided to warn on the following occasions and relayed any such warning to the monitoring-station.

(1) When a fire breaks out in the thermal oil heater.

(2) When the temperature of the thermal oil becomes abnormally high.

(3) When the thermal oil leaks within the thermal oil heater.

(4) When the flow rate of the thermal oil falls, or when the pressure difference of the thermal oil between the inlet and outlet of the heater decreases.

(5) When the liquid level in the expansion tank drops

7 A fixed fire extinguishing and cooling system as deemed appropriate by the Society is to be provided.

9.12.4 Thermal Oil Systems

The thermal oil systems for the thermal oil heaters are to comply with the requirements in **13.11**.

9.13 Incinerators

9.13.1 General

1 Notwithstanding the requirements in **9.2** to **9.12**, incinerators are to comply with the requirements in **9.13**. However, the requirements in **9.13** do not apply to the incinerators with maximum capacity less than 34.5 kW.

2 Notwithstanding **-1**, incinerators for oil or rubbish other than those produced by normal ship operation or the like will be specially considered.

9.13.2 Drawings and Data to be Submitted

Notwithstanding the requirements in **9.1.3**, drawings and data to be submitted are as follows:

(1) Drawings

(a) General arrangement of the incinerator

(b) Arrangement of the incinerator fittings

(c) Other drawings considered necessary by the Society

(2) Data

(a) Particulars

(b) Instruction manuals of safety devices

(c) Operation manual of the incinerator

(d) Other data considered necessary by the Society

9.13.3 Construction and Fittings*

The construction and fittings of incinerators are to comply with the requirements in the following **(1)** to **(9)**.

(1) Major parts of the combustion chamber are to be constructed out of effective material.

(2) Combustion chambers are to be so constructed as to ensure that harmful combustion gas or drainage will not leak.

(3) Uptakes from combustion chambers are to satisfy the following **(a)** to **(c)**:

(a) They are not to be connected to the exhaust gas pipes from reciprocating internal combustion engines and gas turbines.

(b) They are not to lead to such positions where any combustion gas might leak inboard.

(c) When connected to the uptakes from boilers, thermal oil heaters or other incinerators, they are to be subject to the recognition of the Society.

(4) Temperature measuring devices for combustion gas are to be provided.

(5) Fire doors for rubbish are to be arranged so that back-firing from the combustion chamber is prevented.

(6) Over-pressure preventive devices are to be provided to the water jackets of any incinerators equipped with a water jacket.

(7) Waste oil piping systems are to comply with the relevant requirements in **13.9**.

(8) Burning systems are to satisfy the following **(a)** to **(d)**:

(a) They are to be arranged so that the combustion chamber is prepurged by air before ignition.

(b) They are to be arranged so that the supply of fuel does not precede ignition spark in cases where an automatic ignition system is adopted.

(c) They are to be capable of controlling the amount of fuel supplied in cases where an automatic fuel supply system is provided.

(d) They are to comply with the requirements in **18.4.2-2(1)**, **(2)** and **(3)** in cases where an automatic combustion control device is provided.

(9) The location of the remote shut-off device for the incinerators is to comply with the requirements in **5.2.2-4, Part R**.

9.13.4 Safety Devices and Alarm Devices

1 Incinerators fitted with automatic fuel or waste oil supply systems are to be provided with a safety device to automatically stop the supply of fuel and waste oil to the burners in the following cases **(1)** and **(2)**:

(1) When the maximum working temperature of the furnace is exceeded

(2) When the flame vanishes

2 Incinerators are to be provided with alarm devices which operate in the following cases:

(1) When the approved working temperature of the furnace is exceeded

(2) When the flame vanishes

(3) When the power supply to the alarm device stops

(4) When cooling system, if any, stops

(5) When the waste oil supply pressure to the furnace falls, in the case of pressure atomizing

(6) When the fuel supply pressure to the furnace falls, in the case of pressure atomizing

(7) When combustion air supply system, if any, stops

9.13.5 Tests

Operation tests of the safety devices and the alarm devices specified in **9.13.4** as well as a burning test are to be carried out.

Chapter 10 PRESSURE VESSELS

10.1 General

10.1.1 Scope

1 The requirements in this Chapter apply to all liquid or gas containing vessels in which the internal pressure at the top of the vessel exceeds atmospheric pressure. These vessels include heat exchangers, but exclude those exposed to flame, combustion gas or hot gas.

2 For heat exchangers, etc. whose internal pressure does not reach atmospheric pressure, the relevant requirements in this Chapter apply (in this case a negative gauge pressure of the vessel is to be substituted for by a positive gauge pressure of the same value).

10.1.2 Design Pressures

The design pressure used for the strength calculations of the materials used for pressure vessels is not to be less than the following, whichever is the greatest:

(1) Approved working pressure specified in **2.1.21, Part A**.

(2) Maximum working pressure at maximum temperature (maximum working temperature) as designed by the manufacturer

(3) For pressure vessels of liquefied gases that are stored under a pressurized condition that is at or near atmospheric temperature, the following pressure requirement, whichever is the greatest, is to apply:

(a) Vapour pressure of the gas at 45 ℃

(b) Maximum working pressure

(c) 0.7 *MPa*

10.1.3 Classification of Pressure Vessels

1 Pressure vessels are classified into the following three groups in accordance with the thickness of their shell plates and their service conditions.

(1) Pressure vessels, Group I (*PV*-1)

Pressure vessels which conform to either one of the following:

(a) Shell plates exceeding 38 *mm* in thickness (*See* Note 1.)

(b) Design pressure exceeding 4 *MPa* (*See* Note 1.)

(c) Maximum working temperature exceeding 350 ℃

(d) Steam generators with a design pressure exceeding 0.35 *MPa*

(e) Vessels which contain inflammable high pressure gases having a vapour pressure not less than 0.2 *MPa* at 38 ℃ (*See* Note 2.)

Notes:

1. Pressure vessels which have shell plates that exceed 38 *mm* in thickness and/or a design pressure that exceeds 4 *MPa* are classified into “*PV*-2” provided that they are only subjected to hydraulic oil or water pressure at atmospheric temperature.
2. The requirements for “*PV*-2” apply to materials, construction and welding, when the pressure vessel has a capacity of 500 *litres* or less.

(2) Pressure vessels, Group II (*PV*-2)

Pressure vessels which conform to either one of the following:

(a) Shell plates exceeding 16 *mm* in thickness

(b) Design pressure exceeding 1 *MPa*

(c) Maximum working temperature exceeding 150 ℃

(d) Steam generators with a design pressure not exceeding 0.35 *MPa*

(3) Pressure vessels, Group III (*PV*-3)

Pressure vessels not included in Group I and II

2 The classification of those pressure vessels used for dangerous substances not specified in **-1** will be determined on a case by case basis, in accordance with the property of the substance, the service condition, etc.

10.1.4 Drawings and Data

Drawings and data to be submitted are generally as follows. However, for pressure vessels of Group III, no submission is required unless it is specifically requested by the Society.

(1) Drawings (with type and dimensions of materials specified)
- (a) General arrangement
- (b) Details of the shells
- (c) Arrangement of the pressure relief devices
- (d) Details of the washers for fittings and nozzles
- (e) Other drawings considered necessary by the Society

(2) Data
- (a) Principal particulars
- (b) Welding specifications (with welding procedures, welding consumables and welding conditions)
- (c) Other data considered necessary by the Society

10.2 Materials and Welding

10.2.1 Materials*

1 The materials used for the construction of the pressure parts of pressure vessels are to be adequate for their service conditions and are to comply with the requirements in the following **(1)** to **(3)**. However, when special materials are intended to be used, sufficient information related to the design and usage of these materials is to be submitted to the Society for approval.

(1) Pressure vessels, Group I (*PV*-1)

All materials are to comply with the requirements in **Chapter 3** to **Chapter 7, Part K** and they are to be tested in accordance with the requirements in **Chapter 1** and **Chapter 2** of the said Part.

(2) Pressure vessels, Group II (*PV*-2)

Same as those for Group I. However, for those pressure vessels which conform to either one of the following conditions, materials may be in accordance with the requirements in **(3)**.
- (a) Design pressure below 0.7 *MPa*.
- (b) Design pressure not exceeding 2 *MPa*, a maximum working temperature not exceeding 150 ℃ and an internal capacity not exceeding 500 *litres*.

(3) Pressure vessels, Group III (*PV*-3)

Materials complying with the requirements on the recognized standards are to be used.

2 Notwithstanding the requirements in **-1(1)** and **(2)**, the materials used for fittings such as valves, nozzles, etc. that are to be fitted to pressure vessels of Group I and Group II may be in accordance with the requirements in **-1(3)**, where approved by the Society with consideration given to their dimensions and service conditions.

10.2.2 Service Limitations of Cast Iron

1 Grey cast iron is not to be used for the shells of the following pressure vessels:

(1) Those that have a maximum working temperature which exceeds 220 ℃ or a design pressure which exceeds 1 *MPa*.

(2) Those that contain or handle flammable or toxic substances.

2 Special cast iron such as nodular graphite cast iron, etc. may be used, when approved by the Society, for those pressure vessels that have a maximum working temperature not exceeding 350 ℃ and a design pressure not exceeding 1.8 *MPa*.

10.2.3 Service Limitations of Materials Used for Fittings

The service limitation of materials to be used for fittings is to comply with the requirements in **9.9.1**. For the fittings of pressure vessels used to contain or to handle flammable or toxic substances, no cast iron is to be used unless approved by the Society.

10.2.4 Heat Treatment of Steel Plates

In cases where a heat treatment such as hot forming or stress relieving is to be carried out on steel plates during the manufacturing process, the manufacturer of the pressure vessels is to make clear this intention when ordering the materials.

What is expected of the manufactures of steel plates in these cases is specified in **3.3.4, Part K**.

10.2.5 Heat Treatment of Materials Subjected to Cold-forming

In cases where cold-forming is considered harmful to materials of the pressure vessels which are intended for use in an environment where things such as stress corrosion cracking, etc. are expected, suitable measures such as heat treatment are to be taken.

10.2.6 Non-destructive Testing for Cast Steels and Cast Irons*

1 Cast steel and cast iron used for the shells of pressure vessels of Group I subject to internal pressure are to be subjected to radiographic testing or ultrasonic testing as well as magnetic particle testing or dye penetrant testing in order to confirm that they are free from detrimental defects.

2 Cast steel and cast iron used for the shells of pressure vessels of Group II subject to internal pressure are to be subjected to adequate non-destructive testing in order to confirm that they are free from detrimental defects.

10.2.7 Welding

The workmanship of the welding of pressure vessels are to comply with the requirements in **Chapter 11**.

10.3 Design Requirements

10.3.1 Symbols

Unless expressly specified otherwise, the symbols used in this Chapter are as follows:

f : Allowable stress (N/mm^2) conforming to the requirements in **10.4.1-1**, **-2** or **12.2.1**

a : Corrosion allowance (mm) conforming to the requirements in **10.4.3**

T_r : Required thickness (mm) calculated by using design pressure. The allowable pressure means the pressure obtained by substituting the actual thickness for the required thickness

P : Design pressure (MPa)

J : Minimum value of the efficiency specified in **10.4.2**

R : Inside radius of the shell (mm)

R_{20} : Specified tensile strength at room temperature for the material concerned (N/mm^2)

E_{20} : Specified minimum yield point (or 0.2 % proof stress) at room temperature of material concerned (N/mm^2)

10.3.2 Design Loads*

1 The design of a pressure vessel is to take the following loads, in addition to any internal pressure, into account when it is considered to be necessary:

(1) Static head of contained fluid
(2) External pressure
(3) Dynamic loads caused by ship motion
(4) Thermal stress
(5) Loads from fittings
(6) Loads due to reactions exerting on supporting structure
(7) Hydrostatic test pressure loads
(8) Other loads or external forces exerted on the actual pressure vessels

2 If deemed necessary, fatigue analysis and crack propagation analysis are to be carried out in consideration of the loads specified in **-1**.

10.3.3 Pressure Vessels of Unusual Shapes

In cases where, due to the unusual shape of the part being subject to pressure, it is not appropriate to design a pressure vessel according to the requirements in **10.5** and **10.6**, any strain or deformations under a suitable load are to be measured with the approval by the Society. The Society will consider them as complying with the requirements in **10.5** and **10.6** after taking account of the results of these measurements.

10.3.4 Design Considerations

1 Pressure vessels for low temperature service are to have sufficient notch toughness for the lowest service temperature involved.

2 Pressure vessels used in an extremely corrosive environment are to be provided with effective corrosion control means.

3 Heat exchangers are to be provided with an effective sealing mechanism at the joints between tubes and tube plates as well as at joints between tube plates and the shell so as to prevent the two types of heat exchanging fluid from mixing together.

10.3.5 Considerations for Installation

1 Pressure vessels are to be so installed as to minimize the effects of ship motion, vibrations from the machinery installations, external forces exerted by piping and supports as well as thermal expansion due to temperature variation.

2 Pressure vessels and their fittings are to be installed at positions convenient for operation, repair and inspection.

10.4 Allowable Stress, Efficiency and Corrosion Allowance

10.4.1 Allowable Stress

1 The allowable stress of materials used at room temperature is to be determined by the following:

(1) Excluding cast steels, the allowable stress (f) of carbon steels (Including carbon manganese steel. Hereinafter, this definition applies throughout this Chapter) and low alloy steels, is to be the value obtained from the following formulae, whichever is smaller. For pressure vessels used for liquefied gas, the values of the denominators for f_1 and f_2 are to be 3.0 and 1.5, respectively.

$$f_1 = \frac{R_{20}}{2.7}\ ,\ f_2 = \frac{E_{20}}{1.6}$$

(2) The allowable stress of electric resistance welded steel tubes, except where they are used for the shells of pressure vessels, is to be the value specified in **(1)** when subjected to ultrasonic testing or any other compatible flaw detection approved by the Society for the entire length of the weld. In other cases, a value that is 85 % of the value specified in **(1)** is to be used.

(3) The allowable stress of cast steel is to be the value obtained by **(1)** multiplied by the coefficients given in **Table D10.1**.

(4) The allowable stress of cast iron is to be 1/8 of the specified minimum tensile strength.
However, the allowable stress of any special cast iron approved by the Society may be 1/6 of the specified minimum tensile strength.

(5) The allowable stress (f) of austenitic steel is to be obtained from the following f_1 or f_2, whichever is smaller.

$$f_1 = \frac{R_{20}}{3.5}\ ,\ f_2 = \frac{E_{20}}{1.5}$$

(6) The allowable stress (f) of aluminum alloy is to be obtained from the following f_1 or f_2, whichever is smaller.

$$f_1 = \frac{R_{20}}{4.0}\ ,\ f_2 = \frac{E_{20}}{1.5}$$

Table D10.1 Coefficients to be Multiplied to the Allowable Stress of Cast Steels

Type of test	Coefficient
When no radiographic test or any other alternative testing is carried out	0.7
When random a radiographic test or alternative testing is carried out	0.8
When the above tests are carried out all parts	0.9

2 For the allowable stress of materials used for pressure vessels for high temperature service, the requirements in **9.4.1** or the values deemed appropriate by the Society apply.

3 Allowable tensile stress is to conform to the requirements in **-1** and **-2**. However, the allowable tensile stress of bolts is to comply with the following requirements:

(1) In cases where bolts are used at room temperature, the value is to be obtained from the following **(a)** or **(b)**, whichever is smaller. However, for bolts complying with the requirements in the recognized standards the value may be 1/3 of the proof load specified therein.

(a) $\frac{R_{20}}{5.0}$

(b) $\frac{E_{20}}{4.0}$

(2) In cases where bolts are used at high temperatures, the value will be considered by the Society on a case by case basis.

4 Allowable bending stress is to comply with the following requirements:

(1) In cases where the materials are used at room temperature, the requirements in **-1** are to be complied with. However, for cast iron or cast steel, the value used is to be 1.2 times thereof.

(2) In cases where the materials are used at high temperatures, the value will be considered by Society on a case by case basis.

5 The allowable shearing stress for the mean primary shearing stress in the section subjected to shearing loads is to be a value that is 80 % of the allowable tensile stress.

6 The allowable compression stress in the cylindrical shell of pressure vessels used at room temperature that are subject to a load causing compression stress in the longitudinal direction is to be obtained from the following **(1)** or **(2)**, whichever is smaller:

(1) The value specified in **-1**

(2) The allowable buckling stress by the following formula:

$$\sigma_z = \frac{0.3ET_0}{D_m(1 + 0.004\frac{E}{E_{20}})}$$

where

σ_z : Allowable buckling stress (N/mm^2)

E : Modulus of longitudinal elasticity at room temperature (N/mm^2)

T_0 : Net thickness of a shell plate excluding any corrosion allowance from the actual shell plate (mm)

D_m : Average shell diameter (mm)

7 The allowable stress for various stresses of carbon steel or carbon manganese steel used for the shells of pressure vessels formed by a rotating unit when detailed calculations are carried out may be as follows:

$$P_m \leq f$$

$$P_L \leq 1.5f$$

$$P_b \leq 1.5f$$

$$P_L + P_b \leq 1.5f$$

$$P_m + P_b \leq 1.5f$$

$$P_L + P_b + Q \leq 3f$$

where

P_m : Equivalent primary general membrane stress (N/mm^2)

P_L : Equivalent primary local membrane stress (N/mm^2)

P_b : Equivalent primary bending stress (N/mm^2)

Q : Equivalent secondary stress (N/mm^2)

10.4.2 Efficiencies of Joints

The efficiency of joints is to be as follows:

(1) Seamless shells: 1.00

(2) Welded shells: As given in **Table D10.2**

(3) Where electric resistance welded steel tubes are used for the shell: As given in **Table D10.2(1)**

Table D10.2 Joint Efficiency of Welded Joints

Type of joint	Type of radiographic testing		
	Full radiographic testing carried out	Partial radiographic testing carried out	No radiographic testing carried out
(1) Double-welded butt joints or those butt welded joints considered by the Society to be equivalent	1.00	0.85	0.75
(2) Single-welded butt joints where the backing strip is left unremoved or those single-welded butt joints considered by the Society to be equivalent	0.90	0.80	0.70
(3) Single-welded butt joints other than those in (1) and (2) above	-	-	0.60
(4) Double-welded full fillet lap joints	-	-	0.55

Note:

Radiographic testing may be substituted for by ultrasonic testing if approved by the Society.

10.4.3 Corrosion Allowance

1 The corrosion allowance of materials used for strength calculation, except where they are subjected to extreme corrosion or wear and tear, is to be not less than 1.0 *mm* or 1/6 of the required thickness without the corrosion allowance for the inner surface, whichever is smaller.

In cases where corrosion resistance materials are used and effective corrosion control measures are taken or when there is no possibility of corrosion, this value may be reduced accordingly.

2 In cases where the outer surface of a pressure vessel which may suffer corrosion is provided with thermal insulation that prevents external inspection, an appropriate amount of corrosion allowance is also to be provided on the outer surface of the pressure vessel.

10.5 Strength

10.5.1 Minimum Thickness of Each Component

1 The thickness of shell plates and end plates is not to be less than 5 *mm* except where specifically approved by the Society with consideration given to the diameter, pressure, temperature, materials, etc. The thickness of formed end plates, except for full hemispherical end plates, is not to be less than the required thickness (calculated by assuming that the efficiency is 1.00) of the shell to which the end plate is welded.

2 The thickness of nozzles welded to pressure vessels is to comply with the following requirements. These requirements will be modified where approved by the Society with consideration given to the dimensions or shape, materials, etc.

(1) The thickness is not to be less than either the value 2.5 *mm* added to 1/25 of the outside diameter of the nozzle or the value calculated by the formula in **10.5.2-2**. However, this value need not be more than the thickness of the shell at which the nozzle is welded.

(2) Notwithstanding the requirement in **(1)**, for Groups II and III pressure vessels the value need not be more than 4 *mm*, if it is not less than the value calculated by the formula in **10.5.2-2**.

10.5.2 Strength of Shell Plates, End Plates and Flat Plates Subjected to Internal Pressure

1 General

Shell plates, end plates and flat plates without stays or other supports (excluding the tube plates of heat exchangers) subjected to internal pressure are to comply with the requirements specified in **-2** to **-7**. However, the strength of the shell plates of pressure vessels is to be calculated in accordance with suitable formulae considered appropriate by the Society under the following conditions.

(1) Cylindrical pressure vessels

$$\frac{T_r}{D} > 0.25 \text{ or } P > \frac{fJ}{2.5}$$

(2) Spherical pressure vessels

$$\frac{T_r}{D} > 0.185 \text{ or } P > \frac{fJ}{1.5}$$

2 Required thickness of cylindrical shell plates subjected to internal pressure

The required thickness of cylindrical shell plates subject to internal pressure is to be calculated by the following formula. However, in the case of cylindrical shell plates having openings for which reinforcement is required, openings are to be reinforced in accordance with the requirements in **10.6.3**.

$$T_r = \frac{PR}{fJ - 0.5P} + a$$

3 Required thickness of spherical shell plates subjected to internal pressure

The required thickness of spherical shell plates subject to internal pressure is to be calculated by the following formula. However, in the case of spherical shell plates having openings for which reinforcement is required, the openings are to be reinforced in accordance with the requirements in **10.6.3**.

$$T_r = \frac{PR}{2fJ - 0.5P} + a$$

4 Required thickness of formed end plates subjected to pressure on the concave side without stays or other supports

(1) The required thickness of end plates having no openings is to be calculated by the following formula:

(a) Dished and hemispherical end plates

$$T_r = \frac{PR_1W}{2fJ - 0.5P} + a$$

where

$$W = \frac{1}{4}\left(3 + \sqrt{\frac{R_1}{r}}\right) \text{ for dished end plates}$$

W = 1 for hemispherical end plates

R_1 : Inside crown radius

It is to be less than the outside diameter of the skirt of the end plate.

r : Inside knuckle radius

It is not to be less than 6 % of the outside diameter of the skirt of the end plate or 3 times the actual thickness of the end plate, whichever is greater.

(b) Semi-ellipsoidal end plates (in cases where half of the inside minor axis of the end plate is not less than 1/4 of the inside major axis of the end plate)

$$T_r = \frac{PR}{fJ - 0.25P} + a$$

(2) The required thickness of end plates having openings is to comply with the following requirements in **(a)**, **(b)** or **(c)**:

(a) In cases where no reinforcement for openings is necessary according to the requirements in **10.6.2**, or the openings are reinforced in accordance with the requirements in **9.6.3-3** to **-5**, the required thickness is to be calculated by the formula specified in **(1)** above.

(b) In cases where an end plate has a flanged-in manhole or an access opening with a maximum diameter exceeding 150 *mm* and the flanged-in reinforcement complies with the requirement in **9.6.3-7**, the thickness is to be calculated as follows:

i) Dished or hemispherical end plates

The thickness is to be increased by not less than 15% (if the calculated value is less than 3 *mm*, the value is to be 3 *mm*) of the required thickness calculated by the formula specified in **(1)(a)**. In this case, where the inside crown radius of the end plate is smaller than 0.80 times the inside diameter of the shell, the value of the inside crown radius in the formula is to be 0.80 times the inside diameter of the shell.

In calculating the thickness of end plates having two manholes in accordance with **i)**, the distance between the two manholes is not to be less than 1/4 of the outside diameter of the end plate.

ii) Semi-ellipsoidal end plates

The requirements in **(1)(a)** are to be applied, however, in this case $R1$ is to be 0.80 times the inside diameter of shell, and W is to be 1.77.

(c) The required thickness, where the openings are not reinforced in accordance with the requirements in **(a)** or **(b)**, is to be calculated by the following formula. However, this thickness is not to be less than the value obtained by the formula given in **(1)**.

$$T_r = \frac{PD_0}{2f}K + a$$

where

D_0 : Outside diameter of the end plate (mm)

K : As shown in **Fig. D9.6**. However, this is applicable to end plates complying with the following conditions:

Hemispherical end plates:

$0.003D_0 \leq T_e \leq 0.16D_0$

Semi-ellipsoidal end plates:

$0.003D_0 \leq T_e \leq 0.08D_0$

$H \geq 0.18D_0$

Dished end plates:

$0.003D_0 \leq T_e \leq 0.08D_0$

$r \geq 0.1D_0$

$r \geq 3T_e$

$R_1 \leq D_0$

$H \geq 0.18D_0$

or $0.01D_0 \leq T_e \leq 0.03D_0$

$r \geq 0.06D_0$

$H = 0.18D_0$

or $0.02D_0 \leq T_e \leq 0.03D_0$

$r \geq 0.06D_0$

$0.18D_0 \leq H \leq 0.22D_0$

T_e : Actual thickness of the end plate (mm)

H : Depth of the end plate measured on its external surface from the plane of junction of the dished part with the cylindrical part (mm)

R_1 and r : As specified in **(1)(a)**

5 Required thickness of formed end plates subjected to pressure on their convex side

The required thickness of formed end plates subjected to pressure on their convex sides is not to be less than the thickness calculated on the assumption that their concave sides are subjected to a pressure at least 1.67 times the design pressure.

6 Required thickness of flat end plates and cover plates, etc. without stays or other supports

(1) In cases where flat end plates and cover plates without stays or other supports are welded to shell plates, the required thickness is to be calculated by the following formulae:

(a) Circular plates

$$T_r = C_1 d\sqrt{\frac{P}{f}} + a$$

(b) Non-circular plates

$$T_r = C_1 C_2 d\sqrt{\frac{P}{f}} + a$$

where

C_1 : Constant shown in **Fig. D9.11**

$C_2 = \sqrt{3.4 - 2.4\frac{d}{D'}}$ but need not be over 1.6

d : Diameter shown in **Fig. D9.11** (for circular plates), or the minimum length (for non-circular end plates) (mm)

D' : Long span of non-circular end plates or covers measured perpendicular to the short span (mm)

(2) In cases where flat cover plates without stays are bolted to the shell plate, the required thickness is to be calculated by the following formulae:

(a) In cases where full face gaskets are used

For circular plates

$$T_r = d\sqrt{\frac{C_3 P}{f}} + a$$

For non-circular plates

$$T_r = d\sqrt{\frac{C_3 C_4 P}{f}} + a$$

(b) In cases where moment due to gasket reaction is to be taken into account;

For circular plates

$$T_r = d\sqrt{\frac{C_3 P}{f} + \frac{1.78 W h_g}{f d^3}} + a$$

For non-circular plates

$$T_r = d\sqrt{\frac{C_3 C_4 P}{f} + \frac{6 V h_g}{f L d^2}} + a$$

where

C_3 : Constant determined by the bolting methods as shown in **Fig. D9.12**

C_4 = $3.4 - 2.4\dfrac{d}{D'}$ but need not be over 2.5

d : Diameter shown in **Fig. D9.12** (for circular plates, or the minimum length (for non-circular plates) (*mm*)

D' : Long span of non-circular end plates or covers measured perpendicular to the short span (*mm*)

W : Mean load (*N*) of bolt loads necessary for the watertightness and the allowable load for the bolt actually used

L : Total length of the circle passing through the bolt centres (*mm*)

h_g : Arm length of moment due to the gasket reaction shown in **Fig. D9.12** (*mm*)

7 Steam heated steam generators

For steam heated steam generators, the required thickness of flat end plates with stays or other supports, and the required dimensions of the stays are to comply with the requirements in **9.5.7**, **9.5.13** and **9.5.14**.

10.5.3 Required Thickness of Tube Plates for Heat Exchangers

The thickness of tube plates for heat exchangers without tube stays is to comply with the following requirements:

(1) Except for floating head, the required thickness of flat tube plates without tube stays for the heat exchangers and the like is to be either of the values calculated by the following formulae, whichever is greater:

$$T_r = \frac{C_5 D}{2}\sqrt{\frac{P}{f_b}} + a$$

$$T_r = \frac{PA}{\tau L} + a$$

where

f_b : Allowable bending stress of the material (*N/mm*^{2})

τ : Allowable shearing stress of the material (*N/mm*^{2})

C_5 : Factor determined by the supporting method of tube and tube plate. In cases where the tube plates are not integral with the shell, when straight tubes are used this value is to be 1.0. When *U*-tubes are used this value is to be 1.25. In cases where the tube plates are integral with the shell, the value shown in **Fig. D10.1** is to be used.

D : Diameter of outer circle of tube end plate (*mm*). In cases where the tube end plate is bolted to flange, D is the diameter of a circle passing through the positions to which gasket reaction is acted; where the tube plate is fixed to the shell, the inside diameter of the shell (corrosion allowance is to be deducted) is to be taken.

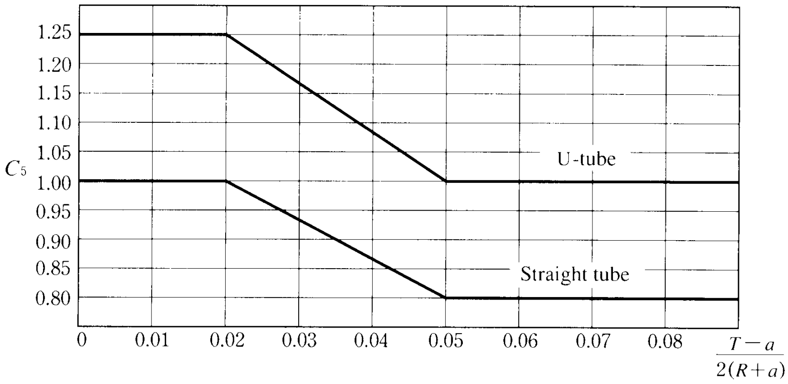
A : Area of a polygon obtained by connecting the centres of the outermost tube holes (*see*, **Fig. D10.2**) (*mm*^{2})

L : Length obtained by deducting the sum of the tube hole diameters of the outermost tubes from the length of the outer periphery of the aforementioned polygon (*mm*)

a : Corrosion allowance (mm). In cases where a groove for the partition plate or a gasket groove with a depth greater than the corrosion allowance specified in **10.4.3** is provided, a is to be taken to the depth of such groove.

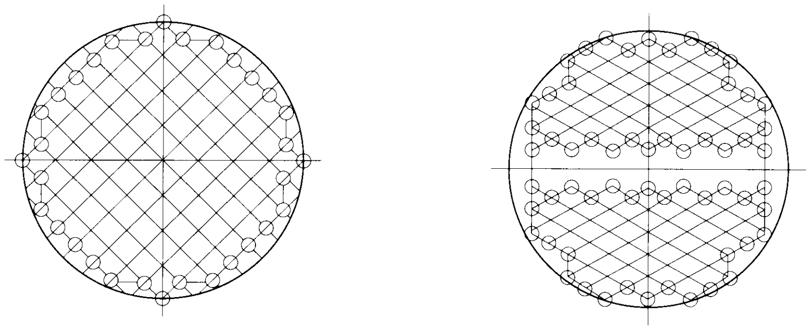
(2) The calculation of T_r in **(1)** is to be carried out on both sides by using the values for P, C_5 and D. However, in cases where a differential pressure calculation is carried out, approval will be given by the Society on a case by case basis.

Fig. D10.1 Value of C_5



T = Actual thickness of shell plate (mm)

Fig. D10.2 Polygon used for Tube Plate Calculation



(a) (b)

10.5.4 Required Thickness of Tubes for Heat Exchangers

1 The materials of the tubes for heat exchangers are to be suitable for their purposes, and the required thickness is to be calculated by the following formula:

$$T_r = \frac{PD_0}{2f} + a$$

where

D_0 : Outside diameter of the tube (mm)

a : 1.5 mm for steel tube; 0.1 T for copper or copper alloy tube

T : Actual thickness of the tube (mm)

f : As given in **10.4.1** or **Table D10.3**

2 The thickness of the bent pipes for U-tube type heat exchangers is to be sufficient and take into account any thickness reduction caused by bending.

Table D10.3 Values of the Allowable Stress of Copper and Copper Alloy Pipes (f)

Kind of materials (Grade)	Design Temperature (℃)										
	50 or less	75	100	125	150	175	200	225	250	275	300
For phosphorous deoxidized copper seamless pipes and tubes (N/mm^2)											
C1201 C1220	41	41	40	40	34	27.5	18.5	-	-	-	-
For brass seamless pipes and tubes for condensers and heat exchangers (N/mm^2)											
C4430	68	68	68	68	68	67	24	-	-	-	-
C6870 C6871 C6872	78	78	78	78	78	51	24.5	-	-	-	-
For copper nickel seamless pipes and tubes for condensers and heat exchangers (N/mm^2)											
C7060	68	68	67	65.5	64	62	59	56	52	48	44
C7100	73	72	72	71	70	70	67	65	63	60	57
C7150	81	79	77	75	73	71	69	67	65.5	64	62

Notes: Intermediate values are to be determined by interpolation.

10.5.5 Strength of Pressure Vessels Subjected to External Pressure

In cases where the internal pressure of pressure vessels may become lower than the external pressure, strength calculations are to be carried out for buckling.

10.5.6 Fatigue Analysis

For pressure vessels subjected to dynamic loads or excessive cyclic external loads, fatigue analysis is to be carried out. The degree of cumulative fatigue in these cases is to comply with the following formula. However, the value on the right side of the formula may be increased to a value considered appropriate by the Society according to the S-N curve used in the calculation, but is not to exceed 1.0.

$$\sum \frac{n_i}{N_i} \le 0.5$$

where

n_i : Number of cycles at each stress level

N_i : Number of cycles to fracture for the respective stress level given by the S-N curve of material used

10.5.7 Considerations for Secondary Stress

In cases where deemed necessary by the Society, consideration is to be given to the strength against secondary stress.

10.5.8 Considerations for Thermal Stress

For pressure vessels which may be subject to excessive thermal stress or which contain fluid with a boiling point below -55℃, consideration is to be given to the strength against thermal stress.

10.5.9 Strength Calculation by Special Method

Even in cases where the dimensions of each component of pressure vessels do not conform to the requirements in **10.5**, if detailed strength calculation sheets are submitted, the Society will examine the data and approve the pressure vessels provided that the results are acceptable to the Society.

10.6 Manholes, Other Openings for Nozzle, etc. and Their Reinforcements

10.6.1 Manholes, Cleaning Holes and Inspection Holes

1 Pressure vessels are to be provided with manholes, cleaning holes and inspection holes on the shell plates or end plates for inspection and maintenance in accordance with **Table D10.4**. However, where considered appropriate by the Society, the number and dimensions of these openings may be reduced.

2 The standard dimensions of manhole, cleaning holes and inspection holes are given in **Table D10.5**.

3 The construction of holes and covers is to comply with the requirements in **9.6.1-2**.

Table D10.4 Number of Manholes, Cleaning Holes and Inspection Holes

Inside diameter of shell	Number of manholes, cleaning holes and inspection holes	
	Vessels with internal volume of not more than 100 *l* and with internal length of not more than 1.5 *m*.	All other vessels than those listed in the left hand column
300 *mm* or below	One or more inspection holes	Two or more inspection holes
More than 300 *mm* up to and including 500 *mm*		Two or more cleaning holes; or, one or more each of cleaning holes and inspection holes
More than 500 *mm* up to and including 750 *mm*	-	One or more manholes; or, two or more cleaning holes; or, one or more each of cleaning holes (1) and inspection holes
More than 750 *mm*		One or more manholes (2)

Notes:

(1) The dimensions of cleaning holes are generally to comply with the values for cleaning holes required by the shell with an internal diameter more than 750 *mm* by the **Table D10.5**

(2) Pressure vessels such as heat exchangers, etc. which are not considered necessary to be provided with manholes for reasons of shape, purpose, etc. may be provided with two or more cleaning holes instead of any manholes.

Table D10.5 Dimensions of Holes

Type of hole	Inside diameter of the shell	Dimensions
Manholes	For all dimensions	Oval : 400 *mm*×300 *mm* Circular : 400 *mm*
Cleaning holes	More than 750 *mm*	Oval : 150 *mm*×100 *mm* Circular : 150 *mm*
	750 *mm* and less	Oval : 100 *mm*×75 *mm* Circular : 100 *mm*
Inspection holes	For all dimensions	50 *mm*

10.6.2 Reinforcement of Opening

In cases where manholes, other openings for nozzles, etc. are provided in the shell, openings are to be reinforced. However, this reinforcement may be omitted for single openings shown in the following:

(1) Openings having a maximum diameter (in a threaded opening, the diameter of the root) of not more than 60 *mm* or more than 1/4 of the inside diameter of the shell or of the flanged part of the end plate.

(2) Openings provided on the shell plate having a maximum diameter not exceeding the value shown in **Fig. D9.8**. In this case, no unreinforced opening is to exceed 200 *mm* in diameter.

(3) Openings provided on end plates complying with the requirement in **10.5.2-4(2)(c)** where no reinforcement is required due to the increased thickness of the end plates.

10.6.3 Reinforcing Procedures of Openings

The reinforcing procedures for openings provided in shell plates and end plates subjected to internal pressure are to comply with the requirements in **9.6.3**. However, the reinforcement of the following openings will be considered by the Society on a case by case basis.

(1) Openings provided in the shell plate and having a diameter not less than 1/2 of the inside diameter of the shell.

(2) Openings whose outer extremity is at a distance of one-tenth of the shell outside diameter from the outer surface of the shell.

(3) Multiple openings which are provided in close proximity of each other.

10.7 Joints and Connections of Each Member

10.7.1 Welded Joints

1 The dimension and shape of edge preparation and the method of tapering plates of unequal thickness are to comply with the requirements in **9.8.1-1** and **-2**.

2 The welded joints of the shells of the pressure vessels of Group I are to comply with the following requirements:

(1) Longitudinal joints

To be double-welded butt joints or other butt welded joints considered by the Society to be equivalent.

(2) Circumferential joints

To be in accordance with **(1)** above. However, when approved by the Society, the double-welded butt joint may be replaced by a single-welded butt joint with a backing strip or another butt welded joint considered by the Society to be equivalent may be used.

3 The welded joints of the shells of the pressure vessels of Group II are to comply with the following requirements:

(1) Longitudinal joints

To be in accordance with **-2(1)**.

(2) Circumferential joints

In addition to those in **(1)**, single-welded butt joints with backing strips or other butt welded joints considered by the Society to be equivalent. However, for plates of not more than 16 *mm* in thickness, a single-welded butt joint may be used.

4 The welded joints of the shells of the pressure vessels of Group III are to comply with the following requirements:

(1) Longitudinal joints

(a) For plates over 9 *mm* in thickness

Same as those in **-3(1)**. However, single-welded butt joints with backing strips or other butt welded joints considered by the society to be equivalent may be used.

(b) For plates of not more than 9 *mm* in thickness

Same as those in **(a)** above. However, a double-welded full fillet lap joint may be used.

(c) For plates of not more than 6 *mm* in thickness

Same as those in **(b)** above. However, a single-welded butt joint may be used.

(2) Circumferential joints

Same as those in **(1)(c).** However, a one-sided welded full fillet lap joint may be used.

10.7.2 Shape of Welded Joint and Connection

The shape of welded joints and connections are to be as shown in **Fig. D9.11**, or be a shape which is considered by the Society to be equivalent.

10.7.3 Construction of Bolted Cover Plates

The construction of unstayed flat cover plates bolted to the shell is to comply with the requirements in **9.8.3**.

10.8 Fittings, etc.

10.8.1 Materials of Fittings

The materials for nozzles, flanges or distance pieces attached directly to the shell of pressure vessels of Group I and Group II are to be equivalent to the material of the shell. However, this requirement may be dispensed with for flanges that are to be bolted or where approved by the Society.

10.8.2 Construction of Fittings

1 Fittings such as valves, flanges as well as bolts, nuts, gaskets, etc. are to be of a construction and have dimensions conforming to the recognized standards. They are also to conform to the service conditions specified in such standards.

2 Fittings are to be attached to the shells of pressure vessels of Group I and Group II by flanged joints or by welding. However, in cases where the thickness of the shell is over 12 *mm* or in cases where a seat for screwing is fitted to the shell, fittings of not more than 32 *mm* in nominal diameter may be attached to the shell by screws.

10.8.3 Installation of Pressure Relief Devices

1 Pressure vessels in which pressure may exceed the design pressure under working conditions are to be provided with relief valves. These relief valves are to be set at a pressure not exceeding the design pressure and be capable of preventing the pressure from exceeding the design pressure by more than 10%.

2 In cases where the exposure of a pressure vessel to fire or some other unexpected source of external heat may create a dangerous condition, a pressure relieving device is to be provided to prevent the pressure from exceeding the design pressure by more than 1.2 times. However, if an air reservoir which is not used for a general emergency alarm system required by the paragraph 4.2, Regulation 6, Chapter III, the Annex to *SOLAS* Convention is provided with a fusible plug that has a melting point, not exceeding 150℃, to release pressure automatically in the case of a fire, such a pressure relieving device may be omitted.

3 Heat exchangers or other similar pressure vessels, where internal pressure may exceed design pressure due to a failure of the heat exchanging tubes, tube plates, partition plates and other internals are to be provided with a suitable relief valve.

4 Steam generators belonging to Group I are to be provided with the safety valve specified in **9.9.3**.

5 No stop valve is to be provided between a pressure vessel and a relief valve or other pressure relieving devices, except where means are provided in such a way that the function of the pressure relieving device is not impaired during the use of the pressure vessels.

6 A rupture disc may be provided between a pressure vessel and a relief valve or at the discharge line of a relief valve. In this case, the bursting pressure of the rupture disc is not to exceed the set pressure of the relief valve. In addition, the discharge capacity of the rupture disc is not to be less than the discharge capacity of the relief valve.

10.8.4 Pressure and Temperature Measuring Devices

Pressure and temperature measuring devices are to be provided on pressure vessel where considered necessary.

10.8.5 Fittings of Air Reservoir

1 Pressure relieving devices for air reservoirs are to comply with the requirements in **10.8.3**.

2 Air reservoirs are to be provided with effective drainage systems.

3 Air reservoirs are to be provided with pressure measuring devices.

10.9 Tests

10.9.1 Shop Tests*

1 Tests for welds are to conform to the requirements in **Chapter 11** of this Part.

2 Pressure vessels and their fittings are to be subjected to hydrostatic tests according to the following requirements after being manufactured:

(1) Shells of pressure vessels

(a) Pressure vessels of Group I and Group II are to be subjected to hydrostatic tests at a pressure equal to 1.5 times their design pressure.
However, when the primary general membrane stress of the shell is expected to exceed 90 % of the specified yield point of the material by this test pressure, the test pressure is to be lowered to such a pressure that the stress becomes 90 % of the specified yield point of the material.

(b) Pressure vessels of Group III are to be subjected to hydrostatic tests in accordance with the requirements in **(a)** above when considered necessary by the Society.

(2) Fittings of pressure vessels
The fittings that are not only directly welded to pressure vessels of Group I and Group II are to be subjected to hydrostatic tests at a pressure equal to 2 times their design pressure.

(3) Hydrostatic tests of heat exchangers which are not specified in **(1)** and **(2)** and other special pressure vessels as well as their fittings will be considered by the Society on a case by case basis.

Chapter 11 WELDING FOR MACHINERY INSTALLATIONS

11.1 General

11.1.1 Scope

1 The requirements in this chapter apply to welding for machinery installations.

2 As for matters other than those specified in this Chapter, the requirements in **Part M** are to apply.

11.1.2 Base Metals

1 Base metals used in welding work are to be those suitable for welding. And, the carbon content is not to exceed 0.23% for carbon steel and low alloy castings and forgings, or 0.35% for other carbon steel and low alloy steel. However, in cases where the Society has, after considering the welding conditions, given its approval, the carbon content may be increased to the Society approved value.

2 The upper limit of the carbon equivalent for high tensile steels is to be as deemed appropriate by the Society.

11.2 Welding Procedure and Related Specifications

11.2.1 Approval of Welding Procedure and Related Specifications*

1 The manufacturer is to obtain the approval of the welding procedures in the following cases:

(1) Where the welding procedures are first adopted for the welding work specified below.

- (a) Welding work for windlasses
- (b) Welding work for boilers (including smoke tubes, stay tubes, superheater tubes, heat exchanging tubes of thermal oil heaters, etc.), pressure vessels of Group I and Group II (including heat exchanging tubes of heat exchangers)
- (c) Welding work for the principal components of prime movers, etc. (these principal components are specified in **Table D2.2**, **3.2.1-1**, **4.1.2(5)** and **5.2.1-1**; hereinafter, this definition applies throughout this Chapter)
- (d) Welding work for pipes belonging to Group I and Group II
- (e) Welding work using special materials
- (f) Welding work using special welding procedures

(2) Where the items described in the approved welding procedure specifications are altered.

(3) Where considered necessary by the Surveyor..

2 The specifications which correspond to the welding procedure referred to in the preceding **-1** are to be treated collectively as the welding procedure specification and to be approved by the Society. The specifications are to include the items specified in **2.2.2-2** and **-3, Part M**.

3 Whenever manufacturers conduct an approval test for a welding procedure and related specifications applied to a welding work specified in any of **-1(1)(b)** to **(f)**, they are to submit detailed data in connection with this welding work to the Society for approval.

11.2.2 Execution of Tests*

The requirements in **4.1.3 of Part M** are to be applied. However, approval tests for the following welding procedures and related specifications are to be in accordance with requirements otherwise specified.

(1) Welding procedures and related specifications applicable to the welding work for windlasses for which the approval test is carried out using a material not specified in **Part K** as a base metal

(2) Welding procedures and related specifications applicable to the welding work for boilers (excluding smoke tubes, stay tubes, superheater tubes, heat exchanging tubes of thermal oil heaters, etc.), pressure vessels of Group I and Group II (excluding heat exchanging tubes of heat exchangers) or principal components of prime movers, etc.

(3) Welding procedures and related specifications applicable to the welding work specified in **11.2.1-1(1)(e)** or **(f)**

11.2.3 Range of Approval*

The requirements in **4.1.4 of Part M** are to be correspondingly applied. However, the ranges of approval for the welding procedures and related specifications for the welding work specified in **11.2.1-1(1)(a)**, **11.2.2(2)** and **11.2.2(3)** are to be in accordance with the requirements otherwise specified.

11.3 Post Weld Heat Treatment

11.3.1 Procedure of Post Weld Heat Treatment*

1 Stress-relieving procedures of the post weld heat treatment for welds using carbon steel, carbon manganese steel and low alloy steel as the base metal are to be as follows:

(1) Furnace heating method

(a) The temperature of the furnace is to be less than 400℃ at the time the object is placed in or taken out of it.

(b) The rates of heating and cooling above 400℃ are to be as follows:

i) The heating rate $\leq$ $220 \times 25/t$ (℃/hr), but under any circumstances not more than 220 (℃/hr)

ii) The cooling rate $\leq$ $275 \times 25/t$ (℃/hr), but under any circumstances not more than 275 (℃/hr)

where

t : Maximum weld thickness (mm)

(c) The holding temperature of the furnace is to be as given in **Table D11.1.** The furnace is to be kept at this temperature for a period of one hour per 25 mm thickness of the welded part and then cooled slowly. When Society approval has been obtained, the furnace temperature may be reduced to that given in **Table D11.2**.

(d) During heating and cooling periods, temperature variation throughout the portion being heated shall not be a greater than 130℃ within any 4500 mm interval of length. During the holding period, the difference between the highest and lowest temperature of each portion being heated shall not be greater than 80℃.

(e) The maximum heating temperature for each portion of the object is to be less than 20℃ below the final temperature of the heat treatment for the material of the portion.

(2) Local heating methods

(a) In post-heating processing, the temperature gradient between the heating and non-heating areas is to be made smooth so that the materials will not suffer any harmful effects.

(b) The heating band is to be greater than such an area with a length of 6 times or more the plate thickness as measured from the centre of the weld for each side respectively. In circumferential joints, the heating band may be 3 times the plate thickness (2 times in the case of pipes) on the outer side of the welding bead at the maximum width.

(c) Heating and cooling rates at temperatures of 400℃ or above are to conform to the requirements in **(1)(b)**.

(d) The holding temperature and the period of the post weld heat treatment are to conform to the requirements in **(1)(c)**. Throughout the holding period, or the heating and cooling periods, the entire band is to be brought up to the required temperature as uniformly as possible.

2 For post weld heat treatment procedures on materials other than those specified in **-1**, the requirements shall be specially considered by the Society according to the type of base metal, the welding materials and the welding procedures.

3 Post weld heat treatments of low alloy steels, alloy steels and other special steels are to be carried out with special consideration being given to avoiding any undue degrading of notch toughness of the material and any cracks in the material that are caused by the heat treatment.

Table D11.1 Post Weld Heat Treatment Temperature

Kind of steel	Minimum holding temperature (°C)
Carbon steel Carbon manganese steel $0.3Mo$ steel $0.5Mo$ steel $0.5Cr$-$0.5Mo$ steel $1Cr$-$0.5Mo$ steel $1\frac{1}{4}Cr$-$0.5Mo$ steel	600
$2\frac{1}{4}Cr$-$1Mo$ steel $5Cr$-$0.5Mo$ steel $0.5Cr$-$0.5Mo$-$0.5Mo$ steel	680

Table D11.2 Temperature Reduction vs. Holding Time2)3)

Minimum holding temperature℃	Minimum holding time (*hour*)
T-30	2
T-60	3
T-90 1)	5

Notes:

1) Applicable to carbon steel and carbon manganese steel only.
2) Intermediate values are to be obtained by interpolation.
3) T is the minimum holding temperature in **Table D11.1**.

11.3.2 Temperature Measurements and Recordings during Post Weld Heat Treatment

In general, the temperature measurements are to be carried out automatically by a thermocouple. However, in cases where the temperature of each part of the heated object can be readily assumed on the basis of furnace temperature, such furnace temperature may be used in place of the temperature of the heated object.

When post weld heat treatments are carried out, the following items are to be recorded:

(1) Type and kind of furnace or heating equipment
(2) Holding temperature and period
(3) Heating and cooling rates
(4) Other items deemed necessary

11.4 Welding of Boilers

11.4.1 General

In cases where the pressure parts of boilers are fabricated by welding, this welding is to be carried out in accordance with the requirements in 11.4 of this Chapter.

11.4.2 Alignment of Joints and Out-of Roundness

1 For the alignment of butt welded joints, maximum offset is not to exceed the following limits:

(1) For longitudinal joints:
- 1 *mm* for plates with a thickness of 20 *mm* or less
- 5% of the plate thickness for plates with a thickness of more than 20 *mm* but less than 60 *mm*
- 3 *mm* for plates with a thickness of 60 *mm* or more

(2) For circumferential joints:

- 1.5 *mm* for plates with a thickness of 15 *mm* or less
- 10% of the plate thickness for plates with a thickness of more than 15 *mm* but less than 60 *mm*
- 6 *mm* for plates with a thickness of 60 *mm* or more

2 The difference between the maximum and minimum inside diameters (out-of roundness) at any cross section is not to exceed 1% of the nominal inside diameter at the cross section under consideration.

11.4.3 Post Weld Heat Treatment*

1 Each boiler, including all mountings and fittings, is to be subjected to post weld heat treatment for stress relieving after the completion of all welding work. However, in cases where the thickness of the welded part is less than 19 *mm* for carbon steel or less than 13 *mm* for alloy steel, the Society, after taking into account the welding procedures as well as the preheating and post weld heating conditions of these parts, may deem the omission of such post weld heat treatment to be acceptable.

(1) Welded joints between tubes, tubes and tube flanges, and tubes and headers

(2) Circumferential joints of headers

(3) Welded parts specifically approved by the Society

2 In cases where minor fillet welding is carried out for the following items **(1)** and **(2)** on boilers subjected to post weld heat treatment, no post weld heat treatment is required after such welding work.

(1) Seal welding

(2) Intermittent welding for attaching fittings provided that the welds do not exceed 6 *mm* in throat thickness and 50 *mm* in length and have an interval of 50 *mm* or more.

11.4.4 Production Weld Tests*

1 For welded joints of boiler shells, production weld tests are to be carried out. The welded joints of furnace plates may be tested by any of the following production weld tests: a guided bend test, a roller bend test, or a radiographic test.

2 Test plates for workmanship tests are to be sampled in accordance with the following requirements:

(1) The test plates are to be attached to each shell in such a manner so that they are welded continuously and correspond to the edges of the longitudinal joint.

(2) The test plates for the circumferential joints of shells are to be made separately under the same welding conditions as those for the circumferential joint. However, test plates for circumferential joints are not required except where the shell has no longitudinal joints or the welding procedure for the circumferential joints is remarkably different from those for the longitudinal joints.

(3) Test plates are to be of the same specification, type and thickness as the base metal (where plates with different thickness are welded, test plates are to be taken from the thinner one), and no warping is to be caused by welding.

(4) Test plates are to be subjected to the same post weld heat treatment as in the actual welding and are not to be heated beyond the heating temperature and holding period as applied in the actual welding.

3 Tests for the welded joints of test plates, such as a tensile test, a bend test and a macro-etching test are to be carried out. Guided bend tests or roller bend tests may be accepted as the bend test. In this case, the number and dimensions of the test specimens are to be as given in **Table D11.3**.

4 Test methods and test standards are required to comply with the following requirements:

(1) Tensile tests for joints

Tensile strength is not to be less than the minimum tensile strength specified for the base metal. However, if the test specimen breaks at the base metal but shows no sign of defect in the welded joint; and, the tensile strength is not less than 95 % of the minimum tensile strength specified for the base metal, the test results may be judged to be acceptable.

(2) Guided bend tests or roller bend tests

The test specimen is to be put on a bend test jig deemed appropriate by the Society and the center line of the welding part is to coincide with the center of the jig. For side bend tests, the test specimen is to be bent with one of its two sides in tension. For root bend tests, the test specimen is to be bent with the narrow side of the weld in tension. In all cases, the test specimens are to be bent in the jig through an angle of 180 *degrees*. Cracks or any other defects exceeding 3 *mm* in length are not to be observed on the outer surface of the bent specimen on the welding part. However, any cracks in the corners of the test specimen may be considered irrelevant to the test results.

(3) Macro-etching tests

Cracks, lack of fusion, incomplete penetration or any other defect are not to be observed.

5 In cases where the tensile strength is not less than 90 % of the values specified in the requirements, or in cases where a guided bend test or a roller bend test fails to meet the requirements from defects other than those in the welded parts, a retest will be allowed. In this case, two additional test specimens are to be taken from the same test plate for each failure and both of these two test specimens are required to satisfy the requirements.

Table D11.3 Number and Dimensions of Test Specimens

<table>
<tr><th colspan="2">Number of test specimens</th><th>Dimensions of test specimens</th></tr>
<tr><td colspan="2">Tensile test for joint : 1</td><td>As specified in Table M3.1, Part M</td></tr>
<tr><td>Guided bend tests or roller bend tests</td><td>Face bend tests and root bend tests: 1 set
or
Side bend tests: 1</td><td>As specified in Table M3.2, Part M</td></tr>
<tr><td colspan="2">Macro-etching tests: 1</td><td>—</td></tr>
</table>

Note:

For test plates not more than 20 *mm* in thickness, face bend tests and root bend tests are to be conducted. For those over 20 *mm* in thickness, side bend tests are to be conducted.

11.4.5 Non-destructive Testing for Longitudinal and Circumferential Joints*

1 For boiler shells (including headers), the entire length of both the longitudinal and the circumferential welded joints is to be subjected to radiographic testing.

2 The radiographic technique employed is to be such as to detect a defect as small as 2 % of the welding depth and the wire of the penetrameter, corresponding to 2 % of the thickness of the base metal, is to be clearly shown on the radiographic film.

3 Each radiograph film is to be clearly marked with respect to the relative position of the welds and the radiograph position.

4 The following items are to be included in the radiographic testing report:

(1) Thickness of the material (flush or reinforced)

(2) Distance from the radiation source to the weld surface

(3) Distance from the film to the weld surface

(4) Type of penetrameter used

5 Reinforcement of the welded joints, where radiographic testing is carried out, is to be evenly finished to ensure trouble free examination. In this case, the height of the reinforcement is to be in accordance with the following standards:

(1) Double-welded butt joints:

To be as given in **Table D11.4**

(2) Single-welded butt joints:

To be 1.5 *mm* or less, regardless of the plate thickness

6 Any defects found as a result of radiographic testing is to be dealt with according to the following requirements:

(1) In cases where there are defects such as crack, lack of fusion, incomplete penetration, etc., the defective part is to be chipped off and rewelded.

(2) Defects such as blow-holes and slag-inclusions are to be reconditioned in accordance with procedures deemed appropriate by the Society after taking into consideration the shape, dimensions and distribution of the defect.

7 In cases where repairs are carried out on welded joints, the repaired part of the joint is to be subjected to a radiographic testing once again.

8 Notwithstanding the requirements in **-1** to **-7** above, other appropriate non-destructive testing may be conducted in lieu of the radiographic testing using a radiograph film in cases where the Society specifically grants approval.

Table D11.4 Allowable Height of Reinforcements

Thickness of base metal (mm)	12 or less	Exceeding 12 but not more than 25	Exceeding 25
Allowable height of reinforcement (mm)	1.5	2.5	3.0

11.4.6 Non-destructive Testing for Other Welds*

1 For important welds other than those specified in **11.4.5**, non-destructive tests are to be carried out as considered appropriate.

2 Non-destructive testing procedures are to comply with the requirements specified in **11.4.5-2** to **-7** and any other non-destructive testing procedures are to be appropriate for the type of tests employed. The radiographic testing, however, may be conducted in another appropriate method in lieu of the radiographic testing using a radiograph film in cases where the Society specifically grants approval.

11.5 Welding of Pressure Vessels

11.5.1 General

In cases where the pressure parts of pressure vessels are fabricated by means of welding, the welding is to be carried out in accordance with the requirements in **11.5** of this Chapter.

11.5.2 Alignment of Joints, Out-of Roundness and Angular Deflection

1 For the alignment of the butt welded joints, the maximum offset is not to exceed the following limits:

(1) For longitudinal joints, joints in end plates and joints between hemispherical end plates and shells:

(a) 1/4 t for plates with an actual thickness of 50 mm or less (t) (maximum: 3.2 mm)

(b) 1/16 t for plates with an actual thickness of more than 50 mm (t) (maximum: 9 mm)

(2) For circumferential joints:

(a) 1/4 t for plates with an actual thickness of 40 mm or less (t) (maximum: 5 mm)

(b) 1/8 t for plates with an actual thickness of more than 40 mm (t) (maximum: 19 mm)

(3) For welding joints of spherical shells and end plates and welding joints between hemispherical end plates and shells, the values for longitudinal joints are applied.

2 The out-of-roundness of shells subjected to internal pressure is to be in accordance with the requirements in **11.4.2**.

3 Welds are to be free from any remarkable angular deflection.

4 The out-of-roundness and angular deflection of shells subjected to external pressure are to be examined in each case in consideration of buckling strength.

11.5.3 Stress Relieving*

1 Pressure vessels of Group I are to be subjected to post weld heat treatment for stress relieving after all fittings, such as flanges, nozzles and reinforcement plates, have been welded in place.

2 Pressure vessels of Group II corresponding to the following **(1)** or **(2)** are to be subjected to stress relieving heat treatment in accordance with the requirements in **-1**.

(1) The thickness of the shell plates exceeds 30 mm

(2) The thickness of the shell plate is not less than 16 mm and is greater than the value of T_n determined by the following formula:

$$T_n = \frac{D}{120} + 10$$

where

D : Inside diameter of shell (mm)

3 Notwithstanding the requirements in **-1** and **-2**, mechanical stress reliving by pressurizing for pressure vessels made of carbon steel or carbon manganese steel may be employed as an alternative to post weld heat treatment with the approval of the Society and subject to the following conditions **(1)** through **(4)**:

(1) Complicated welded pressure vessel parts such as nozzles are to be heat treated before they are welded to larger parts of the pressure vessels.

(2) The plate thickness is not to exceed the value given by the standard acceptable to the Society.

(3) A detailed stress analysis is to be made to ascertain that the maximum primary membrane stress during mechanical stress relieving closely approaches, but does not exceed, 90% of the yield stress of the material. Strain measurements during stress relief pressurization may be required by the Society for verifying the calculations.

(4) The procedure for mechanical stress relieving is to be submitted to the Society for approval in advance.

4 In cases where materials having superior notch toughness are used, stress relieving may be omitted if approved by the Society.

5 In cases where the following welding is carried out on stress relieved pressure vessels, post weld stress relieving may be omitted:

(1) For carbon steels and carbon manganese steels

(a) When fittings with inside diameter not more than 50 *mm* are fitted by fillet welding with a throat thickness of not more than 12 *mm*

(b) When non-pressured fittings are fitted by fillet welding with a throat thickness of not more than 12 *mm*

(c) Stud welded parts

(2) Welds specifically approved by the Society for other materials except those specified in **(1)**. In this case, appropriate preheating is to be carried out during the welding.

11.5.4 Production Weld Tests

1 In cases where pressure vessels of Group I are of welded construction, production weld tests specified in **11.5.4** are to be carried out.

(1) Test plates are to be sampled in accordance with the following requirements:

(a) The test plates are to be attached to each shell in such a manner so that they are welded continuously and correspond to the edges of the longitudinal joint. Furthermore, any deformation of the test plates during their manufacture is to be restricted to a minimum as far as practicable.

(b) The test plates for the circumferential joints of shells are to be made separately under the same welding conditions as those for circumferential joints. However, test plates for circumferential joints are not required except where the shell has no longitudinal joints or the welding procedure for the circumferential joints is remarkably different from those for the longitudinal joints.

(c) As a general rule, test plates are to be taken from the same materials used for manufacturing the pressure vessels.

(2) Mechanical tests for test plates such as a tensile test for joints, a bend test and a Charpy impact test are to be carried out. Guided bend tests or roller bend tests may be accepted as the bend test. In this case, the number and dimensions of the test specimens are to be as given in **Table D11.5**.

(3) Test methods and test standards are required to comply with the following requirements:

(a) Tensile tests and guided bend tests as well as roller bend test for joints are required to comply with the requirements in **11.4.4-4(1)** and **(2)**.

(b) Impact tests

Impact test specimens are to be sampled from welded joint portions so that its longitudinal axis is at a right angle to the welding line and its surface is 5 *mm* inside the surface of the plate. Notches on test specimens are to coincide with the centres of weld lines and their surfaces are to be at right angles to the plate surface. The mean value of the absorbed energy of three test specimens is not to be less than the Society approved value.

2 Production weld tests of pressure vessels of Group II of welded construction are to be conducted in accordance with the requirements in **-1**. However, the guided bend tests or roller bend tests specified in **-1(2)** may be omitted.

3 Retest

(1) In cases where a tested part fails, a retest may be conducted. For tensile and bend tests, two additional test specimens are to be taken from the same test plate or from other test plates manufactured in the same lot of the original test plate for each failure. In retests, both of the test specimens are to conform to the requirements. For impact tests, 1 set (three specimens) of additional test specimens is to be taken from the same test plate or other test plates manufactured in the same lot; and, if the mean value of the test results on a total of 6 test specimens is higher than the required mean value, the test plates are to be judged acceptable.

(2) Retests are allowed in the following cases:

(a) In cases where the results of tensile and impact tests are not less than 90 % of the values specified in the requirements.

(b) In the cases where the cause of failure in guided or roller bend tests is attributed to defects other than those in the welded parts.

Table D11.5 Number and Dimensions of Test Specimens

Number of test specimens		Dimensions of test specimens
Tensile test for joint : 1		As specified in **Table M3.1, Part M**
Guided bend tests or roller bend tests	Face bend tests and root bend tests : 1 set or Side bend tests : 1	As specified in **Table M3.2, Part M**
Charpy impact tests : 1 set		U4 type test specimens as specified in **2.2.4, Part K**

Note:

For test plates not more than 20 *mm* in thickness, face bend tests and root bend tests are to be conducted. For those over 20 *mm* in thickness, side bend tests are to be conducted.

4 Reduction of test

Surveyors may modify and reduce the degree of a production weld tests for pressure vessels after taking into account past results.

11.5.5 Non-destructive Testing for Welded Joints*

1 The entire length of butt weld joints corresponding to the following **(1)** or **(2)** are to be subjected full radiographic testing.

(1) Longitudinal and circumferential weld joints of pressure vessels of Group I

(2) Weld joints whose joint efficiency has been determined by full radiographic testing.

2 For the pressure vessels whose joint efficiency has been determined by spot testing, radiographic testing is to be carried out in accordance with the following requirements.

(1) For welds that were welded by the same method and by the same welder, a length which is not less than 20% (minimum 300 *mm*) of the length of the longitudinal joint as well as the weld at the intersecting section of any circumferential joints with a longitudinal joint are to be spot radiographed.

(2) Locations to be spot radiographed are to be chosen by the Surveyor.

3 Radiographic testing procedures and disposal of test results are to conform to the requirements in **11.4.5-2** to **-7**. The radiographic testing, however, may be conducted in another appropriate method in lieu of the radiographic testing using a radiograph film in cases where the Society specifically grants approval.

4 Notwithstanding the requirements specified in **-1** and **-2**, ultrasonic testing may be conducted in lieu of the radiographic testing in cases where the Society specifically grants approval.

11.5.6 Non-destructive Testing for Other Welded Parts*

1 The welds for fittings such as the openings and their reinforcements for the pressure vessels requiring full radiographic testing are to be subjected to radiographic testing or magnetic particle testing considered appropriate by the Society. However, in cases where the application of these testing methods is considered impractical or where, in consideration of the welding position and welding shape, the Society approval has been received, radiographic testing may be replaced with liquid penetrant testing, ultrasonic testing or other appropriate testing.

2 Welds at the fitted parts of fittings such as the openings and their reinforcements of the pressure vessels requiring radiographic spot testing are to be subjected to the non-destructive testing specified in **-1** according to the sampling method.

3 The requirements in **11.5.5** apply mutatis mutandis to non-destructive testing procedures and the disposal of defects, etc..

11.6 Welding of Piping

11.6.1 General

1 The requirements in **11.6** apply to the welding of pipes.

2 The requirements in **11.6.2**, **11.6.3** and **11.6.4** apply to the welding of pipes belonging to Group I and II specified in **Chapter 12** and valves and pipe fittings used for these pipes.

11.6.2 Assembling, etc. for Joints*

1 Edge preparation is to be in accordance with recognized standards and/or approved drawings. The preparation of the edges is to be preferably carried out by mechanical means. When flame cutting is used, care is to be taken to remove any oxide scales and notches due to irregular cutting by matching, grinding or chipping back to sound metal.

2 Joint preparations are to be appropriate to the welding process.

3 The maximum offset of joints between pipes is to be appropriate for the welding process and is not to exceed the maximum offset specified in **Table D11.6**.

4 Assembling for welding is to be appropriate and within prescribed tolerances. Tack welds are to be made with an electrode suitable for the base metal. Tack welds which form part of the finished weld are to be made using approved procedures. When welding materials require preheating, the same preheating is to be applied during tack welding.

Table D11.6 Maximum offset of joints between pipes

<table>
<tr><td colspan="2" rowspan="2"></td><td colspan="3">Diameter (inside diameter) and thickness of pipes during welding</td></tr>
<tr><td>Inside diameter less than 150 mm, and thickness up to 6 mm</td><td>Inside diameter less than 300 mm, and thickness up to 9.5 mm (excluding the left column)</td><td>Inside diameter 300 mm and over, or thickness over 9.5 mm</td></tr>
<tr><td rowspan="2">Maximum offset</td><td>Without backing ring</td><td>1.0 mm or 1/4 of thickness of pipe, whichever is less.</td><td>1.5 mm or 1/4 of thickness of pipe, whichever is less.</td><td>2.0 mm or 1/4 of thickness of pipe, whichever is less.</td></tr>
<tr><td>With backing ring</td><td colspan="3">0.5 mm</td></tr>
</table>

11.6.3 Preheating of Welds*

1 When welding pipes, dryness is to be ensured in all cases using suitable preheating if necessary.

2 Materials are to be preheated to the minimum preheating temperature specified in **Table D11.7** in accordance with the kind of material and thickness of the welds. However, consideration is to be given to using a higher preheating temperature when a low hydrogen process is not used.

3 The preheating of materials not specified in **Table D11.7** is to be as deemed appropriate by the Society in accordance with the kind of material, welding consumable used and welding method.

Table D11.7 Minimum Preheating Temperature

Kind of material	Thickness of weld (1) (t) (mm)	Minimum preheating temperature (°C)
$C + \frac{Mn}{6} \leq 0.4$ (2)	$t \geq 20$ (7)	50
$C + \frac{Mn}{6} > 0.4$ (2)	$t \geq 20$ (7)	100
0.3Mo steel 0.5Mo steel (3)	$t \geq 13$ (7)	100
1Cr-0.5M_0 steel (4) 1.25Cr-0.5M_0-0.75Si steel (5)	$t < 13$	100
	$t \geq 13$	150
2.25Cr-1M_0 steel (6) (8) 0.5Cr-0.5M_0-0.25V steel (8)	$t < 13$	150
	$t \geq 13$	200

Notes:

1. Excludes the thickness of any excess weld metal.
2. Corresponds to Grade1, Grade 2 and Grade 3 specified in **4.2, Part K**.
3. Corresponds to Grade 4, No.12 specified in **4.2, Part K**.
4. Corresponds to Grade 4, No.22 specified in **4.2, Part K**.
5. Corresponds to Grade 4, No.23 specified in **4.2, Part K**.
6. Corresponds to Grade 4, No.24 specified in **4.2, Part K**.
7. For welding at ambient temperatures below 0 °C, the minimum preheating temperature required is to be independent of thickness unless specially approved by the Society.
8. For these materials, preheating may be omitted for thicknesses up to 6 mm if the results of hardness tests carried out on welding procedure qualifications are considered acceptable by the Society.

11.6.4 Post Weld Heat Treatment

1 After any welding, pipes of a thickness specified in **Table D11.8** are to be subject to stress relieving heat treatment in accordance with the kind of the material. The heat treatments are not to impair the specified properties of the materials, and verifications may be required to this effect as necessary.

2 Stress relieving heat treatment after welding for other than the oxy-acetylene welding process is required as specified in **Table D11.8** in accordance with the kind of material and thickness. In cases where oxy-acetylene welding is applied, the heat treatment as specified in **Table D11.9** in accordance with the kind of material is required.

3 The heat treatment temperature is to be 20 °C lower than the temperature of the final tempering treatment of the material or below.

4 Regarding the post weld heat treatment of pipes and piping systems that are made of materials other than those given in **Table D11.8**, treatment is to be made in accordance with the kind of the base metals, the weld materials, the welding procedure, etc. as deemed appropriate by the Society.

Table D11.8 Stress Relieving Heat Treatment

Kind of material	Maximum thickness of weld [1] (t) (mm)	Minimum holding temperature (°C)
Carbon steel Carbon manganese steel[2]	$t \geq 15$[7][8]	600
0.3Mo steel 0.5Mo steel[3]	$t \geq 15$[7]	
1Cr-0.5Mo steel[4] 1.25Cr-0.5Mo-0.75Si steel[5]	$t > 8$	
2.25Cr-1Mo steel[6] 0.5Cr-0.5Mo-0.25V steel	Any[9]	680

Notes:

1. Excludes the thickness of any excess weld metal.
2. Corresponds to Grade 1, Grade 2 and Grade 3 specified in **4.2, Part K**.
3. Corresponds to Grade 4, No.12 specified in **4.2, Part K**.
4. Corresponds to Grade 4, No.22 specified in **4.2, Part K**.
5. Corresponds to Grade 4, No.23 specified in **4.2, Part K**.
6. Corresponds to Grade 4, No.24 specified in **4.2, Part K**.
7. When steels with specified Charpy V notch impact properties at low temperature are used, the thickness above which post weld heat treatment is to be applied may be increased by special agreement with the Society.
8. Stress relieving heat treatment may be omitted up to 30 *mm* thickness by special agreement with the Society.
9. Heat treatment may be omitted for pipes having thicknesses not greater than 8 *mm*, diameters not greater than 100 *mm* and minimum service temperatures of 450 °C.

Table D11.9 Heat Treatment

Kind of material	Type and temperature of heat treatment (°C)
Carbon steel Carbon manganese steel[1]	Normalizing: from 880 to 940
0.3Mo steel 0.5Mo steel[2]	Normalizing: from 900 to 940
1Cr-0.5Mo steel[3] 1.25Cr-0.5Mo-0.75Si steel[4]	Normalizing: from 900 to 960 Tempering: from 640 to 720
2.25Cr-1Mo steel[5]	Normalizing: from 900 to 960 Tempering: from 650 to 780
0.5Cr-0.5Mo-0.25V steel	Normalizing: from 930 to 980 Tempering: from 670 to 720

Notes:

1. Corresponds to Grade 2, No.4 and Grade 3, No.4 specified in **4.2, Part K**.
2. Corresponds to Grade 4, No.12 specified in **4.2, Part K**.
3. Corresponds to Grade 4, No.22 specified in **4.2, Part K**.
4. Corresponds to Grade 4, No.23 specified in **4.2, Part K**.
5. Corresponds to Grade 4, No.24 specified in **4.2, Part K**.

11.6.5 Non-destructive Testing*

1 In general, the welded joints including the inside wherever possible are to be visually examined and the following **(1)** to **(3)** non-destructive tests are required depending on the class of pipes and type of joint.

(1) The following **(a)** to **(d)** apply to butt-welded joints.

(a) Butt-weld joints of Group I pipes whose nominal diameters are greater than 65 *A* are to be subjected to full radiographic testing.

(b) Butt-weld joints of Group II pipes whose nominal diameters are greater than are greater than 90 *A* are be subjected to at least 10 % random radiographic testing.

(c) The Society may additionally require butt-weld joints other than those specified in **(a)** and **(b)** to be subjected to random radiographic testing, depending on the kind of materials, welding procedure and controls during the fabrication.

(d) Subject to approval by the Society, an ultrasonic testing procedure may be accepted, in lieu of the radiographic testing specified in **(a)** to **(c)** above when the conditions are such that a comparable level of weld quality is assured.

(2) Fillet welds of flange pipe connections in the case of Group I pipes are to be examined by the magnetic particle method or by other appropriate non-destructive methods. In other cases (e.g. fillet welds of flange pipe connections in the case of Group II or Group III pipes and sleeve welded joints), the Surveyor, in consideration of the material, dimensions and service conditions of the pipes, etc., may require a magnetic particle examination or equivalent non-destructive testing.

(3) The Society, in consideration of the material, dimensions and service conditions of the pipes, etc., may require, additional ultrasonic examinations, in addition to the non-destructive tests specified in **(1)** and **(2)** above.

2 Radiographic and ultrasonic examination is to be performed with an appropriate technique by trained operators.

3 Magnetic particle examinations are to be performed with suitable equipment and procedures, and with a magnetic flux output sufficient for defect detection. The Society may require the equipment to be checked against standard samples.

4 The requirements in **11.4.5-2** to **-7** are to be applied to radiographic testing. The radiographic testing, however, may be conducted in another appropriate method in lieu of the radiographic testing using a radiograph film in cases where the Society specifically grants approval.

11.7 Welding of Principal Components of Prime Movers, etc.

11.7.1 General

1 Welding for the principal components of prime movers, etc. is to comply with the requirements in **11.7**.

2 In cases where the principal components of prime movers, etc. are of welded construction, approval is to be obtained from the Society for the shape and dimensions of the welded parts, welding materials, welding procedures, heat treatments and non-destructive testing requirements.

11.7.2 Alignments of Joints and Edge Preparations

1 Alignments in butt welded joints are to be in accordance with the following requirements:

(1) A maximum of 5 *mm* or 1/4 of the thickness for welded parts with a thickness of 40 *mm* or less

(2) A maximum of 19 *mm* or 1/8 of the thickness for welded parts with a thickness of more than 40 *mm*

2 In butt weldings between plates of different thickness, the end of the thicker plate is to be smoothly tapered down to that of the thinner plate.

3 Butt weldings and T-joint weldings of important strength members are to be subjected to back chipping or effectively controlled so as to avoid any defects at the roots of the welds.

4 In cases where fillet welding is carried out in areas subjected to bending stress, toe parts are to have a smooth finished.

5 Welding is to be carried out in such a way as to not cause any excessive distortion at the welds.

11.7.3 Preheating of Welds

1 Preheating is to be carried out on the welds in the case of welding of thick plates, steels or low alloy steels with a carbon content exceeding 0.23 %, or alloy steels where deemed necessary by the Society.

2 The preheating method and the minimum preheating temperature are to be determined as considered appropriate by the Society according to the types of base metals and welding materials as well as the thickness of weld and the welding method.

11.7.4 Post Weld Heat Treatment

In cases where thick materials are used or restraint conditions are severe, etc., post weld heat treatments are to be carried out where it is recognized that a considerable degree of post welding residual stress with a detrimental effect on the strength of structure is expected.

11.7.5 Non-destructive Testing

For examining welds, the Society, taking into consideration the materials used, dimensions and service conditions, may require ultrasonic tests, magnetic particle tests, liquid penetrant tests and other non-destructive tests as deemed appropriate.

Chapter 12 PIPES, VALVES, PIPE FITTINGS AND AUXILIARIES

12.1 General

12.1.1 Scope

1 The requirements in this Chapter apply to the design, fabrication and testing of pipes, valves, pipe fittings and auxiliaries.

2 The following piping systems are also to comply with relevant requirements in other Parts of the Rules as specified below.

(1) Chemical cargo piping systems of ships subject to **Part S** and shipboard hydrocarbon/chemical process piping system

(2) Gas cargo/fuel and process piping systems of ships, subject to **Part N** and gas fuel piping systems of ships subject to **Part GF**.

(3) Piping systems for low flashpoint fuels defined in **2.2.1-28, Part GF** but which do not fall under **(2)** above.

12.1.2 Terminology

1 Design Pressure

Design pressure is defined as the maximum working pressure of a medium inside pipes. However, it is not to be less than any of the following pressures given in **(1)** to **(4)**:

(1) For piping systems fitted with relief valves or other overpressure protective devices, the pressure based on the set pressure of the relief valve or overpressure protective device. However, for steam piping systems connected to boilers or piping systems fitted to pressure vessels, the design pressure of the boiler shell (nominal pressure if the boiler has a superheater) or the design pressure for the shells of pressure vessels.

(2) For piping on the discharge side of pumps, the pressure based on the delivery pressure of the pump when the valve on the discharge side is closed and the pump is running at rated speed. However, for pumps having relief valves or overpressure protective devices, the pressure based on the set pressure of the relief valves or the set pressure of the over pressure protective devices.

(3) For the blow-off piping of boilers, the design pressure is specified in **9.9.6-3**.

(4) For pipes, valves and fittings containing fuel oil, the maximum working pressure or 0.3 *MPa*, whichever is greater. However, for those containing fuel oil with a working temperature above 60℃ and a working pressure above 0.7 *MPa*, the maximum working pressure or 1.4 *MPa*, whichever is greater.

2 Design Temperature

Design temperature is the highest working temperature of a medium inside pipes at a designed condition.

3 Pipe Fittings

Pipe fittings in this Part are those pipes connecting fittings such as pipe flanges, mechanical joints, pipe pieces, expansion joints, flexible hose assemblies, etc. and any items provided in piping systems such as strainers and separators.

4 Flexible Hose Assemblies

Flexible hose assemblies are short length metallic or non-metallic hoses that are normally flexible with end fittings. Flexible hose assemblies for essential services or containing either flammable or toxic media are not to exceed 1.5 *m* in length.

12.1.3 Classes of Pipes

1 Pipes are classified according to the type of medium, design pressure and design temperature as shown in **Table D12.1**. However, pipes having open ends (such as drain pipes, overflow pipes, exhaust gas pipes, exhaust pipes of safety valves) and steam relief pipes are classified into Group III regardless of their respective design temperature.

2 Piping systems for media other than those specified in **-1** will be classified by the Society after consideration has been given to the nature of the medium and the service conditions of the pipes.

Table D12.1 Classes of Pipes

Type of Medium	Design Pressure (P) and Design Temperature (T)		
	Group I	Group II [(1)]	Group III
Steam	P>1.6 MPa or T>300 °C	P≦1.6 MPa and T≦300 °C	P≦0.7 MPa and T≦170 °C
Thermal oil	P>1.6 MPa or T>300 °C	P≦1.6 MPa and T≦300 °C	P≦0.7 MPa and T≦150 °C
Fuel oil, lubricating oil and flammable hydraulic oil	P>1.6 MPa or T>150 °C	P≦1.6 MPa and T≦150 °C	P≦0.7 MPa and T≦60 °C
Air, carbon dioxide gas, water, non-flammable hydraulic oil and urea for selective catalytic reduction (SCR) systems[(2)]	P>4.0 MPa or T>300 °C	P≦4.0 MPa and T≦300 °C	P≦1.6 MPa and T≦200 °C

Note:

(1) Excluding any pipes meeting the conditions for Group III

(2) When piping materials are selected according to ISO 18611-3:2014 for urea in SCR systems.

12.1.4 Materials

1 Materials used for auxiliary machinery are to be adequate for their service conditions. Materials used for any essential parts of auxiliary machinery are to comply with recognized standards.

2 Materials for pipes are to be adequate for their service conditions and are to comply with the following requirements:

(1) Materials for pipes belonging to Group I or II are to comply with the requirements in **Part K**.

(2) Materials for pipes belonging to Group III are to comply with recognized standards.

3 Materials for valves or cocks (hereinafter referred to as "valves" in this Chapter) and pipe fittings are to be adequate for their service conditions and are to comply with the following requirements:

(1) Materials for valves and pipe fittings used for the pipes belonging to Group I or II as well as valves and pipe fittings directly fitted to the shell plating and collision bulkhead are to comply with the requirements in **Part K**. However, other materials that comply with recognized standards may be used where approved by the Society after consideration has been given to the dimensions and service conditions of the valves and piping fittings.

(2) Materials for valves and pipe fittings used for the pipes belonging to Group III are to comply with recognized standards.

4 Pipes, valves and pipe fittings for fire fighting systems are to be of corrosion resistance materials or to be protected effectively in order to prevent the fire fighting capability of the system from deteriorating due to inside corrosion.

12.1.5 Service Limitations for Materials*

1 Pipes are, as a rule, to be made of steel, copper, copper alloy or cast iron and the material is to meet the requirements for the service limitations listed below according to design temperature, classification, service, etc., unless otherwise specified. However, for pipes which have an opening and are classed in Group III regardless of design temperature, these service limitations regarding temperature do not apply.

(1) Steel pipes are not to be used for any of the following pipes:

(a) Any pipes specified as Grade 1 and Grade 2 in **4.2, Part K that have** a design temperature over 350°C. However, steel pipes may be used up to 400°C if an allowable stress value is guaranteed.

(b) Any pipes specified as Grade 3, No.2 and No.3 in **4.2, Part K that have** a design temperature over 450°C.

(c) Any pipes specified as Grade 3, No.4 in **4.2, Part K that have** a design temperature over 425°C.

(d) Any pipes specified as Grade 4, No.12 in **4.2, Part K** that have a design temperature over 500°C.

(e) Any pipes specified as Grade 4, No.22, No.23 and No.24 in **4.2, Part K** that have a design temperature over 550°C.

(f) Any carbon steel pipes for ordinary piping specified in **4.2, Part K** that are pipes of Group I, pipes with a design pressure over 1.0 MPa or pipes with a design temperature over 230°C.

(g) All other steel pipes as to be deemed appropriate by the Society.

(2) Copper pipes and copper alloy pipes are not to be used for any of the following pipes:

(a) Any pipes used for phosphorous-deoxidized-copper seamless pipes and brass seamless pipes and tubes for condensers that have a design temperature over 200°C.

(b) Any pipes used for cupro-nickel seamless pipes and tubes for condensers that have a design temperature exceeding 300°C.

(c) Section of pipes which penetrate partitions of Class *A* or Class *B* for copper alloy pipes except in cases where the Society has given special approval.

(d) All service limitations regarding temperature for other copper pipes and copper alloy pipes are to be as deemed appropriate by the Society.

(3) Cast iron pipes are not to be used for the following pipes:

(a) Pipes of Group I and Group II for cast iron pipes that have an elongation that is less than 12%.

(b) Pipes of Group I for cast iron pipes that have an elongation of 12% and over.

(c) Pipes which are susceptible to water hammering as well as pipes subject to large deflection or vibrations.

(4) In addition to **(2)** and **(3)**, copper pipes, copper alloy pipes and cast iron pipes are to conform to the requirements in **Table D12.2** according to their application. However, the requirements may be waived when deemed acceptable by the Society.

Table D12.2 Service Limitations for Pipes according to Application

Pipe Application (Note 1)	Material		
	Copper	Copper alloy	Cast iron
Fuel oil pipes Lubricating oil pipes in machinery spaces Hydraulic oil pipes in machinery spaces Thermal oil pipes in machinery spaces Cargo oil pipes Air pipes Sounding pipes outside of sounding areas	× (Note 2)	× (Note 2)	× (Note 3)
Overflow pipes Bilge pipes Ballast pipes Drain pipes opening outboard and sanitary pipes Pipes below the freeboard deck Pipes used for fire fighting aboard ship Pipes in danger of rupturing leading to flooding during a fire Boiler water blow off pipes	×	×	×
Control oil pipes in machinery spaces	○	× (Note 2)	×
Air pipes for the remote closing of tank suction stop valves Air pipes for the remote control of auxiliaries, valves, etc. used during a fire	○	×	×

Notes:

1. Pipes used for measurements, drain pipes and vent pipes are not included.
2. The portion of pipes which is inside a tank is usable.
3. Including those outside machinery spaces.

Remarks:

1○ : Usable

2× : Use prohibited

2 Valves and pipe fittings are, as a rule, to be made of steel, copper alloy or cast iron and, except where otherwise specified, they are to conform to the requirements below for service limitations according to their design temperature, class, application, etc. However, for valves and pipe fittings which have an opening and are classified as Group III notwithstanding their design temperature, the service limitations regarding temperature do not apply.

(1) Cast steel products and forged steel products are not to be used for the following valves and pipe fittings:

(a) Valves and pipe fittings with a design temperature over 425°C for cast carbon steel products and forged carbon steel products specified in **5.1** and **6.1, Part K**.

(b) Valves and pipe fittings with a design temperature over 550°C for cast low alloy steel products and forged low alloy products specified in **5.1** and **6.1, Part K**.

(c) Other cast steel products and forged steel products when deemed appropriate by the Society.

(2) Valves and pipe fittings made of copper alloy are not to be used for valves and pipe fittings with a design temperature over 210°C. However, special bronze, when approved by the Society, can be used for valves and pipe fittings with a design temperature of 260°C or less.

(3) Cast iron products with an elongation less than 12% are not to be used for the following valves and pipe fittings:

(a) Valves and pipe fittings with a design temperature over 220°C.

(b) Valves and pipe fittings used for pipes of Group I and Group II (except steam pipes), except where deemed appropriate by the Society after consideration has been given to their construction and purpose.

(c) Valves fitted on the external walls of fuel oil tanks or lubrication oil tanks that are subjected to the static head of internal fluid.

(d) Valves, seats and distance pieces mounted on shell plating or sea chests.

(e) Valves directly mounted onto collision bulkheads.

(f) Valves and pipe fittings of boiler water blow-off piping systems.

(g) Piping systems which are liable to receive water hammering as well as valves and pipe fittings of piping systems which are subject to large deflection or vibrations.

(h) Valves and pipe fittings of clean ballast piping systems which penetrate cargo oil tanks and reach the forepeak tank.

(i) Valves and pipe fittings of cargo oil piping systems with a design pressure over 1.6 *MPa*.

(j) Valves provided at the ship/shore connection of a flammable liquid cargo line.

(4) Cast iron products with an elongation of 12% or above are not to be used for valves and pipe fittings for pipes of Group I, except where deemed appropriate by the Society after consideration has been given to their construction and purpose.

12.1.6 Use of Special Materials*

Notwithstanding the provisions in **12.1.5** above, special materials such as rubber hoses, plastic pipes (including vinyl pipes) complying with **Annex 12.1.6**, aluminum alloys, etc. may be used in cases where approved by the Society in accordance with requirements specified otherwise after taking into account their safety against fire and flooding as well as their service conditions.

12.2 Thickness of Pipes

12.2.1 Required Thickness of Pipes Subject to Internal Pressure

1 The required thickness of pipes subject to internal pressure is to be determined by the following formula:

$$t_r = t_0 + b + C$$

where

t_r : Required thickness of pipe (*mm*)

$$t_0 = \frac{PD}{2fJ + P}$$

P : Design pressure (*MPa*)

D : External diameter of the pipe (*mm*)

f : Allowable stress specified in **-3** (N/mm^2)

J : Joint efficiency as given in the following:

Seamless pipes 1.00

Electric resistance welded pipes 0.85

(However, a value of 1.00 may be adopted in cases where an ultrasonic flaw test or an alternative flaw test, considered appropriate by the Society, is conducted for the entire length of the welded joint)

b : Allowance for bending as given in the following formula:

$$b = \frac{1}{2.5}\frac{D}{R}t_0$$

R : Mean radius of the bend (*mm*)

However, b need not be considered when it has been ascertained that the calculated membrane stress in the bend does not exceed the allowable stress.

C : Corrosion allowance specified in **-5** (*mm*)

2 The thickness of pipes having a negative tolerance in thickness is not to be less than value t_1 determined by the following formula:

$$t_1 = \frac{t_r}{1 - \frac{a}{100}}$$

where

t_r : Same as in **-1**.

a : Maximum negative tolerance (%)

3 The allowable stress of each material is to comply with the following requirements:

(1) The allowable stress (f) of carbon steel pipes and low alloy steel pipes is to be chosen as the lowest of the values given by the following formulae, or the value shown in **Table D12.3(1)**.

However, where the design temperature is not in the creep region of the material, the value of f_3 need not be considered.

$$f_1 = \frac{R_{20}}{2.7}, f_2 = \frac{E_t}{1.6}, f_3 = \frac{S_R}{1.6}$$

where

R_{20} : Minimum tensile strength of the material at room temperature (N/mm^2)

E_t : Yielding point or 0.2% proof stress of the material at design temperature (N/mm^2)

S_R : Average stress for material concerned to produce rupture after 100,000 *hours* at design temperature (N/mm^2)

(2) The allowable stress of copper pipes, brass pipes and copper nickel pipes is to be the value shown in **Table D12.3(2)**

(3) The allowable stress of material other than those specified in **(1)** and **(2)** will be considered by the Society in each case.

4 For the steel pipes with a design temperature that does not exceed 250°C, in cases where the value for t_0 specified in **-1** is calculated by using an allowable stress to the value of 1/5 of the specified minimum tensile strength of the material at room temperature instead of using the value for allowable stress specified in **-3(1)**, the value for b required to be considered in the formula of t_r specified in **-1** and the increment for the negative tolerance required by **-2** need not be taken into consideration.

5 The corrosion allowance for steel pipes as well as copper and copper alloy pipes is to comply with **Table D12.4** and **Table D12.5** respectively.

Table D12.3(1) Values of Allowable Stress of Steel Pipes (f)

Material \ Design Temperature		Allowable stress of steel pipes (f) N/mm^2													
		100 or less	150	200	250	300	350	375	400	425	450	475	500	525	550
Grade 1	No.2	123	114	105	96	87	78	-	-	-	-	-	-	-	-
	No.3	138	128	118	107	96	90	-	-	-	-	-	-	-	-
Grade 2	No.2	123	114	105	96	87	78	-	-	-	-	-	-	-	-
	No.3	138	128	118	107	96	90	-	-	-	-	-	-	-	-
	No.4	156	145	133	122	117	113	-	-	-	-	-	-	-	-
Grade 3	No.2	123	114	105	96	87	78	75	70	63	56	-	-	-	-
	No.3	138	128	118	107	96	90	87	84	71	57	-	-	-	-
	No.4	156	145	133	122	117	113	105	96	77	-	-	-	-	-
Grade 4	No.12	119	112	105	97	89	85	83	80	77	73	70	65	-	-
	No.22	121	116	111	105	99	93	91	89	85	80	76	71	55	38
	No.23	121	116	111	105	99	93	91	89	85	80	76	71	56	40
	No.24	121	116	111	105	99	93	91	89	85	80	76	71	56	41

Notes:

1. Intermediate values are to be determined by interpolation.
2. The materials of steel pipes shown in this Table are to comply with the requirements in Part K.

Table D12.3(2) Values of Allowable Stress of Copper and Copper Alloy Pipes (*f*)

Kind of materials (Grade)	Design Temperature (Material°C)										
	50 or less	75	100	125	150	175	200	225	250	275	300
For phosphorous deoxidized copper seamless pipes and tubes (N/mm^2)											
*C*1201 *C*1220	41	41	40	40	34	27.5	18.5	-	-	-	-
For brass seamless pipes and tubes for condensers and heat exchangers (N/mm^2)											
*C*4430	68	68	68	68	68	67	24	-	-	-	-
*C*6870 *C*6871 *C*6872	78	78	78	78	78	51	24.5	-	-	-	-
For copper nickel seamless pipes and tubes for condensers and heat exchangers (N/mm^2)											
*C*7060	68	68	67	65.5	64	62	59	56	52	48	44
*C*7100	73	72	72	71	70	70	67	65	63	60	57
*C*7150	81	79	77	75	73	71	69	67	65.5	64	62

Notes: Intermediate values are to be determined by interpolation.

Table D12.4 Corrosion Allowance for Steel Pipes(*C*)

Piping service		*C* (*mm*)
Superheated steam systems		0.3
Saturated steam systems	General service	0.8
	Steam coil systems in cargo oil tanks	2
	Steam coil systems in fuel oil tanks	1
Feed water systems for boilers	Open circuit systems	1.5
	Closed circuit systems	0.5
Blow-off systems for boilers		1.5
Compressed air systems		1
Lubricating and hydraulic oil systems		0.3
Fuel oil systems		1
Cargo oil systems		2
Primary refrigerant systems for refrigerating plants		0.3
Fresh water systems		0.8
Sea water systems		3

Notes:

1. For pipes efficiently protected against the internal corrosion, the corrosion allowance in this Table may be reduced by 50% where approved by the Society.
2. In cases where special alloy steels with sufficient corrosion resistance are used, the corrosion allowance may be reduced to zero.
3. For sea water steel pipes whose nominal diameter is 25 *A* or below, the corrosion allowance may be reduced to 1.5 *mm*.
4. Where it is difficult to apply this Table or where a medium not specified in this Table is used, the corrosion allowance will be considered by the Society in case taking into account the corrosion conditions.

5. In cases where pipes pass through tanks, consideration is to be given to any external corrosion; and, depending on the type of external medium, a corrosion allowance is to be added according to the figures given in this Table.

Table D12.5 Corrosion Allowance for Copper and Copper Alloy Pipes (C)

Kind of material	C (mm)
Phosphorous-deoxidized copper seamless pipes and brass seamless pipes specified in **Table D12.3(2)**	0.8
Copper nickel seamless pipes specified in **Table D12.3(2)**	0.5

Note:

For media without corrosive action in respect of the material employed, the corrosion allowance may be reduced to zero.

12.2.2 Minimum Thickness of Pipes*

1 The thickness of steel pipes is to comply with the requirements in **12.2.1** and is not to be less than the value shown in **Table D12.6** depending on the service and location of the pipes. However, where corrosion resistant alloy steel pipes are used in lieu of steel pipes, the minimum thickness of these pipes will be considered by the Society in each case.

2 For pipes efficiently protected against corrosion, the minimum thickness specified in **Table D12.6(2)** may be reduced by an amount up to but not more than 1 *mm* except for steel pipes for CO_2 fire extinguishing.

3 In determining the thickness of pipes from **Table D12.6(2)**, no allowance need be made for any negative tolerance and reduction in thickness due to bending. However, for threaded pipes their minimum thickness is to be measured at the bottom of the thread, with the exception of the threaded portions for fitting the pipe head of air pipes, overflow pipes and sounding pipes as well as the threaded portions of pipes used for CO_2 fire extinguishing from the distribution station to the nozzles.

4 The minimum thickness of copper and copper alloy pipes is to be as shown in **Table D12.7**.

5 Minimum wall thickness of pipes for which mechanical joints are used is also to comply with the requirements in **12.3.3-3**, in addition to **-1** to **-4**.

Table D12.6(1) Minimum Thickness of Steel Pipes

<table>
<tr><th>Services of pipes</th><th colspan="2">Location of pipes</th><th>Minimum thickness of the encircled alphabets correspond to those in Table D12.6.(2)</th></tr>
<tr><td rowspan="3">Bilge pipes</td><td colspan="2">Passing through tanks except for cargo oil tanks</td><td>Ⓔ</td></tr>
<tr><td colspan="2">Passing through cargo oil tanks</td><td>16 mm</td></tr>
<tr><td colspan="2">Not passing through tanks</td><td>Ⓗ</td></tr>
<tr><td rowspan="5">Ballast pipes</td><td colspan="2">Passing through tanks except for cargo oil tanks (Note 2)</td><td>Ⓔ</td></tr>
<tr><td rowspan="3">Passing through cargo oil tanks</td><td>For outboard discharge</td><td>16 mm</td></tr>
<tr><td>For the ballast tanks forward of the collision bulkhead</td><td>16 mm</td></tr>
<tr><td>For other cases</td><td>Ⓔ,but Ⓓ when $D \geqq 100 A$</td></tr>
<tr><td colspan="2">Not passing through tanks</td><td>Ⓗ</td></tr>
<tr><td rowspan="7">Scupper pipes
Sanitary pipes
(Note 1)</td><td colspan="2">Penetrating shell plating except for cargo oil tanks and cargo holds and automatic non-return valves being required</td><td>Ⓖ</td></tr>
<tr><td colspan="2">Penetrating shell plating except for cargo oil tanks and cargo holds and automatic non-return valves being omitted</td><td>Ⓓ</td></tr>
<tr><td colspan="2">Led form exposed deck and passing through cargo oil tanks</td><td>Ⓐ, but 16 mm when $D \geqq 150 A$</td></tr>
<tr><td rowspan="2">Passing through cargo holds</td><td>Not protected</td><td>Ⓐ (Note 5)</td></tr>
<tr><td>protected</td><td>Ⓒ (Note 5)</td></tr>
<tr><td colspan="2">Passing through ballast tanks</td><td>Ⓖ</td></tr>
<tr><td colspan="2">Not passing through tanks</td><td>Ⓖ</td></tr>
<tr><td rowspan="6">Air pipes,
Overflow pipes,
Sounding pipes</td><td colspan="2">passing through tanks except for cargo oil tanks</td><td>Ⓔ</td></tr>
<tr><td colspan="2">passing through cargo oil tanks</td><td>Ⓑ</td></tr>
<tr><td colspan="2">For air pipes and sounding pipes for fuel oil tanks passing through the cargo holds of the bulk carrier defined in 1.3.1(13), Part B</td><td>Ⓓ</td></tr>
<tr><td colspan="2">For tanks forming a part of ship's structure</td><td>Ⓖ</td></tr>
<tr><td rowspan="2">Exposed portions of air pipes which terminate above freeboard deck and superstructure deck (Note 1)</td><td>(Note 3)</td><td>Ⓔ</td></tr>
<tr><td>(Note 4)</td><td>Ⓖ</td></tr>
<tr><td>Fuel oil pipes</td><td colspan="2">Passing through tanks except for fuel oil tanks</td><td>Ⓔ</td></tr>
<tr><td rowspan="2">Sea water pipes</td><td colspan="2">Passing through tanks</td><td>Ⓔ</td></tr>
<tr><td colspan="2">Not passing through tanks</td><td>Ⓗ</td></tr>
<tr><td>Fresh water pipes</td><td colspan="2">Passing through tanks</td><td>Ⓔ</td></tr>
<tr><td rowspan="3">Cargo oil pipes</td><td colspan="2">Passing through ballast tanks</td><td>Ⓕ, but Ⓓ when $D \geqq 100 A$</td></tr>
<tr><td colspan="2">Passing through cargo oil tanks</td><td>Ⓔ, but Ⓕ when $D \geqq 250 A$</td></tr>
<tr><td colspan="2">Not passing through tanks</td><td>Ⓕ</td></tr>
<tr><td rowspan="2">Pipes for CO_2, fire extinguishing</td><td colspan="2">From bottles to distribution station</td><td>Ⓘ</td></tr>
<tr><td colspan="2">From distribution station to nozzles</td><td>Ⓙ</td></tr>
<tr><td colspan="3">Pipes other than the above</td><td>Ⓚ</td></tr>
</table>

Notes:

1 This is not applied for scupper pipes and sanitary pipes for ships not engaged in international voyages and ships of less

than 24m in length.

2 Ⓗ is applied when a safe (dangerous) ballast pipe passes through a safe (dangerous) ballast tank. A dangerous ballast pipe means a pipe for suction and discharge of the ballast in a dangerous ballast tank (a ballast tanks adjacent to a cargo oil tank or a ballast tank connected to a cargo oil tank through an open-ended pipe). A safe ballast pipe means a pipe for suction and discharge of the ballast in a safe ballast tank (a ballast tank other than a dangerous ballast tank).

3 For air pipes in the position I or II defined in **1.4.3.2, Part 1, Part C** leading to spaces below the freeboard deck, enclosed super structure or enclosed deck house.

4 For air pipes other than described in Note 3.

5 The thickness of the pipe need not exceed the thickness of the shell plating in way of the pipe penetration.

Table D12.6(2) Minimum Thickness of Steel Pipes(1)(3) (*mm*)

Nominal dia. (*A*) \ Corresponding Alphabet	Ⓐ	Ⓑ	Ⓒ	Ⓓ	Ⓔ	Ⓕ	Ⓖ	Ⓗ	Ⓘ(2)	Ⓙ(2)	Ⓚ
6	—	—	—	—	—	—	—	—	—	—	1.6
8	—	—	—	—	—	—	—	—	—	—	1.8
10	—	—	—	—	—	—	—	—	—	—	1.8
15	—	—	—	—	—	2.8	—	3.2	3.2	2.6	2.0
20	—	—	—	—	—	2.9	—	3.2	3.2	2.6	2.0
25	—	—	—	—	—	3.4	—	3.2	4.0	3.2	2.0
32	6.4	—	4.9	—	6.3	3.6	4.5	3.6	4.0	3.2	2.0
40	7.1	—	5.1	—	6.3	3.7	4.5	3.6	4.0	3.2	2.3
50	8.7	8.7	5.5	—	6.3	3.9	4.5	4.0	4.5	3.6	2.3
65	9.5	8.7	7.0	7.0	6.3	5.2	4.5	4.5	5.0	3.6	2.6
80	11.1	8.7	7.6	7.6	7.1	5.5	4.5	4.5	5.6	4.0	2.9
90	12.7	8.7	8.1	8.0	7.1	5.7	4.5	4.5	6.3	4.0	2.9
100	13.5	11.1	8.6	8.6	8.0	6.0	4.5	4.5	7.1	4.5	3.2
125	15.9	11.1	9.5	9.5	8.0	6.6	4.5	4.5	8.0	5.0	3.6
150	18.2	11.1	11.0	11.0	8.8	7.1	4.5	4.5	8.8	5.6	4.0
175	20.6	11.1	11.9	11.8	8.8	7.7	5.3	5.3	—	—	4.5
200	23.0	12.7	12.7	12.5	8.8	8.2	5.8	5.8	—	—	4.5
225	25.8	12.7	13.9	12.5	8.8	8.8	6.2	6.2	—	—	5.0
250	28.6	15.1	15.1	12.5	8.8	9.3	6.3	6.3	—	—	5.0
300	33.3	15.1	17.4	12.5	8.8	10.3	6.3	6.3	—	—	5.6
350	35.7	—	19.0	12.5	8.8	11.1	6.3	6.3	—	—	5.6
400	40.5	—	21.4	12.5	8.8	12.7	6.3	6.3	—	—	6.3
450	45.2	—	23.8	12.5	8.8	12.7	6.3	6.3	—	—	6.3

Notes:

1. In cases where the thickness of pipes specified in the standards does not comply with the minimum thickness in this Table, the standard pipe may be used if the difference is 0.4 *mm* or less.
2. Pipes, except those fitted in the engine room, are at least to be galvanized on their insides.
3. For pipes with a nominal diameter other than that shown in this Table, their minimum diameter will be considered by the Society in each case.

Table D12.7 Minimum Thickness of Copper and Copper Alloy Pipes (*mm*)

Outside diameter	Copper pipes	Copper alloy pipes
8-10	1	0.8
12-22	1.2	1
25-45	1.5	1.2
50-76.2	2	1.5
80-120	2.5	2
130-190	3	2.5
200-270	3.5	3
280	4	3.5

12.3 Construction of Valves and Pipe Fittings

12.3.1 General*

Valves, pipe fittings, gaskets and packings are to be suitable for their service conditions. They are also to be constructed according to standards deemed appropriate by the Society or be constructed in a manner considered equivalent thereto.

12.3.2 Special Valves and Pipe Fittings

Valves, pipe fittings, gaskets and packing used for pipes of Group I and Group II that are of a special construction or produced by a special manufacturing process are to be approved by the Society.

12.3.3 Mechanical Joints*

1 Mechanical joints are to be of construction and type according to the examples of mechanical joints shown in **Fig. D12.1**. Similar joints certified by the Society to comply with the requirements in this **12.3.3** and **13.2.4** may be acceptable.

2 Pipe unions, compression couplings, slip-on joints and similar joints are to be of type approved by the Society for the service conditions, the intended application and pressure ratings in accordance with standards separately specified by the Society.

3 In cases where the application of mechanical joints results in reduction in pipe wall thickness due to the use of bite type rings or other structural elements, this is to be taken into account in determining the minimum wall thickness of the pipe to withstand the design pressure.

4 The material of mechanical joints is to be compatible with the piping material and internal and external media.

5 Mechanical joints are to be tested to a burst pressure of 4 *times* the design pressure. For design pressures above 20 *MPa* the required burst pressure will be specially considered by the Society.

6 Where required by **Table D12.8**, mechanical joints are to be of fire resistant type approved by the Society.

7 Mechanical joints are to be tested in accordance with a program approved by the Society in accordance with standards separately specified by the Society; such a programme is to include at least the following **(1)** to **(8)**:

(1) leakage test;
(2) vacuum test (where deemed necessary by the Society);
(3) vibration (fatigue) test;
(4) fire endurance test (where deemed necessary by the Society);
(5) burst pressure test;
(6) pressure pulsation test (for Group I and II mandatory, for Group III where pressure pulsation other than water hammer is expected);
(7) assembly test (where deemed necessary by the Society); and
(8) pull out test (where deemed necessary by the Society).

Fig D12.1 Examples of Mechanical Joints

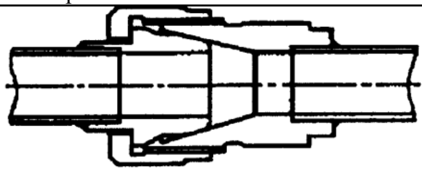
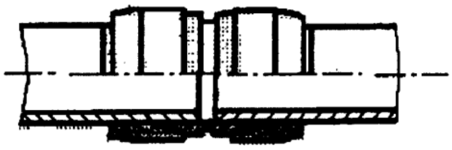
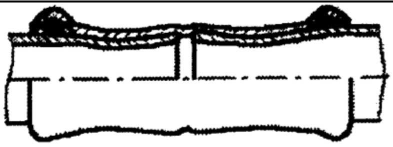
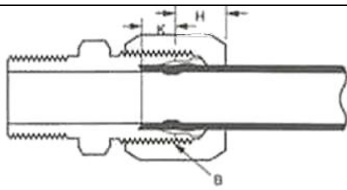
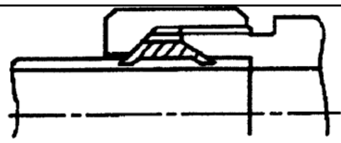
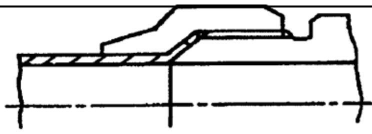
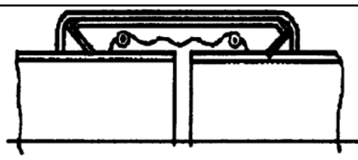
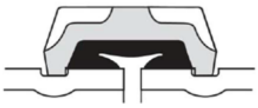
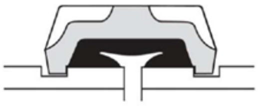
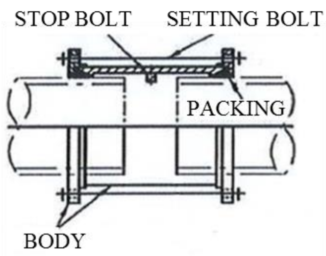
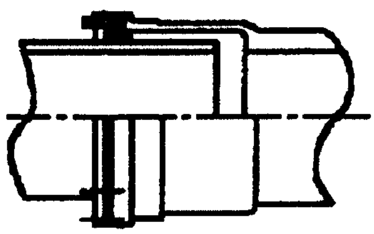
<table>
<tr><td colspan="3">Pipe Unions</td></tr>
<tr><td>Welded
and
Brazed Types</td><td colspan="2"></td></tr>
<tr><td colspan="3">Compression Couplings</td></tr>
<tr><td>Swage Type</td><td colspan="2"></td></tr>
<tr><td>Press Type</td><td colspan="2"></td></tr>
<tr><td>Typical
Compression
Type</td><td colspan="2"></td></tr>
<tr><td>Bite Type</td><td colspan="2"></td></tr>
<tr><td>Flared Type</td><td colspan="2"></td></tr>
<tr><td colspan="3">Slip-on Joints</td></tr>
<tr><td>Grip Type</td><td colspan="2"></td></tr>
<tr><td>Machine
Grooved Type</td><td colspan="2">
Roll Groove

Cut Groove</td></tr>
<tr><td>Slip Type</td><td></td><td></td></tr>
</table>

Table D12.8 Application Classifications of Mechanical Joints(1)

Application Purpose	System	Kind of Connections(3)			Classification of pipe system	Fire endurance test condition (12)
		Pipe Union	Compression Coupling	Slip-on Joint(11)		
Flammable fluids(10) (Flash point ≦60 Material °C)	Cargo oil lines (4)	+	+	+	dry	30 min dry (2)
	Crude oil washing lines (4)	+	+	+	dry	
	Vent lines (6)	+	+	+	dry	
Inert gases	Water seal effluent lines	+	+	+	wet	30 min wet (2)
	Scrubber effluent lines	+	+	+	wet	30 min wet (2)
	Main lines (4)(5)	+	+	+	dry	30 min dry (2)
	Distributions lines (4)	+	+	+	dry	30 min dry (2)
Flammable fluids(10) (Flash point > 60 °C)	Cargo oil lines (4)	+	+	+	dry	30 min dry (2)
	Fuel oil lines (5)(6)	+	+	+	wet	30 min wet (2)
	Lubricating oil lines (5)(6)	+	+	+	wet	
	Hydraulic oil (5)(6)	+	+	+	wet	
	Thermal oil (5)(6)	+	+	+	wet	
Sea Water	Bilge lines (7)	+	+	+	dry/wet	8 min dry + 22 min wet(2)
	Water filled fire extinguishing systems, e.g. fire main, sprinkler systems (6)	+	+	+	wet	30 min wet (2)
	Non water filled fire extinguishing systems, e.g. foam, drencher systems and fire main (6)	+	+	+	dry/wet	8 min dry + 22 min wet(2) (comply with Chapter 26, Part R)
	Ballast systems (7)	+	+	+	wet	30 min wet (2)
	Cooling water systems (7)	+	+	+	wet	30 min wet (2)
	Tank cleaning services	+	+	+	dry	Fire endurance test not required

Sea Water	Non-essential systems	+	+	+	dry dry/wet wet	Fire endurance test not required
Fresh water	Cooling water systems (7)	+	+	+	wet	30 min wet (2)
	Condensate returns (7)	+	+	+	wet	30 min wet (2)
	Non-essential systems	+	+	+	dry dry/wet wet	Fire endurance test not required
Sanitary/ Drains/ Scuppers	Deck drains (internal) (8)	+	+	+	dry	Fire endurance test not required
	Sanitary drains	+	+	+	dry	
	Scuppers and discharges (overboard)	+	+	-	dry	
Sounding/Vents	Water tanks/Dry spaces	+	+	+	dry, wet	Fire endurance test not required
	Oil tanks (f.p.> 60 °C) (5)(6)	+	+	+	dry	
Miscellaneous	Starting/Control air (7)	+	+	-	dry	30 min dry (2)
	Service air (non-essential)	+	+	+	dry	Fire endurance test not required
	Brine	+	+	+	wet	
	CO_2 systems (outside protected space)	+	+	-	dry	30 min dry (2)
	CO_2 systems (inside protected space)	+	+	-	dry	Mechanical joints are to be constructed of materials with melting point above 925°C. (refer to Chapter 25, Part R)
	Steam	+	+	+ (9)	wet	Fire endurance test not required

Notes:

(1) +: Application is allowed; -: Application is not allowed

(2) Fire endurance test in accordance with **9.3.2(6), Part 6 of Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

(3) If mechanical joints include any components which readily deteriorate in case of fire, the following **(4)** to **(7)** apply.

(4) Fire endurance test is to be applied when mechanical joints are installed in pump rooms and open decks.

(5) Slip-on joints are not accepted inside machinery spaces of category *A* or accommodation spaces. May be accepted in machinery spaces other than those of category *A* provided that the joints are located in easily visible and accessible positions (refer to *MSC/Circ*.734).
(6) Fire resistant types approved by the Society except in cases where such mechanical joints are installed on open decks as defined in **9.2.3-2(10), Part R of the Rules**; this excludes spaces in the cargo areas of tankers, ships carrying liquefied gases in bulk and ships carrying dangerous chemicals in bulk (as defined in **3.2.6, Part R**, **1.1.4(6), Part N** and **1.3.1(4), Part S**), but not used for fuel oil lines, fire extinguishing systems and fire mains.
(7) Fire endurance test is to be applied when mechanical joints are installed inside machinery spaces of category *A*
(8) Only above the freeboard deck.
(9) Slip type slip-on joints as shown in **Fig. D12.1** may be used for pipes on deck with a design pressure of 1.0 *MPa* or less.
(10) Piping where mechanical joints are used is also to comply with the requirements specified in **13.2.4-4**.
(11) Piping where slip joints are used is also to comply with the requirements specified in **13.2.4-6**.
(12) If a connection has passed the “30 min dry” test, it is considered suitable also for applications for which the “8 min dry + 22 min wet” and/or “30 min wet” tests are required. If a connection has passed the “8 min dry+22 min wet” test, it is considered suitable also for applications for which the “30 min wet” test is required.

Table D12.9 Application Classifications of Mechanical Joints Depending upon the Class of Pipes to which the Mechanical Joints are Fitted (1)

Types of Joints		Classes of Pipes		
		Group I	Group II	Group III
Pipe Unions	Welded and brazed type	+(2)	+(2)	+
Compression Couplings	Swage type	+	+	+
	Bite type	+	+	+
	Typical compression type	+	+	+
	Flared type	+	+	+
	Press type	-	-	+
Slip-on joints	Machine grooved type	+	+	+
	Grip type	-	+	+
	Slip type	-	+	+

Notes:

(1) + Application is allowed, - Application is not allowed.

(2) May be used for pipes whose nominal diameter is 50*A* or less.

12.3.4 Flexible Hose Assemblies*

1 Flexible hose assemblies may be used for the following pipes:

(1) Fuel oil pipes (except fuel oil injection pipes)

(2) Lubricating oil pipes

(3) Hydraulic oil pipes

(4) Thermal oil pipes

(5) Compressed air pipes

(6) Bilge and ballast pipes

(7) Fresh water and sea water pipes

(8) Steam pipes of Group III (metallic pipes only)

(9) Exhaust gas pipes (metallic pipes only)

2 Flexible hose assemblies, used for the pipes of Group I or II as well as for pipes likely to cause a fire or flooding in cases where they have been fractured, are to be approved by the Society.

3 Installation, design and construction of flexible hose assemblies are to comply with follows.

(1) Installation requirements

(a) Flexible hoses are not to be subjected to torsional deflection (twisting) under normal operating conditions.

(b) Flexible hoses are to be installed in clearly visible and readily accessible locations.

(c) The number of flexible hoses is to be kept to a minimum.

(d) Flexible hoses are to be limited to the necessary minimum length.

(e) Any hose contact that could cause rubbing and abrasion is to be avoided.

(f) The installation of flexible hoses is to take into account the allowable minimum bend radius.

(g) In cases where flexible hoses are intended to be used for flammable oil pipes which are in close proximity to heated surfaces, the risk of ignition due to a failure of the hose assembly and the subsequent release of any fluids is to be mitigated by the use of screens or other similar protection.

(h) Flexible hoses are to be installed in accordance with the manufacturer's instruction.

(2) Design requirements

(a) The design of flexible hoses is to take into account ambient conditions, compatibility with fluids under working pressure and temperature conditions.

(b) Hose clamps and other similar types of end fittings are not to be used for flexible hoses in pipes for steam, flammable oil, starting air and for sea water where failure may result in flooding. For other pipes, the use of hose clamps may be accepted where the working pressure is less than 0.5 *MPa* and provided there are double clamps at each end connection.

(c) The design of flexible hoses, where pressure pulses and/or high levels of vibration are expected to occur during use, is to take into account the maximum expected impulse peak pressure as well as any other forces due to vibration.

(3) Construction requirements

Non-metallic flexible hoses are to conform to the following requirements:

(a) Non-metallic flexible hoses are to incorporate woven integral wire braid or other suitable material reinforcement where used for pipes specified in **12.3.4-1(1)** through **(6)**. Where specially approved by the Society, the reinforcement may be exempted.

(b) In cases where non-metallic flexible hoses are to be used for fuel oil supply lines to burners, they are to have external wire braid protection.

(c) Non-metallic flexible hoses used for flammable oil and sea water pipes, where failure may result in flooding, are to be of a fire resistant type except in cases where such hoses are installed on exposed open decks and are not used for fuel oil lines.

4 The end fittings of flexible hose assemblies are to have flanges or to comply with **12.3.3** or **12.4.2**.

12.4 Connection and Forming of Piping Systems

12.4.1 Welding of Piping Systems

1 The welding for piping systems is also to comply with the requirements in **Chapter 11**.

2 Welding consumables used for the welding work of pipes belonging to Group I and Group II are to be type-approved by the Society in accordance with the requirements in **Part M,** as specified in **11.1.1-2**. In cases where compliance with this requirement, however, is deemed impractical, welding consumables that satisfy the following **(1)** and **(2)** may be accepted:

(1) Welding consumables that conform to standards recognized by the Society; and

(2) Welding consumables subjected to deposited weld metal tests, the results of which are deemed appropriate by the Surveyor

12.4.2 Direct Connection of Pipe Lengths*

1 Butt welded joints of pipe lengths are to comply with the following **(1)** and **(2)**.

(1) Butt welded joints are generally to be of a full penetration type.

(2) Except for pipes belonging to Group II and III, welding is to be as follows:

(a) double welded,

(b) use of a backing ring or inert gas back-up on first pass, or

(c) other equivalent methods recognized by the Society.

2 Slip-on sleeve welded joints are to comply with the following **(1)** and **(2)**.

(1) Slip-on sleeve welded joints are to have sleeves, sockets and weldments of adequate dimensions conforming to standards recognized by the Society.

(2) Except for pipes belonging to Group III, slip-on sleeve welded joints are not to be used for pipes specified in any of the following **(a)** to **(c)**.

(a) Pipes having a nominal diameter of more than 80 *A*

(b) Pipes conveying toxic media

(c) Pipes servicing where fatigue, severe erosion or crevice corrosion is expected to occur

3 Threaded joints are to comply with the following **(1)** to **(3)**.

(1) Threaded joints are to comply with the requirements of standards recognized by the Society.

(2) Threaded pipe joints are not to be used for the following pipes. However, the Society may allow use for pipes specified in **(e)** or **(f)** after considering the service of the pipes.

(a) Pipes conveying flammable media, except for pipes with outside diameters of 25 *mm* or less used for instrumentation.

(b) Pipes conveying toxic media.

(c) Pipes servicing where fatigue, severe erosion or crevice corrosion is expected to occur.

(d) Pipes for CO_2 systems, except inside protected spaces and in CO_2 cylinder rooms.

(e) Pipes belonging to Group I with a nominal diameter exceeding 25 *A*.

(f) Pipes belonging to Group II and Group III with a nominal diameter exceeding 50 *A*.

(3) For pipes belonging to Group I or Group II, threaded joints with tapered threads are to be used.

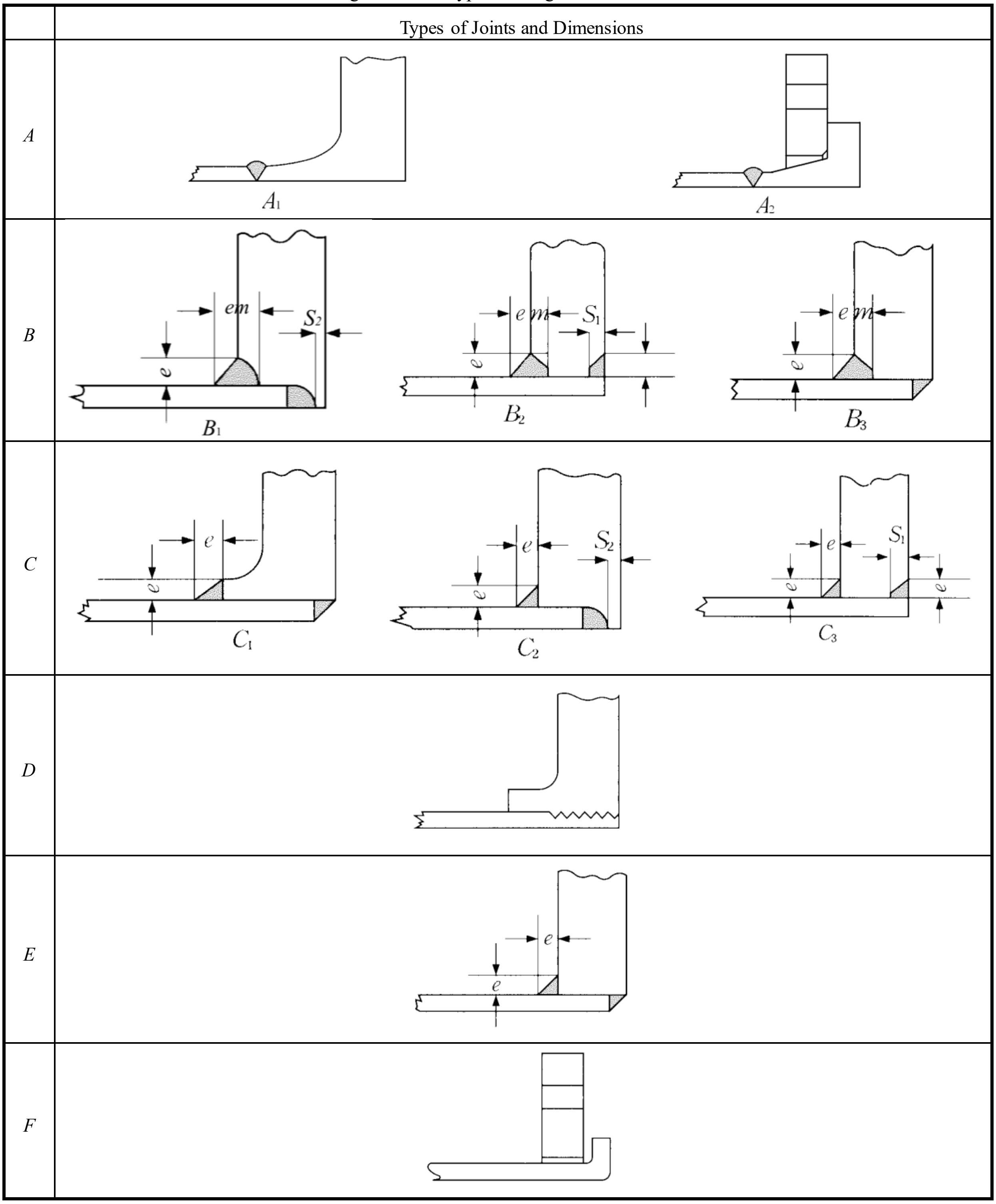
12.4.3 Connection of Pipes with Pipe Fittings*

1 Joints between pipes and pipe flanges are to be adequate for their service conditions, and their construction and strength are to conform to the requirements in **Fig. D12.2** according to their application classification shown in **Table D12.10**, or other type of joints as deemed appropriate by the Society.

2 Valves and pipe fittings made of non-ferrous metal may be mounted on non-ferrous metal pipes by brazing or soldering. In this case, the type of brazing and soldering and the method of application are to be suitable for their service conditions.

3 Joints between pipes and pipe fittings, except flanges, are to be in compliance with the requirements in **12.4.2** and **-1**.

Fig. D12.2 Type of Flange Connections



Notes:

1. Standard dimensions of welds are as follows:

   $e = 1.4\,t$

   $m = t$

   $S_1 = t$

   $S_2 = 0.5\,t$

where t is the required thickness of the pipe

2. For type D, the pipe and flange are to be screwed with a tapered thread and the pipe is to be secured to the flange by means of expansion. However, the outside diameter of the screw portion of the pipe over the thread is not to be appreciably less than the outside diameter of an unthreaded pipe.

Table D12.10 Types of Joints between Pipe and Pipe Flange and Their Application Classification

Class of Pipes	Design temperature °C	Type of Joints	
		Steam, air and water	Fuel oil, lubricating oil, hydraulic oil and thermal oil
Group I	over 400	*A*, *B* (Note 1.)	*A*, *B*
	400 or below	*A*, *B* (Note 2.)	
Group II	over 250	*A*, *B*, *C*	*A*, *B*, *C*
	250 or below	*A*, *B*, *C*, *D*, *E*	*A*, *B*, *C*, *E* (Note 3.)
Group III	-	*A*, *B*, *C*, *D*, *E*, *F* (Note 4.)	*A*, *B*, *C*, *E* (Note 3.)

Notes:

1. Type (*B*) joints may be used for steam pipes of a nominal diameter of 50 *A* or below.
2. Type (*B*) joints may be used for steam pipes of a nominal diameter of 150 *A* or below.
3. Type (*E*) joints may be used for pipes with a design pressure of 1.0 *MPa* or less.
4. Type (*F*) joints may be used for water pipes or pipes with an open end.

12.4.4 Forming of Pipes and Heat Treatment after Forming*

1 Hot forming of pipes of Group I and Group II is to conform to the following requirements:

(1) Hot forming is to be generally carried out in a temperature range of 1000 °C – 850 °C, however the temperature may be decreased to 750 °C during the forming process.

(2) For chromium-molybdenum steel and chromium-molybdenum-vanadium steel, stress relieving heat treatment is to be carried out as specified in **Table D11.8** in accordance with the kind of material. For carbon steel, carbon-manganese steel and carbon-molybdenum steel, no subsequent heat treatment is required.

(3) In cases where the hot forming is carried outside the temperature range of **(1)** above, a subsequent new heat treatment as specified in **Table D11.9** is required.

2 When pipes of Group I and Group II are subjected to cold-forming, a suitable heat treatment is to be carried out as specified in the following **(1)** and **(2)** in accordance with the kind of material with consideration given to any harmful plastic deformation due to cold-forming and development of residual stresses that may occur. For carbon steel and carbon-manganese steel with minimum tensile strengths of 320, 360 and 410 N/mm^2 (including Grade 1; Grade 2, No.2; Grade 2, No.3; Grade 3, No.2; and Grade 3, No.3 as specified in **4.2, Part K**), the heat treatment may be omitted.

(1) In cases where pipes are subjected to bending processes in such a manner that the bending radius of the pipe centreline is 4 times or less the outside diameter of the pipe, heat treatment as specified in **Table D11.9** is required.

(2) Stress relieving heat treatment as specified in **Table D11.8** is required for all materials except in the case of **(1)** above.

3 Regarding the forming and after forming heat treatment for pipes other than those specified in **Table D11.8**, they are to be approved by the Society.

12.5 Construction of Auxiliary Machinery and Storage Tanks

12.5.1 General*

1 Auxiliary machinery and storage tanks are to have sufficient strength and to be constructed so that maintenance and inspection can be easily carried out.

2 The thickness of the steel plating used for fuel oil storage tanks is not to be less than 6 *mm*, but in the case of small tanks, the thickness may be reduced to 3 *mm*.

3 Storage tanks for fuel oil and for heated lubricating oil, hydraulic oil, etc. which are installed in machinery spaces are not to have openings in the machinery space.

12.6 Tests

12.6.1 Shop Tests*

1 Tests of welds in piping systems and auxiliaries are to comply with the requirements in **Chapter 11**.

2 All pipes in Group I and Group II and integral fittings as well as, in all cases, all steam pipes, feed pipes such as feed water pipes, compressed air pipes and fuel oil pipes having a design pressure greater than 0.35 *MPa* are to be subjected to hydrostatic tests together with the relative integral fittings, after completion of manufacture but before insulation and coating at a pressure equal to 1.5 *times* the design pressure. For steel pipes and integral fittings having design temperatures above 300 °C, the requirements specified in **-3** below apply to the test pressure of hydrostatic tests.

3 The value pressure of hydrostatic tests for steel pipes and integral fittings having a design temperature above 300 °C is to that determined by the formula below. However, in cases where the pressure determined by said formula is greater than 2 *times* the design pressure, the pressure of hydrostatic tests may be reduced to a pressure equal to 2 *times* the design pressure. In addition, in order to avoid excessive stress in way of bends, T-pieces, etc., the value of test pressure may be reduced to 1.5 *times* the design pressure, subject to the approval by the Society.

$$P_h = 1.5\frac{K_{100}}{K_T}P$$

where

P_h : Test pressure (*MPa*)

K_{100}: Permissive stress of pipe material at 100 °C (N/mm^2)

K_T : Permissive stress of pipe material at the design temperature (N/mm^2)

P : Design pressure (*MPa*)

4 In cases where primary general membrane stress in the pipe wall is expected to exceed 90 % of the specified yield stress or proof stress, at the test pressure specified in **-2** and **-3**, the test pressure is to be reduced so that such stress does not exceed the stress to 90 % of the specified yield stress or proof stress at the testing temperature.

5 Valves and pipe fittings non-integral with the piping system intended for pipes in Group I and Group II are to be subjected to hydrostatic tests in accordance with standards recognized by the Society, but at a pressure equal to 1.5 *times* the design pressure.

6 Valves and distance pieces fitted to the ship's side below the load line are to be subjected to hydrostatic tests at a pressure of 1.5 times the design pressure or 0.5 *MPa*, whichever is greater.

7 The pressure parts of auxiliaries (excluding auxiliary machinery for specific use etc.) are to be subjected to hydrostatic tests at a pressure equal to 1.5 times the design pressure or 0.2 *MPa*, whichever is greater.

8 Free standing fuel oil storage tanks are to be subjected to hydrostatic tests at a pressure corresponding to a water head of 2.5 *m* above the top plate.

9 Auxiliaries (excluding auxiliary machinery for specific use etc.) are to be subjected to running tests as deemed appropriate by the Society.

10 When, for technical reasons, it is not possible to carry out complete hydrostatic testing specified in **-2** and **-3** above before assembly on board, for all sections of piping, the testing may be carried out in conjunction with the checking required by **13.17.2-3** or **14.6.2-2** provided that the test plans referred to in **2.1.7-1(2), Part B** containing the closing lengths of piping, particularly in respect to the closing seams, are submitted to the Society and approved.

12.6.2 Tests after Installation On Board.

The applicable requirements in **13.17.2-3** and **-4** or **14.6.2-2** to **-5**, apply to tests of piping systems after assembly on board.

Chapter 13 PIPING SYSTEMS

13.1 General

13.1.1 Scope

The requirements in this Chapter apply to piping systems.

13.1.2 Drawings and Data*

Drawings and data to be submitted are generally as follows:

(1) Drawings (with materials, sizes, kinds, design pressures, design temperatures, etc. of pipes, valves, etc.)

(a) Piping diagrams for the entire ship

(b) Piping diagrams for the engine room

(c) Methods for preventing oil from spraying out from flange joints and special joints (threaded pipe joints, mechanical joints, etc.) in fuel oil, lubricating oil and other flammable oil piping (if any)

(d) Other drawings considered necessary by the Society

(2) Data

(a) Machinery particulars

(b) Other data considered necessary by the Society

13.2 Piping

13.2.1 General*

1 Installation of pipes

(1) Ample provision is to be made in consideration of the effects of expansion, contraction, deflection of the hull and vibration. Pipes are to be supported at suitable spans to avoid any excessive load.

(2) The number of detachable pipe connections is to be minimized as far as practicable.

2 Radius of curvature of pipes

In cases where pipes are bent, the radius of curvature at the center line of a pipe is generally not to be less than twice the external diameter of the pipe.

3 Functions of pipes

Pipes are to be so arranged so that any lingering drainage and air pockets as well as any pressure loss in the pipes do not have any adverse effects on the performance of any machinery.

4 Piping in the vicinity of electrical equipment

Pipes are not to be laid in way of electrical equipment such as generators, switchboards, control gears, etc. as much as possible. In case where such a situation is unavoidable, care is to be taken to make sure that no flanges or joints are arranged over or near any electrical equipment, unless provisions are made to prevent any leakage from pouring onto the equipment.

5 Protection of piping systems and fittings

(1) All pipes, including seawater pipes, valves, cocks, pipe fittings, valve operating rods, handles, etc. in cargo holds for dry cargoes (including cargo spaces of container ships and ro-ro ships) are to be protected from impact of cargo where they are liable to be damaged. Where a casing is provided for protection, the casing is to be constructed so as to facilitate easy removal for inspection.

(2) For pipes arranged in positions inaccessible for maintenance and inspection, due consideration such as corrosion protection is to be given to prevent corrosion.

6 Relief valves

(1) All piping systems which may be subjected to an internal pressure that exceeds design pressure are to be safeguarded with relief valves or, as an alternative, overpressure protective devices.

(2) Discharge ends of relief valves or overpressure protective devices are to be led into safe spaces.

7 Pressure and temperature measuring devices

(1) Pressure and temperature measuring devices are to be provided on piping systems where considered necessary.

(2) Cocks or valves are to be provided at the root of pressure measuring devices in order to isolate them from the pipes under a pressurized condition.

(3) In cases where thermometers are fitted in fuel oil, lubricating oil and other flammable oil piping or apparatuses, the thermometer is to be put in a safe protective pocket to prevent any oil from spraying out if the thermometer should fracture or be removed.

8 Distinct marking of piping systems

(1) Pipes located in spaces where deemed necessary for safety are to be marked with distinctive colours to avoid any mishandling.

(2) Identification plates, which show the purpose of a valve, are to be affixed to valves where deemed necessary for safety, and all valves which are used for fire extinguishing are to be painted red.

(3) Identification plates are to be affixed to the open ends of air pipes, sounding pipes and overflow pipes.

9 Cleaning of piping systems

Piping systems are to be cleaned after fabrication or installation on ship where considered necessary.

13.2.2 Connection and Common Use of Pipes

1 Connection of oil pipes with other pipes

(1) Fuel oil pipes are to be entirely separate from other pipes, unless means are provided to prevent any accidental contamination with other liquids while in operation.

(2) Lubricating oil pipes are to be entirely separate from all other pipe lines.

(3) Fresh water pipes, used for boiler feed water or drinking water, are to be entirely separate from other pipes to avoid any accidental contamination with oil or oily water.

(4) Oil pipes and heating pipes in deep tanks which may be used for carriages of general cargo are to be capable of being disconnected or are to be provided with suitable arrangements such as blank flanges or spool pieces. Bilge pipes and ballast pipes in such deep tanks are to comply with the requirements in **13.5.1-10**.

2 Common use of sea water pipes and fresh water pipes

Sea water pipes and fresh water pipes are to be separated, unless adequate measures are taken to avoid any accidental contamination between the two.

13.2.3 Penetration of Pipes*

In cases where pipes pass through watertight bulkheads, decks, top plates, bottom plates as well as bulkheads of deep tanks and inner bottom plating, measures are to be taken to ensure the watertightness of the structures.

13.2.4 Mechanical Joints*

1 The requirements of this **13.2.4** applies to pipelines where the pipe unions, compression couplings and slip-on joints specified in **12.3.3** are used and also correspondingly apply to pipelines where similar joints are used.

2 Application of mechanical joints and their acceptable use is to be in accordance with the following **-3** to **-7**, in addition to the requirements in **Table D12.8** for each service and the requirements in **Table D12.9** depending upon the class of pipes and pipe dimensions. In particular cases, sizes of mechanical joints in excess of those specified in **Table D12.9** may be accepted by the Society in cases where such mechanical joints comply with a national and/or international standard recognized by the Society.

3 Mechanical joints, which in the event of damage could cause fire or flooding, are not to be used in piping sections directly connected to the ship's side below the freeboard deck of cargo ships or tanks containing flammable fluids.

4 The number of mechanical joints in the flammable fluid systems specified in **Table D12.8** is to be kept to a minimum. In general, flanged joints conforming to standards recognized by the Society are to be used.

5 Piping in which a mechanical joint is fitted is to be adequately adjusted, aligned and supported. Supports or hangers are not to be used to force alignment of piping at the point of connection.

6 The following **(1)** and **(2)** limitations apply to use of slip-on joints, in addition to **-2** to **-5** above.

(1) Slip-on joints are not to be used on pipelines in cargo holds, tanks and other spaces which are not easily accessible (refer to *MSC/Circ.*734), except that these joints may be permitted in tanks that contain the same media.

(2) Usage of slip type slip-on joints as the main means of pipe connection is not permitted except in cases where compensation of axial pipe deformation is necessary.

7 The installation of mechanical joints is to be in accordance with the manufacturer's assembly instructions. Where special tools and gauges are required for installation of the joints, these are to be supplied by the manufacturer.

13.2.5 Bulkhead Valves*

1 Valves or cocks, such as drain valves, which do not constitute any part of a piping system is not to be fitted on collision bulkheads.

2 Pipes passing through collision bulkheads are to be fitted with a remotely controlled valve capable of being operated from above the freeboard deck. The valve is to be normally closed. If the remote control system failure during operation of the valve, the valve is to be close automatically or be capable of being closed manually from a position above the freeboard deck. The valve is to be located at the collision bulkhead on either the forward or aft side, provided the space on the aft side is not a cargo space.

3 Valves and cocks, such as drain valves, which do not constitute any part of a piping system, may be fitted on watertight bulkheads other than collision bulkheads, provided that they are readily accessible at any time for inspection. Such valves and cocks are to be operable from above the bulkhead deck and are to be provided with an indicator to show whether they are open or closed, except in cases where the valves or cocks are secured to a fore or aft bulkhead located inside the engine room.

4 Means for controlling valves or cocks from above freeboard decks or bulkhead decks are to be constructed so that the weights thereof are not supported by the valves or the cocks.

13.2.6 Prevention of Freezing of Pipes

Suitable measures are to be taken to prevent the freezing of any bilge pipes, air pipes, sounding pipes, drain pipes, etc. that pass through or are arranged near any refrigerated chambers, in cases where the inner surfaces of the pipes are at risk of freezing.

13.2.7 Prevention of Counterflow through Drain Pipes

When any drain pipes in the engine room are led into double-bottom tanks, and when, in cases where sea water flows into the tank by grounding, etc., there is a danger of flooding from these drain pipes, a stop valve or other suitable device that stops the counterflow of sea water is to be provided. This device is to be readily operable from the engine room floor. However, these requirements do not apply to ships of a length less than 100 *m*.

13.2.8 Drain Installation around Boilers

A coaming of at least 100 *mm* in height is to be provided around boilers, and the drain inside the coaming is to be into a bilge well or bilge tank etc.

13.3 Sea Suction Valves and Overboard Discharge Valves

13.3.1 Sea Suction Pipes and Overboard Discharge Pipe Connections*

Sea inlet and overboard discharge pipes are to be connected to valves or cocks which are fitted in accordance with the requirements given in **13.3.2-2** and **-3**. However, discharge pipes, which are located above the freeboard deck and are not required to have non-return valves of substantial wall thickness in accordance with the provisions of **13.4.1-7**, need not, up to an appropriate level above the freeboard deck, comply with the provisions of **13.3.2-3**.

13.3.2 Location and Construction of Sea Suction Valves, Overboard Discharge Valves, etc. *

1 The locations of overboard discharges are not to be such that water can be discharged into lifeboats and liferafts at fixed launching positions, including those under launching devices, when they are launched, unless special provisions have been taken to prevent such discharge.

2 Sea suction valves and overboard discharge valves or cocks fitted to the ship’s side as well as any sea chests forming a part of the ship’s structure or any distance pieces attached to the shell plating are to be located at easily accessible positions.

3 Valves or cocks prescribed in **-2** are to be fitted in accordance with the following **(1)** or **(2)**:

(1) Valves or cocks are to be fitted to doublings which are welded to the shell plating or sea chest by using stud bolts not passing through the shell plating and sea chest.

(2) Valves or cocks are to be fitted by bolts to distance pieces attached to the shell plating. In this case, the distance piece is to be of rigid construction and as short as practicable.

4 Valve spindles of sea suction valves are to be extended above the lower platform where they are easily operable. Power-operated sea suction valves are to be also arranged for manual operation. Sea suction valves are to be provided with indicators to show whether they are open or closed.

5 Overboard discharge valves and cocks are to be fitted with spigots passing through the shell plating and protection rings specified in **-6(1)**, but spigots on valves or cocks may be omitted if these fittings are attached to pads or distance pieces which themselves form spigots in way of the shell plating and protecting rings. Overboard discharge valves and cocks are to be provided with indicators to show whether they are open or closed.

6 Blow-off valves or cocks of boilers and evaporators are to comply with the following requirements in **(1)** and **(2)**:

(1) Blow-off valves or cocks of boilers and evaporators are to be fitted in accessible positions and to be provided with protection rings on the outside of the shell plating to prevent corrosion.

(2) Cock handles are not to be capable of being removed unless the cocks are shut, and, if valves are fitted, wheel handles are to be suitably fastened to the spindle.

13.3.3 Construction of Sea Chests

Sea chests are to be of substantial construction and not to block off suction due to airlocking.

13.3.4 Gratings of Sea Suctions

1 Gratings are to be fitted at the sea inlets. The area of grating openings is not to be less than twice the total inlet sea suction valve opening area.

2 Provisions are to be taken for cleaning the gratings specified in **-1** by use of low pressure steam, compressed air, etc.

13.4 Scuppers, Sanitary Discharges, etc.

13.4.1 General*

1 Scupper pipes, sufficient in number and size, to provide effective drainage are to be provided for all decks. However, the Society may permit this means of drainage to be dispensed with in any particular compartment of any ship or class of ship, if it is satisfied that, due to the size or internal subdivision of those spaces, the safety of the ship is not thereby impaired. For the special hazards associated with loss of stability when fitted with fixed pressure water-spraying fire-extinguishing systems refer to **20.5.1-4, Part R**.

2 Scupper pipes, draining weather decks and spaces within superstructures and deckhouses, which are not provided with access openings equipped with closing means in accordance with the requirements in **11.3.2.6, Part 1, Part C**, are to be led overboard.

3 Scupper pipes, originating from within enclosed superstructures or enclosed deckhouses on freeboard decks, are to be led directly into inboard bilge wells. Alternatively, they may be led overboard in cases where they are provided with valves in accordance with the following requirements:

(1) Each separate discharge is to have one automatic non-return valve with a positive means of closing it from a position above the freeboard deck, or one automatic non-return valve having no positive closing means and one stop valve controlled from above the freeboard deck. However, where the scuppers are led overboard through the shell plating in way of a manned engine room, fitting a locally operated positive closing valve, together with a non-return valve inboard to the shell plating will also be accepted. The means for operating the positive action valve from above the freeboard deck are to be readily accessible and provided with an indicator showing whether the valve is open or closed.

(2) However, in cases where the vertical distance from the load line to the inboard end of the scupper pipe exceeds $0.01L_f$, the scupper pipe may have two automatic non-return valves without any positive means of closing in lieu of valves prescribed in **(1)**. In this case, the inboard valve is to be located above the level of the tropical load line and is to always be accessible for inspection under service conditions. If it is not practicable to fit an inboard valve above the specified waterline then it can be accepted below provided that a locally controlled stop valve is fitted between the two automatic non-return valves.

(3) In cases where the vertical distance prescribed in **(2)** exceeds $0.02L_f$, a single automatic non-return valve, without any positive means of closing, may be accepted in lieu of valves prescribed in **(1)** and **(2)** subject to Society approval.

4 Scupper pipes from spaces below the freeboard deck are to be led directly into inboard bilge wells. Alternatively, they may be led to overboard in cases where they are provided with valves in accordance with the following requirements:

(1) Each separate discharge is to have one automatic non-return valve with a positive means of closing it from a position above the freeboard deck, or one automatic non-return valve having no positive closing means and one stop valve controlled from above the freeboard deck. The means for operating the positive action valve from above the freeboard deck are to be readily accessible and provided with an indicator showing whether the valve is open or closed.

(2) However, in cases where the vertical distance from the load line to the inboard end of the scupper pipe exceeds $0.01L_f$, the scupper pipe may have two automatic non-return valves without any positive means of closing in lieu of valves prescribed in

(1). In this case, the inboard valve is to be located above the level of the deepest subdivision draught specified in **2.3.1.2(3), Part 1, Part C** and is to always be accessible for inspection under service conditions.

5 Notwithstanding the requirements in **-3**, scupper pipes from enclosed cargo spaces on the freeboard deck are to be in accordance with the following requirements:

(1) In cases where the freeboard deck is such that the deck is immersed when the ship heels more than 5 degrees, scupper pipes are to be led directly overboard and fitted in accordance with the requirement specified in **-3**. If the requirements specified in **(2)(a)** thorough **(c)** are fulfilled, scupper pipes may be led directly into inboard bilge wells.

(2) In cases where the freeboard deck is such that the deck is immersed when the ship heels 5 degrees or less, scupper pipes are to be in accordance with the following requirements:

(a) Scupper pipes are to be led directly into inboard bilge wells.

(b) High water level alarms are to be provided in the bilge wells that are fed into by scupper pipes.

(c) In cases where enclosed cargo spaces are protected by a carbon dioxide fire-extinguishing system, deck scuppers are to be fitted with means to prevent any escape of fire-extinguishing gas.

6 Notwithstanding the requirements in **-3** and **-4**, only one stop valve may be arranged for overboard discharge pipes which are always closed, except when discharging, during navigation. However, this stop valve is to be capable of being closed from an easily accessible position during navigation by closing devices with indicators.

7 Scuppers originating at any level and penetrating the shell plating at either more than 450*mm* below the freeboard deck or below 600*mm* above the load line are to be provided with non-return valves at the shell plating. These valves, unless specifically required by **-3** and **-4**, may be omitted provided that the thickness of the scupper pipes complies with the requirements in **Table D12.6**.

8 In cases where fixed pressure water-spraying systems are fitted in closed vehicle and Ro-Ro spaces and special category spaces, drainage systems are to comply with **20.5.1-4** and **20.5.1-5, Part R of the Rules** in addition to those requirements specified in **-1** to **-7** above.

13.4.2 Common Overboard Discharge

The number of scuppers, sanitary discharges and other similar openings in the shell plating is to be kept to a minimum by either making each discharge serve as many sanitary and other pipes as possible, or using other satisfactory means. However, different systems of overboard discharges are, in general, not to be connected to each other, unless specially approved by the Society.

13.4.3 Sanitary Discharge

Sanitary discharges are to comply with the requirements in **13.4.1** and **13.4.2**.

13.4.4 Ash-shoots and Rubbish-shoots

1 For ash-shoots and rubbish-shoots, instead of a non-return valve with a positive means of closing from a position above the freeboard deck, two gate valves, which comply with the following requirements, are acceptable.

(1) The two gate valves are to be controlled from the working deck of the chute.

(2) The lower gate valve is to be controlled from a position above the freeboard deck. An interlock system between the two valves is to be arranged.

(3) The inboard end is to be located above the waterline formed by an 8.5 *degrees* heel to port or starboard at a draft corresponding to the assigned summer freeboard, but not less than 1,000 *mm* above the summer waterline. Where the inboard end exceeds $0.01L_f$ above the summer waterline, valve control from the freeboard deck is not required provided that the inboard gate valve is always accessible under service conditions.

2 A hinged weathertight cover at the inboard end of the chute together with a discharge flap may be acceptable in lieu of the upper and lower gate valves that comply with the requirements in **-1**. In this case, the cover and flap are to be arranged with an interlock so that the discharge flap cannot be operated until the hopper cover is closed.

3 Controls for the gate valves and/or hinged covers are to be clearly marked: "Keep closed when not in use."

4 For those ships in which the damage stability requirements specified in **2.3, Part 1, Part C** are applied; the following requirements are to be satisfied in cases where the inboard end of the chute is below the freeboard deck.

(1) Inboard-end hinged covers/valves are to be watertight.

(2) Valves are to be a screw-down non-return valve fitted in an easily accessible position above the deepest load line.

(3) Screw-down non-return valves are to be controlled from positions above the bulkhead deck and provided with open/closed indicators. Valve controls are to be clearly marked: "Keep closed when not in use."

13.5 Bilge and Ballast Piping

13.5.1 General*

1 An efficient bilge pumping system is to be provided, capable of pumping out and draining any watertight compartment under practical conditions, except for tanks specially used to hold liquids and those spaces provided with efficient means of pumping.

2 An efficient ballast piping system, capable of pumping ballast water into and out off any tanks for holding ballast water under practical conditions, is to be provided.

3 In cases where fixed pressure water-spraying fire-extinguishing systems or other fixed systems, which will supply copious quantities of water, are fitted for cargo spaces as required by **19.3.1-3**, **19.3.9**, **20.2.1**, **20.5.1-1(3)**, **20.5.1-2** or **20.5.1-4**, **Part R**, bilge pumping systems for such cargo spaces are to comply with these requirements as well in addition to the requirements in this Chapter.

4 Suitable measures are to be taken so that bilge pumping systems prevent the possibility of any ingress of sea water into any watertight compartments and to prevent any bilge from inadvertently passing from one compartment to another. To achieve this requirement, all bilge distribution boxes and manually operated valves in connection with bilge pumping systems are to be in positions which are accessible under ordinary conditions. All valves in bilge distribution boxes are to be of a non-return type.

5 Bilge suction pipes used for draining cargo holds and machinery room and shaft tunnels are to be entirely separate from any other pipe that is not a bilge suction pipe.

6 Bilge pipes passing through deep tanks used exclusively for ballasting and bilge pipes and ballast pipes passing through deep tanks other than ballast tanks are to be led through an oiltight or watertight pipe tunnels; or, are to be of sufficient thicknesses in accordance with the requirements in **Table D12.6** and all of their joints are to be welded.

7 Bilge pipes passing through double bottom tanks are to be led through oiltight or watertight pipe tunnels; or, they are to be of sufficient thickness in accordance with the requirements in **Table D12.6**.

8 Bilge pipes passing through double bottoms, side tanks, bilge hopper tanks or void spaces, in cases where there is a possibility of these pipes being damaged due to grounding or collision, are to be provided with non-return valves near their bilge suctions or stop valves capable of being closed from readily accessible positions.

9 Ballast piping systems are to be provided with suitable provisions, such as non-return valves or stop valves, which can be kept closed at all times, excluding times of ballasting and de-ballasting; and, which are provided with indicators to show whether such valves are opened or closed, in order to prevent the possibility of any inadvertent ingress of sea water into the ballast tanks or of any ballast water passing from one ballast tank to another.

10 In cases where a hold is intended to alternate between carrying ballast water and cargo, adequate provisions, such as blank flanges or spool pieces, are to be made in the ballast piping system to prevent any inadvertent ingress of sea water through ballast pipes when carrying cargo as well as in bilge piping systems to prevent any inadvertent discharge of ballast water through bilge pipes when carrying ballast water.

11 Ballast piping system is not to be connected with a fuel oil tank. However, the requirements may be dispensed with where as deemed appropriate by the Society in consideration of the arrangements of the ballast piping system.

13.5.2 Terminology

1 A Main Bilge Line is the part of a bilge suction line which forms the main of bilge suction line connected to independently powered bilge pumps specified in **13.5.4-1** and to which all branch bilge suction pipes from the bilge suctions specified in **13.5.5** and **13.5.7-1** to **-4** are connected.

2 A Branch Bilge Suction Pipe is a pipe connected to the main bilge line from the bilge suction of each compartment.

3 A Direct Bilge Suction Pipe is a bilge suction pipe which is connected directly to an independently powered pump specified in **13.5.4-1** and arranged entirely separately from other pipes.

4 An Emergency Bilge Suction Pipe is a bilge suction pipe which is to be used in an emergency and is connected directly to an independently powered pump specified in **13.5.7-6(1)** or **-7(1)**.

13.5.3 Size of Bilge Suction Pipes*

1 The internal diameter of main bilge lines, direct bilge suction pipes and branch bilge suction pipes of watertight compartments is to be calculated using the following formulae **(1)** and **(2)** or, standard pipes nearest in internal diameters to the calculated diameter are to be used. In cases where the internal diameter of the standard pipes closest to the calculated value is short of that value by 13*mm* or more, a standard pipe of one grade higher is to be used.

(1) For main bilge lines and direct bilge suction pipes:

$$d = 1.68\sqrt{L_f(B+D)} + 25\ (mm)$$

(2) For branch bilge suction pipes:

$$d' = 2.15\sqrt{l(B+D)} + 25\ (mm)$$

where

d : Internal diameter of the main bilge line or direct bilge suction pipes (mm).

d' : Internal diameter of branch bilge suction pipes (mm).

B and D : Ship length, breadth and depth respectively (m)

L_f : Length (m) for freeboard specified in **2.1.3, Part A of the Rules**.

However, for ships to which the requirement **13.4.1-5(2)** is applied, "D" is to be considered as follows:

(a) For ships which have enclosed cargo spaces that extend for the full length of the ship, "D" is to be considered as the depth of ship measured to the next deck above the freeboard deck (m)

(b) For ships which have enclosed cargo spaces that do not extend for the full length of the ship, "D" is to be considered as the depth of ship plus $l' \times h/L_f$ (m), where l' and h are the aggregate length and height respectively of all the enclosed cargo spaces.

l : Length of the compartment to be served by the branch bilge suction pipes (m).

2 Internal diameters of main bilge lines are not to be less than the internal diameters of any branch bilge suction pipes obtained from the formula in **-1(2)**.

3 Internal diameters of direct bilge suction pipes are also to comply with the requirements in **13.5.7-5(1)** and **(2)**.

4 In cases where bilge suctions are provided at the fore and after parts of the cargo hold in accordance with the requirements in **13.5.5-1**, the internal diameter of the branch bilge suction pipe at the fore part may be reduced to 0.7 times that obtained from the formula in **-1(2)**.

5 In cases where bilge pumps in engine rooms are exclusively used for bilge drainage in the engine room, the internal diameters of the main bilge line and any direct bilge suction pipes may be reduced to that obtained from the following formula:

$$d = \sqrt{2}(2.15\sqrt{l(B+D)} + 25)(mm)$$

where

l : Length of the engine room (m).

d, B and D: As defined in **-1**.

6 The internal diameters of branch bilge suction pipes are not to be less than 50mm. However, the internal diameters of those used for the drainage of a small compartment may be reduced to 40mm where considered acceptable by the Society.

7 The internal sectional area of bilge suction pipes connecting two or more branch bilge suction pipes to the main bilge line is not to be less than the sum of internal sectional areas of the largest two branch bilge suction pipes, but need not exceed the internal sectional area of the main bilge line obtained from the formula in **-1(1)**.

8 The internal diameters of bilge suction pipes in fore and after peaks as well as shaft tunnels are not to be less than 65mm. However, the internal diameter of these pipes may be reduced to 50mm in ships less than 60m in length.

13.5.4 Bilge Pumps*

1 Number of bilge pumps

(1) All ships are to be provided with at least two independently powered bilge pumps that are connected to the main bilge suction pipes. However, in ships not more than 90m in length, one of the required pumps may be driven by the main propulsion machinery.

(2) Ballast, sanitary and general service pumps driven by independent power may be accepted as independently powered bilge pumps, provided that they are connected properly to the main bilge line.

(3) In cases where considered acceptable by the Society, one of the independently powered bilge pumps prescribed in **(1)** may be substituted for by an eductor that is driven by a sea water pump and not driven by a bilge pump. In this case, the capacity of the eductor is to comply with the requirement in **-2**.

2 Capacity of bilge pumps

Each pump specified in **-1** is to be capable of discharging bilge, through the main bilge line specified in **13.5.3**, of an amount not less than that obtained from the following formula:

$$Q = 5.66d^2 \times 10^{-3}$$

where

Q : Required quantity (m^3/hr).

d : Internal diameter of the main bilge line specified in **13.5.3** (mm).

In cases where one of these pumps is of a capacity slightly less than what is required, the deficiency may be made good by any excess capacity of the other pump.

3 Types of bilge pumps

All of the independently powered bilge pumps prescribed in **-1** are to be of a self-priming type or an equivalent thereto; and, they are to be so arranged that they always available for immediate use.

4 Connection of bilge pumps to suction pipes

All of the power driven pumps prescribed in **-1** are to be arranged for discharging bilge from all holds, engine rooms and shaft tunnels. However, in cases where, an eductor is used exclusively for bilge drainage in a hold, the bilge suction pipe of this hold need not be connected to the bilge pumps prescribed in **-1**. In this case, the eductor is to be so arranged as to be driven by two or more pumps. The capacity of the sea water pump for sending water to drive the eductor, the capacity of the eductor and the internal diameter of the suction pipe are to all be considered appropriate by the Society.

13.5.5 Bilge Suction Arrangement in Holds

1 In ships having only one hold exceeding $33m$ in length, bilge suctions are to be provided in suitable positions in both the after half-length and in the forward half-length of the hold.

2 In cases where inner bottom plating extends to the ship's sides, bilge suctions are to be placed in wells at both wings and also at the centre line if the top plating has an inverse camber.

3 In cases where a ceiling is fitted over the bilges of the holds, proper arrangement is to be made whereby water in the hold compartments may find its way to the suctions.

4 In refrigerated chambers, the insulation for bilge wells and bilge suction hoses in bilge ways is to be of plug type and removable.

5 In refrigerated chambers, the insulation in way of bilge suction pipes is to be removable only to a degree necessary to allow proper inspection.

13.5.6 Bilge Drainage from the Top of Deep Tanks, Fore and After Peak Tanks and Chain Lockers*

1 Bilge of the fore and after peak tanks, decks forming the top of these tanks and chain lockers may be drained by eductors or hand pumps. These eductors or hand pumps are to be capable of being operated at any time from accessible positions above the load water line.

2 Efficient means are to be provided for draining bilge from the top of deep tanks and other watertight flats.

3 Drainage from spaces above deep tanks may be led to bilge wells in the shaft tunnel or an accessible compartment. In this case, these pipes are not to be more than $65A$ in nominal diameter and are to be provided with quick-acting self-closing valves located in an accessible position.

4 In cases where a suction line passes through a collision bulkhead, it is to comply with the requirements in **13.2.5-2**.

13.5.7 Bilge Suction Arrangements in Engine Rooms

1 In cases where there is no double bottom in the engine room, at least two bilge suctions are to be provided near the centre line of the ship. One of these suctions is to be for a branch bilge suction pipe and the other is to be for a direct bilge suction pipe. If the rise of floor is less than 5 *degrees*, additional bilge suctions are to be provided at wings.

2 In cases where there is a double bottom in the engine room with bilge ways on both wings, one branch bilge suction and one direct bilge suction are to be provided at each wing.

3 In cases where double bottom plating extends to the ship's sides, bilge wells are to be placed at each side so far as is reasonable and practicable, and one branch bilge suction and one direct bilge suction are to be provided for each bilge well.

4 In cases where the engine room is separated by watertight bulkheads from a boiler compartment and auxiliary engine room, the bilge suction pipe arrangements in the boiler room and the auxiliary engine room are to comply with the requirements in **-1** in the case of no double bottom construction; and, they are to comply with the requirements in **-2** or **-3** in the case of double bottom construction. However, only one direct bilge suction will be accepted even in the case of double bottom construction.

5 Direct bilge suction pipes are to comply with the following requirements:

(1) The internal diameter of direct bilge suction pipes is not to be less than that obtained from the formula in **13.5.3-1(1)**. In cases where a direct bilge suction pipe is provided on each side of the engine room in accordance with the requirements in **-2** or **-3**, the internal diameter of one of these direct bilge suction pipes may be reduced to that obtained from the formula in **13.5.3-1(2)**. In this case, the pipe reduced in diameter is to be located on the same side as the emergency bilge suction pipes specified in **-6** or **-7**.

(2) Notwithstanding the requirements in **(1)**, in cases where the compartments with small dimensions, the internal diameter of the direct bilge suction pipes may be adequately reduced.

6 Emergency bilge suction pipes for ships with steam turbines used as main propulsion machinery (excluding electric propulsion ships) are to comply with the following requirements:

(1) In the above ships, an emergency bilge suction pipe with a screw-down non-return valve having a wheel handle which is extended above the floor grating in the engine room, is to be fitted to the suction end of the main circulating pump. The suction pipe of this pump is to be fed into a suitable level in the engine room in order to discharge bilge in case of emergency. The internal diameter of such a suction pipe is not to be less than two-thirds of the diameter of that of pump suction.

(2) In cases where the main circulating pump is not considered suitable for bilge discharge, an emergency bilge suction pipe may be fed into the largest available power pump in the engine room other than the bilge pumps specified in **13.5.4-1** in lieu of the main circulating pump. The capacity of this pump is not to be less than that required by **13.5.4-2**. The internal diameter of such a suction pipe is to be equal to that of pump suction.

(3) In cases where the pump prescribed in **(1)** or **(2)** is of a self-priming type, the direct bilge suction arranged on the same side of the ship as emergency bilge suction may be omitted.

7 Emergency bilge suction pipes for ships with reciprocating internal combustion engines or gas turbines used as main propulsion machinery (excluding electric propulsion ships) are to comply with the following requirements:

(1) In the avobe ships, an emergency bilge suction pipe with a screw-down non-return valve having a wheel handle which is extended above the lower platform in the engine room is to be fitted to the main cooling water pump. The suction pipe is to be fed into a suitable level in the engine room to discharge bilge in case of emergency. The internal diameter of such suctions pipe is to be equal to that of pump suction.

(2) In cases where the main cooling water pump is not considered suitable for bilge discharge, the emergency bilge suction pipe may be fed into the largest available power pump in the engine room other than the bilge pumps specified in **13.5.4-1** in lieu of the main cooling water pump. The capacity of this pump is not to be less than that required by **13.5.4-2**. The internal diameter of such a suction pipe is to be equal to that of pump suction.

(3) In cases where the pump prescribed in **(1)** or **(2)** is of a self-priming type, any direct bilge suction arranged on the same side of the ship as the emergency bilge suction may be omitted.

13.5.8 Bilge Wells*

1 The depth of bilge wells constructed in double bottoms and the vertical distance between the bottom plating and the bottom of bilge wells are to comply with the requirements in **10.2.1.2, Part 1, Part C**.

2 The capacity of each bilge well is not to be less than $0.17 m^3$.

3 Bilge wells may be substituted for by steel bilge hats of a reasonable capacity where the spaces to be drained are small or not capable of being provided with bilge wells of the volume prescribed in **-2**.

4 In cases where access manholes to bilge wells of cargo holds are necessary, they are to be located as near to the bilge suctions as practicable. The placing of any of the aforementioned manholes on the fore and aft bulkheads as well as the inner bottom plating of the engine room is to be avoiding as far as practicable.

13.5.9 Mud Boxes and Strum Boxes*

1 Bilge suction pipes, except for those emergency bilge suction pipes in engine rooms and shaft tunnels, are to be provided with mud boxes that are easily accessible from above the platform in the engine room, and have covers which are easily opened and closed. In addition, straight tail pipes to bilge wells are to be fitted to the suction side of these mud boxes.

2 Bilge suction ends in hold spaces are to be provided with strum boxes that have a perforation approximately $10 mm$ in diameter, except in cases approved by the Society, and that have an open area of more than twice the area of the suction pipes. In addition, strum boxes are to be so constructed that they can be cleaned without disconnecting any joint of the suction pipes.

13.5.10 Dewatering Arrangements for Bulk Carriers, etc. *

For bulk carriers defined in **An1.2.1(1), Annex 1.1, Part 2-2, Part C**, bilge or ballast systems capable of being brought into operation from a readily accessible enclosed space, the location of which is accessible from the navigation bridge or continuously manned propulsion machinery control rooms without traversing exposed decks, are to be provided for draining and pumping those spaces specified in the following **(1)** and **(2)**.

(1) Ballast tanks forward of the collision bulkhead specified in **2.2.1.1, Part 1, Part C**

(2) Dry or void spaces other than chain lockers, in which any part extends forward of the foremost cargo hold and a volume that exceeds 0.1 % of the ship's maximum displacement volume

13.6 Air Pipes

13.6.1 General

1 All tanks, cofferdams and similar spaces are to be provided with air pipes having sufficient sectional areas to permit easy venting from any part of the tank, cofferdam and similar spaces.

2 Tanks having top plates not less than 7 *meters* either in length or in width are to be provided with two or more pipes arranged a suitable distance apart. However, tanks having inclined top plates may be provided with one air pipe located at the highest part of the top plate.

3 For tanks requiring more than one air pipe, overflow pipes which comply with requirement **13.7.2** may be used in lieu of air pipes as long as proper air flow from the tank to the atmosphere is ensured; all tanks, however, are to be provided with at least one air pipe.

4 In cases where tanks or cofferdams are of a complicated profile, special consideration is to be given to the number and positions of all air pipes.

5 Air pipes are to be arranged to be self-draining.

6 Vent pipes for fuel oil service, settling and lubrication oil tanks are to be located and arranged so that, in cases where such pipes break, there is no direct risk of any ingress of seawater or rainwater.

13.6.2 Open Ends of Air Pipes*

1 The position of the open ends of air pipes are to be in accordance with the following requirements **(1)** to **(4)** depending on the type and purpose of tanks.

(1) Air pipes to the following tanks and cofferdams are to be led above the bulkhead deck.

  (a) Double bottom tanks

  (b) Deep tanks

  (c) Tanks which allow for ingress of sea water

  (d) Cofferdams

(2) Air pipes to the following tanks and cofferdams are to be led to the weather part.

  (a) Fuel oil tanks and thermal oil tanks

  (b) Cargo oil tanks

  (c) Heated lubricating oil tanks and hydraulic oil tanks

  (d) Tanks which hold liquids and are filled by pumps, (only for tanks which are situated outside machinery spaces and not provided with overflow pipes)

  (e) Cofferdams adjacent to fuel oil tanks and cargo oil tanks.

(3) Air pipes to tanks, which hold liquids and are filled by pumps, are to be led to a safe position where the equipment cannot suffer any damage from any overflowing which may occur when the tank is being filled with a liquid.

(4) Air pipes to tanks carrying combustible or flammable liquids are to be led to a safe position where there is no possibility of fire caused by any oil or gas leaking from the openings when the tank is being filled.

2 All openings of air pipes leading above the weather deck are to be provided with automatic closing devices.

3 The open ends of air pipes to fuel oil and cargo oil tanks are to be provided with a flame arresting wire gauze of corrosion resistant materials that is easy to clean and detach as well as have a clear area through the mesh of not less than the required sectional area of the air pipe.

13.6.3 Size of Air Pipes*

Sizes of air pipes are to be as follows:

(1) The total sectional area of air pipes to tanks, which hold liquids and are filled by pumps, is not to be less than 1.25 *times* the total sectional area of the filling pipes. In cases where the tank is provided with an overflow pipe specified in **13.7**, the total sectional area may include that of air pipes to tanks to which overflow pipes are connected. The internal diameter of the air pipes is not to be less than 50 *mm*.

(2) Provisions are to be made for relieving vacuum when the tanks are being pumped out.

(3) The internal diameter of air pipes to cofferdams, tanks or similar spaces which form part of ship's structure is not to be less than 50 *mm*.

13.6.4 Height of Air Pipes*

In cases where air pipes extend above the freeboard deck or the superstructure deck, all exposed parts of the pipes are to be of substantial construction; the height from the upper surface of the deck to the point where water may have access below is to be at least:

- 760 *mm* on the freeboard deck, and
- 450 *mm* on the superstructure deck

In cases where these heights may interfere with the working of the ship, the height may be reduced to values approved by the Society, provided that the Society is satisfied that the closing arrangement and other circumstances justify this lower height.

13.6.5 Additional Requirements for Air Pipes Fitted on Exposed Fore Decks*

For ships with a L_C, specified in **1.4.3.1-1, Part 1, Part C**, of 80 *m* or more and where the height of the exposed deck in way of the item is less than 0.1 L_C or 22 *m* above the designed maximum load line, whichever is the lesser, all air pipes located on the exposed deck over the forward 0.25 L_C are to be of sufficient strength to resist green sea force.

13.7 Overflow Pipes

13.7.1 General

1 In cases where tanks which hold liquids and are filled by pumps, fall under either one of the following categories, overflow pipes are to be provided:

(1) In case where the sectional area of the air pipes does not comply with the requirements in **13.6.3(1)**;

(2) In cases where there is any opening below the open ends of air pipes fitted to the tanks; and

(3) Fuel oil settling tanks and fuel oil service tanks.

2 Overflow pipes other than those to tanks for fuel oil, lubricating oil and other flammable oils are to be led to the open air, or to positions where any overflow can be properly disposed of.

3 Overflow pipes are to be arranged to be self-draining.

4 In addition to **13.7**, overflow pipes for tanks for fuel oil, lubricating oil and other flammable oils are to comply with the requirements in **4.2.2(4)**, **Part R**.

13.7.2 Sizes of Overflow Pipes

1 The aggregated sectional area of overflow pipes which come under **13.7.1-1** is to be not less than 1.25 times the aggregated sectional area of the filling pipes.

2 The internal diameter for overflow pipes is not to be less than 50*mm*.

13.7.3 Overflow Pipes to Fuel Oil, Lubricating Oil and Other Flammable Oil tanks

1 Overflow pipes are to be fed into an overflow tank of adequate capacity or into a storage tank having sufficient space reserved for overflow purposes.

2 Overflow pipes are to be provided with sight glasses at readily visible positions on the vertical pipes, except in cases where an alarm device to give warning, when the oil level rises to a predetermined point in the tanks, is installed.

13.7.4 Preventive Means of Counter-flow of Overflow

1 In cases where overflow pipes to deep tanks which are used alternately to carry fuel oil, cargo oil, ballast water, general cargo, etc. are connected to an overflow main common to other tanks, arrangements are to be made to prevent any liquid, gases, etc. from other tanks from leaking into the deep tank carrying general cargo, and to prevent any liquid of different quality from leaking into those other tanks from the deep tank carrying the liquid.

2 Adequate means are to be provided for overflow pipes so that in the event of any one of the tanks being bilged, the other tanks cannot be flooded from the sea through the overflow pipes.

3 Overflow pipes discharging through the ship's sides are to extend above the load line, and are to be provided with non-return valves fitted on the ship's sides. In case where overflow pipes do not extend above the freeboard deck, additional effective means are to be provided to prevent the sea water from passing inboard.

13.8 Sounding Devices

13.8.1 General

1 All tanks, cofferdams and similar spaces are to be provided with a sounding pipe or a liquid level indicator. These devices are to be capable of checking the liquid levels in such spaces at readily accessible positions at all times.

2 Name plates are to be affixed to the upper ends of all sounding pipes.

3 In addition to **13.8**, sounding pipes for tanks for fuel oil, lubricating oil and other flammable oils are to comply with the requirements in **4.2.2(3)(e)**, **Part R**.

13.8.2 Upper Ends of Sounding Pipes

Sounding pipes are to be led to positions above the bulkhead deck which are at all times readily accessible, and are to be provided with an effective closing means at their upper ends. However, sounding pipes may be led to readily accessible positions from the platform of a machinery space provided that the closing means specified in **4.2.2(3)(e)**, **4.2.2(9)** and **4.2.3(2)**, **Part R** are provided according to the kinds of tanks. Sounding pipes for tanks other than those for flammable oil and cofferdams may be led to readily accessible positions from the platform of a machinery space provided that sluice valves, cocks or screw caps attached to the pipes by chain are provided.

13.8.3 Construction of Sounding Pipes*

1 Sounding pipes are to be as straight as practicable and if they are curved, the curvature is to be sufficiently large.

2 Striking plates of adequate size and sufficient thickness are to be fitted to the bottom plating under open ended sounding pipes to prevent any damage to the plating by the striking of the sounding rods. In cases where sounding pipes that have closed ends are employed, all closing plugs are to be of substantial construction.

3 The internal diameter of sounding pipes passing through a refrigerated chamber that has been cooled down to 0℃or below is not to be less than 65*mm* and is not to be less than 32*mm* for all other sounding pipes.

13.8.4 Construction of Liquid Level Indicators*

A liquid level indicator which is specified in **13.8.1** above is to be of a type that has been approved by the Society. However, when a liquid level indicator conforms to other standards approved by the Society or when it is provided with a certificate recognized by the Society, these requirements do not apply. The liquid level indicator for tanks for fuel oil, lubricating oil and other flammable oils are to comply with the requirements in **4.2, Part R of the Rules**.

13.8.5 Water Level Detection and Alarm Systems for Bulk Carriers, etc. *

1 For bulk carriers defined in **An1.2.1(1), Annex 1.1, Part 2-2, Part C**, water level detection and alarm systems are to be provided for giving audible and visual alarms in the navigation bridge, in accordance with the following **(1)** to **(4)**:

(1) In each cargo hold, the systems are to give alarms when the water level reaches the following **(a)** and **(b)** at the aft end of the cargo hold.

  (a) A height of 0.5 *m* above the inner bottom

  (b) A height not less than 15 % of the depth of the cargo hold but not more than 2.0 *m*

(2) In any ballast tank forward of the collision bulkhead specified in **2.2.1.1, Part 1, Part C**, the system is to give an alarm when the liquid in the tank reaches a level not exceeding 10 % of the tank capacity.

(3) In any dry or void space other than a chain locker, any part of which extends forward of the foremost cargo hold and the volume of which exceeds 0.1 % of the ship's maximum displacement volume, the system is to give an alarm at a water level of 0.1 *m* above the deck.

(4) The systems are to have constructions and functions deemed appropriate by the Society.

2 Alarms given by the water level detection and alarm systems specified in **-1** are to be capable of identifying the space where the water level reaches the alarm level and the water level specified in **-1(1)** at the navigation bridge. The above alarms are also to be capable of being easily distinguishable from alarms given by other installations in the navigation bridge.

3 The water level detection and alarm systems specified in **-1** for ballast tanks and cargo holds which have been designed to carry water ballast may be provided with override devices that are deemed appropriate by the Society.

4 Manuals documenting operating and maintenance procedures for the water level detection and alarm systems specified in **-1** are to be kept on board.

13.8.6 Water Level Detection and Alarm Systems for Single Hold Cargo Ships*

1 Cargo ships, other than bulk carriers defined in **An1.2.1(1), Annex 1.1 Part 2-2, Part C**, having a length (L_f) of less than 80 *m* and a single cargo hold below the freeboard deck or cargo holds below the freeboard deck which are not separated by at least one bulkhead made watertight up to that deck, are to be fitted in such space or spaces with water level detection and alarm systems in accordance with the following **(1)** to **(3)**:

(1) These water level detection and alarm systems are to give an audible and visual alarm at the navigation bridge when the water level above the inner bottom in the cargo hold reaches a height of not less than 0.3 *m*, and another when such level reaches not more than 15 % of the mean depth of the cargo hold.

(2) The systems are to be fitted at the aft end of the hold, or above its lowest part where the inner bottom is not parallel to the designed waterline. In cases where webs or partial watertight bulkheads are fitted above the inner bottom, the fitting of additional detectors may be required.

(3) The systems are to have constructions and functions deemed appropriate by the Society.

2 Alarms given by the water level detection and alarm systems specified in **-1** are to be capable of identifying the space where the water level reaches the alarm level and the water level specified in **-1(1)** at the navigation bridge. The above alarms are also to be capable of being easily distinguishable from alarms given by other installations in the navigation bridge.

3 Manuals documenting operating and maintenance procedures for the water level detection and alarm systems specified in **-1** are to be kept on board.

4 Notwithstanding the provisions of **-1**, water level detection and alarm systems need not to be fitted in ships complying with the requirements of **13.8.5**, or in ships having watertight side compartments on each side of the entire length of the cargo hold and that extend vertically at least from inner bottom to freeboard deck having a breadth that is deemed appropriate by the Society.

13.8.7 Water Level Detection and Alarm Systems for Multiple-Hold Cargo Ships*

1 For cargo ships having multiple holds (excluding the bulk carriers defined in **Annex 1.1 An1.2.1(1), Part 2-2, Part C** and tankers), water level detection and alarm systems are to be fitted in cargo holds intended for dry cargoes in order to give audible and visible alarms at the navigation bridge in accordance with the following **(1)** and **(2)**. However, water level detection and alarm systems are not required for cargo holds located entirely above the freeboard deck.

(1) Systems are to give alarms when water levels reach the following **(a)** and **(b)** at the aft ends of cargo holds. In cases where inner bottoms are not parallel to the designed waterline, systems are to be fitted above lowest parts of cargo holds.

(a) A height not less than 0.3 *m* above the inner bottom

(b) A height not less than 15% of the depth of the cargo hold but not more 2.0 *m*

(2) Systems are to have constructions and functions deemed appropriate by the Society.

2 Alarms given by the water level detection and alarm systems specified in **-1** above are to be capable of identifying the space where the water level reaches the alarm level and the water level specified in **-1(1)** above at the navigation bridge. The above alarms are also to be capable of being easily distinguishable from alarms given by other installations at the navigation bridge.

3 The water level detection and alarm systems specified in **-1** above for ballast tanks and cargo holds which have been designed to carry water ballast may be provided with override devices that are deemed appropriate by the Society.

4 Bilge alarm systems which are fitted in cargo hold bilge wells or other suitable locations may be used as the water level detection and alarm systems required by **-1(1)(a)** on the condition that they give audible and visible alarms in accordance with the following **(1)** to **(3)**.

(1) Systems are to give audible and visible alarms at the navigation bridge when water levels above the inner bottoms of cargo holds reach heights not less than 0.3 *m*. In cases where the bottoms of bilge wells are lower than the inner bottoms of cargo holds, alarms are to be given when water levels reach heights not less than 0.3 *m* above the bottoms of bilge wells.

(2) Alarms are to be capable of identifying the spaces where water levels reach alarm levels and being easily distinguishable from other alarms given by the systems specified in **-1** above.

(3) Systems are to have constructions and functions deemed appropriate by the Society.

5 Manuals documenting operating and maintenance procedures are to be kept on board for the water level detection and alarm systems specified in **-1** above and the bilge alarm systems used as water level detection and alarm systems in accordance with **-4** above.

13.9 Fuel Oil Systems

13.9.1 General*

1 Fuel oil systems in the machinery spaces where main propulsion machinery is installed and where a boiler is installed are to be such that easy maintenance and inspection can be performed. All valves or cocks are to be capable of being operated from above the platform.

2 Stop valves or cocks are to be fitted on both the suction and the delivery sides of fuel oil pumps.

3 Valves and pipe fittings with a design temperature above 60°C and a design pressure above 1*MPa* are to be suitable for use under a pressure of not less than 1.6*MPa*. Valves and pipe fittings used for fuel oil transfer piping lines, fuel oil suction piping lines and other low pressure fuel oil piping lines are to be suitable for use under a pressure of not less than 0.5*MPa* .

4 Union joints used for any connections of fuel oil injection pipes of reciprocating internal combustion engines or any pipes of burning systems of boilers are to be of rigid construction and to have metal contact capable of providing sufficient oil tightness.

5 Two fuel oil service tanks for each type of fuel used on board that is necessary for propulsion and vital systems or equivalent arrangements are to be provided.

6 The capacity of each fuel service tank required in **-5** is to be sufficient for at least 8 *hours* at maximum continuous rating of the main engine and normal operating load of the generators at sea.

7 In addition to **13.9**, fuel oil systems are to comply with the requirements in **4.2**, **Part R**.

13.9.2 Fuel Oil Filling Pipes

1 Fuel oil filling pipes from outboard are to be used exclusively for fuel. The open ends of these pipes are to be led above decks as far as possible and to be provided with rigid covers.

2 In cases where fuel oil filling pipes are not fitted on or near the top of the fuel oil tanks, non-return valves are to be fitted close to tanks; or, valves or cocks able to be closed by remote control as specified in **4.2.2(3)(d)**, **Part R** are to be provided.

3 Notwithstanding the requirements in **-1**, in cases where fuel oil filling pipes are connected to suction pipes, stop valves are to be provided on the filling pipes. In addition, stop valves are to be provided in cases where the tanks are situated on a higher position than the double bottom and in cases where there is the fear that fuel oil may pass to other fuel oil tanks through the filling pipes thereto or of any overflow from the openings of sounding pipes, etc.

13.9.3 Fuel Oil Transfer Pumps

In ships where power pumps are used for pumping fuel into settling and service tanks, at least two independently powered fuel oil transfer pumps are to be provided; and, these pumps are to be connected and ready for use. In cases where any suitable independently powered driven fuel oil pump for other purposes is available for use as a fuel oil transfer pump, such a pump may be used as a fuel oil transfer pump.

13.9.4 Drip Trays and Drainage Systems*

1 Metal drip trays of a sufficient depth are to be provided under all equipment that uses or handles fuel oil such as reciprocating internal combustion engines (excluding main propulsion machinery of ships other than electric propulsion ships), burners, fuel oil pumps, fuel oil heaters, fuel oil coolers and fuel oil filters as well as fuel oil tanks such as fuel oil settling and service tanks. In cases where it is not practicable to provide metal drip trays, coamings are to be provided to hold any oil spillage.

2 Fuel oil settling tanks and service tanks are to be provided with drain valves or cocks for draining water from the bottom of the tanks.

3 Drain valves or cocks fitted to fuel oil tanks are to be of a self-closing type.

4 Drainage arrangements are to comply with the following requirements:

(1) Oil in drip trays or in the coamings prescribed in **-1**, **-2** as well as any drainage from drain valves or cocks fitted to fuel oil tanks are to be led into fuel oil drain tanks, or some other suitable arrangement.

(2) The fuel oil drain tanks prescribed in **(1)** are not to be part of an overflow system.

(3) Suitable means are to be provided for the disposal of any fuel oil drainage stored in the fuel oil drain tanks prescribed in **(1)**.

13.9.5 Fuel Oil Heaters*

1 In cases where heaters are provided for fuel oil systems, they are to be equipped with temperature controllers as well as high temperature alarm devices or low flow alarm devices, unless where the oils would not be heated to a temperature that is 10°C or less below the flash point of the fuel oil.

2 Double bottom tanks and deep tanks are not to be provided with electric heaters unless approved by the Society.

3 Electric heaters for heating fuel oil are to comply with the following requirements:

(1) Heaters are to be provided with automatic temperature controlling devices.

(2) Safety switches with independent temperature sensors are to be provided. These safety switches are to cut off the electrical power supply in order to prevent the surface temperature of heating elements from rising to 220°C or above; and, they are to be provided with manual reset devices.

(3) Electric heaters are to be adequately protected against any mechanical damage during times of tank cleaning.

13.9.6 Fuel Oil Systems for Reciprocating Internal Combustion Engines*

1 Number and capacity of fuel oil supply pumps for the main propulsion machinery

(1) The main propulsion machinery is to be provided with one main fuel oil supply pump of sufficient capacity to maintain the supply of the fuel oil at the maximum continuous output of the machinery as well as one stand-by fuel oil supply pump of sufficient capacity to supply fuel under normal service conditions. These pumps are to be connected and ready for use.

(2) In cases where two or more main propulsion machinery is provided and where each of them has a built-in main fuel oil supply pump as well as in cases where it is possible to obtain navigable speed even if one of them is out of use, stand-by fuel oil supply pumps may be dispensed with on the condition that one complete spare pump is carried on board.

2 Number and capacity of fuel oil supply pumps for reciprocating internal combustion engines driving auxiliary machinery and electrical generators

(1) Reciprocating internal combustion engines for driving electrical generators and auxiliary machinery for which duplication is required are to be provided with main and stand-by fuel oil supply pumps of sufficient capacity to maintain the supply of oil at the maximum continuous output of the engine. These pumps are to be connected and ready for use.

(2) In cases where each engine prescribed in **(1)** is provided with an exclusive main fuel oil supply pump, the stand-by fuel oil supply pump may be omitted.

3 Driving system of stand-by fuel oil supply pumps and use of other pumps

(1) Stand-by fuel oil supply pumps are to be driven by an independent power source.

(2) In cases where any fuel oil pump driven by an independent power source and intended for other purposes is available for use as a stand-by fuel oil supply pump; this pump may be used as a stand by fuel oil supply pump.

4 Fuel oil filters

(1) Fuel oil filters are to be provided for fuel oil supply piping lines of reciprocating internal combustion engines.

(2) Fuel oil filters for reciprocating internal combustion engines that are used as main propulsion machinery are to be capable of being cleaned without stopping the supply of filtered oil. The filters are to be provided with valves or cocks for depressurizing before being opened.

5 Fuel oil heating devices and fuel oil purifying devices

In cases where low grade oil is used for fuel oil, suitable fuel oil heating devices and fuel oil purifying devices are to be provided.

13.9.7 Burning Systems for Boilers*

1 Burning systems for main boilers

(1) In cases where the main boiler is provided with a combustion system of pressurized fuel injection type, at least two units of burning pumps and fuel oil heaters are to be provided respectively with each unit being capable of supplying a sufficient amount of oil to generate steam at the maximum evaporation rate of the boiler even in the case of failure of one unit. These pumps are to be connected and ready for use.

(2) Filters are to be provided for the suction and delivery sides of fuel injection pumps. These filters are to be capable of being cleaned without stopping the supply of filtered oil.

(3) The fuel oil filters specified in the above **(2)** are to be provided with valves or cocks for depressurizing before being opened.

2 Burning systems for auxiliary boilers

(1) With respect to essential auxiliary boilers and all other boilers that supply steam for fuel oil heating necessary for the operation

of the main propulsion machinery or cargo heating that is required continuously, burning systems are to be provided in accordance with the requirements in -1. However, where alternative means are available to ensure normal navigation and cargo heating even in cases where the burning system is out of operation, only one unit of burning system will be accepted.

(2) In cases where fuel oil is supplied to the burners by gravity, fuel oil filters capable of being cleaned without stopping the supply of filtered oil are to be provided.

3 Prevention of mixing of oil into steam pipes and air pipes

In cases where the removal of residual fuel oil in burners is conducted by means of steam or air, measures are to be taken to prevent the mixing of oil with any steam or air.

13.10 Lubricating Oil Systems and Hydraulic Oil Systems

13.10.1 General*

1 The location, drip trays, drainage arrangements and heaters of lubricating oil systems are to comply with the requirements in **13.9.1-1**, **13.9.4-1** and **-4**, and **13.9.5** respectively (in these cases the term "fuel oil" is to be read as "lubricating oil").

2 The location, drip trays and drainage arrangements of hydraulic oil systems are to comply with the requirements in **13.9.1-1**, **13.9.4-1** and **-4** (in these cases the term "fuel oil" is to be read as "hydraulic oil").

3 In addition to **13.10**, lubricating oil systems and hydraulic oil systems are to comply with the requirements in **4.2.3**, **Part R** and **4.2.4**, **Part R** respectively.

13.10.2 Lubricating Oil Pumps

1 Number and capacity of lubricating oil pumps for main propulsion machinery, propulsion shafting and power transmission systems

(1) Main propulsion machinery, propulsion shafting and their power transmission systems are to be provided with one main lubricating oil pump of sufficient capacity to maintain the supply of oil at the maximum continuous output of the machinery and one stand-by lubricating oil pump of sufficient capacity to supply oil under normal navigating conditions. These pumps are to be connected and ready for use.

(2) In cases where two or more main propulsion machinery as well as propulsion shafting and their respective power transmission systems are each provided with a built-in main lubricating oil pump; and, in cases where it is possible to obtain navigable speed even if one of them is out of use, stand-by lubricating oil pumps may be dispensed with on the condition that one complete spare pump is carried on board.

2 Number and capacity of lubricating oil pumps for auxiliary machinery, electrical generators and their prime movers

(1) Electrical generators and auxiliary machinery for which duplication is required and their prime movers are to be provided with main and stand-by lubricating oil pumps of sufficient capacity to maintain the supply of oil at the maximum continuous output of the machinery. These pumps are to be connected and ready for use.

(2) In cases where each system prescribed in **(1)** is provided with an exclusive main lubricating oil pump, the stand-by lubricating oil pump may be omitted.

3 Driving systems of stand-by lubricating oil pumps and use of other pumps

(1) Stand-by lubricating oil pumps are to be driven by an independent power source.

(2) In cases where any lubricating oil pump driven by an independent power source and intended for other purposes is available for use, this pump may be used as a stand-by lubricating oil pump.

4 Number and capacity of lubricating oil pumps for waterjet propulsion systems and azimuth thrusters

Lubricating oil pumps for waterjet propulsion systems and azimuth thrusters are to comply with the requirements in **-1**. In this case the term "main propulsion machinery, propulsion shafting and power transmission systems" is to be read as "waterjet propulsion systems" or "azimuth thrusters" respectively.

13.10.3 Stop Valves between Engine and Sump Tank

For ships of 100 *meters* or longer in length, in cases where a double bottom is used as a lubricating oil sump tank, a stop valve which can be easily operated from the engine room floor or a suitable counterflow prevention device is to be provided.

13.10.4 Lubricating Oil Filters*

1 In cases where a forced lubrication system (including gravity tanks) is adopted for the lubrication of machinery installations, lubricating oil filters are to be provided. Additionally, it is recommended to use strainers with magnets for waterjet propulsion systems and azimuth thrusters.

2 Filters used for the lubricating oil systems of the main propulsion machinery, power transmissions of propulsion shafting, controllable pitch propeller systems, waterjet propulsion systems, and azimuth thrusters are to be capable of being cleaned without stopping the supply of filtered oil.

3 Lubricating oil filters specified in the above **-2** are to be provided with valves or cocks for depressurizing before being opened.

13.10.5 Lubricating Oil Purifying Devices

Lubricating oil systems are to be provided with lubricating oil purifying systems such as lubricating oil purifiers or filters in lieu of purifiers.

13.11 Thermal Oil Systems

13.11.1 General

The location of thermal oil systems and the valves fitted to the pumps of such systems are to comply with the requirements in **13.9.1-1** and **-2**. Any filling pipes from outside the ship are to comply with the requirements in **13.9.2-2**. Drip trays and drainage systems are to comply with the requirements in **13.9.4-1** and **-4**. In these cases the term "fuel oil" is to be read as "thermal oil". In addition to **13.11**, these systems are to comply with the requirements in **4.2.4**, **Part R**.

13.11.2 Thermal Oil Systems

Thermal oil systems are to comply with the following requirements:

(1) Expansion tanks are to be provided with liquid level indicators.

(2) Circulating pumps are to be provided with a pressure measuring device at a suitable position on both the delivery and suction sides.

(3) The inlet and outlet valves on thermal oil heaters are to be controllable from outside the compartment where they are installed, unless an arrangement for the quick drainage by gravity of any thermal oil contained in the system into a collecting tank is made.

13.11.3 Pumps for Thermal Oil Heaters*

Thermal oil heaters of important use are to be provided with two thermal oil circulating pumps and two fuel injection pumps. However, only one fuel injection pump may be acceptable, in cases where alternative means are available to ensure normal navigation and cargo heating in case of pump failure.

(1) Thermal oil circulating pumps

(2) Fuel injection pumps

13.11.4 Heating of Liquid Cargo with Flash Points below 60℃*

The heating of liquid cargo with flash points below 60℃ is to be arranged by means of a separate secondary system, located completely within the cargo area unless in those cases deemed appropriate by the Society.

13.12 Cooling Systems

13.12.1 Cooling Pumps*

1 Number and capacity of cooling pumps for main propulsion machinery.

(1) Main propulsion machinery is to be provided with a main cooling pump of sufficient capacity to maintain the supply of water (oil) at the maximum continuous output of the machinery as well as a stand-by cooling pump of sufficient capacity to supply cooling water (oil) under the normal navigating conditions. However, the capacity of the stand-by circulating pumps of ships in which steam turbines are used as main propulsion machinery is considered by the Society on a case-by-case basis. These pumps are to be connected and ready for use.

(2) In ships with steam turbines used as main propulsion machinery, an adequately installed scoop arrangement may be used as the main cooling water pump. In such cases, the main condenser is to be so arranged as to be sufficiently cooled by other cooling systems while the ship runs at low speed, in addition to any cooling system performed by the stand-by cooling water pumps specified in **(1)** above.

(3) In cases where two or more main propulsion machinery are provided, each of which has a built-in main cooling pump as well as cases where it is possible to obtain navigable speed in case of the failure of one of the main propulsion machinery, stand-by cooling pumps may be dispensed with on the condition that one complete spare pump is carried on board.

2 Number and capacity of cooling pumps for auxiliaries, electrical generators and their prime movers.

(1) Electrical generators and auxiliaries for which duplication is required and their prime movers are to be provided with main and stand-by cooling pumps of sufficient capacity to maintain the supply of water (oil) at the maximum continuous output of the machinery. These pumps are to be connected and ready for use.

(2) In cases where each of prime mover specified in **(1)** above is provided with an exclusive main cooling pump, the stand-by cooling pump may be omitted.

3 Drive system of stand by cooling pumps and use of other pumps.

(1) Stand-by cooling pumps are to be driven by an independent power source.

(2) In cases where a suitable pump driven by an independent power source and intended for other purposes is available for use, this pump may be used as a stand-by cooling pump.

13.12.2 Suction of Sea Water

Arrangements are to be provided to introduce cooling sea water from sea suction valves fitted on two or more sea chests or sea suction inlets.

13.12.3 Cooling Systems for Reciprocating Internal Combustion Engines*

In cases where sea water is used for the direct cooling of the propulsion machinery, reciprocating internal combustion engines driving electrical generators, or any auxiliary machinery for which duplication is required, strainers, which are arranged so as they are capable of being cleaned without stopping the supply of filtered cooling water to the respective engines, are to be provided between the sea suction valve and the cooling sea water pump.

13.13 Pneumatic Piping Systems*

13.13.1 Arrangement of Air Compressors and Pressure Relief Systems

1 Air compressors are to be so arranged that any mixing between oil and incoming air is minimized as much as possible.

2 Each air compressor is to be provided with a relief valve to prevent the pressure from rising more than 10% above the maximum working pressure of its cylinders.

3 In cases where water jackets of air coolers might be subject to dangerous level of excessive pressure due to any leakage of compressed air into them, suitable pressure relief arrangements are to be provided for these water jackets.

13.13.2 Relief Devices and Other Fittings for Air Tanks

Relief devices and other fittings for air tanks are to comply with the requirements in **10.8**.

13.13.3 Number and Total Capacity of Air Compressors

1 In cases where the main propulsion machinery is designed for starting by compressed air, two or more starting air compressors are to be provided and arranged so as to be able to charge each air reservoir. However, in cases where cylinders are provided with air charging valves, these charging valves will be considered to be equivalent to any air compressors driven by the main propulsion machinery.

2 At least one of the air compressors specified in **-1** above is to be independently driven by a prime mover that is not the main propulsion machinery. In addition, either the capacity of one of the independently driven compressor or the combined capacity of all independently driven compressors is to be not less than 50 % of the total capacity specified in **-3**.

3 The total capacity of air compressors is to be sufficient to supply air into the air reservoirs from atmospheric pressure to the pressure required for the consecutive starts prescribed in **2.5.3-2** or **4.4.3-2**, corresponding to the type of prime mover, within one *hour*. The capacity is to be approximately equally divided between the number of starting air compressors (excluding emergency compressors installed to satisfy **1.3.1-5**) fitted for main propulsion machinery.

13.13.4 Emergency Air Compressors

1 In cases where prime movers driving air compressors specified in **13.13.3** are arranged for air starting, an independently power driven emergency air compressor is to be provided.

2 Prime movers driving emergency air compressors are to be capable of starting without compressed air.

3 The capacity of emergency air compressors is to be sufficient to start the prime movers of the air compressor prescribed in **13.13.3**. For this purpose, a small air reservoir for such an emergency air compressor may be provided.

13.13.5 Compressed Air Piping

1 Drainage systems are to be provided for compressed air piping to remove any drainage remaining inside the pipes.

2 All discharge pipes for starting air reservoirs are to be laid directly from starting air compressor.

3 Starting air pipes from the air reservoirs to main propulsion machinery or auxiliary engines are to be entirely separate from the compressor discharge system prescribed in **-2**.

13.13.6 Pneumatic Piping Systems for Essential Services

The following **(1)** and **(2)** requirements are to be applied to the supply of compressed air required by essential services on board ships other than the supply of compressed air for engine starting.

(1) The arrangements for the supply of compressed air to essential services are to ensure that sufficient compressed air to satisfy the total demand of the essential services is available at all times during normal operation, during maintenance, and in the event of a failure of the compressed air system.

(2) Where compressed air is supplied from the engine starting air system, either continuously in normal operation, or periodically during maintenance or in the event of a failure of the compressed air system, the required compressed air demand is not to reduce the capacity and availability of the engine starting air required by **2.5.3-2** and **4.4.3-2**.

13.14 Steam Piping Systems and Condensate Systems

13.14.1 Drainage Arrangements

Drainage arrangements are to be installed at suitable locations in steam pipes.

13.14.2 Heating Coil for Oil

In cases where steam is used for heating fuel oil or lubricating oil, steam drain pipes are to be led to observation tanks or other oil detectors in a well-lit and accessible position in the machinery space.

13.14.3 Steam Pipes Passing through Cargo Holds

In principle, steam pipes are not to be led through cargo holds. However, in cases where it is impracticable to avoid such an arrangement, these pipes are to be insulated and protected by steel plates and all of the joints are to be welded.

13.14.4 Condensate Systems*

1 Main condensers are to be provided with at least two independently power driven condensate pumps which have the capacity to deal with maximum designed rate of condensate from main condenser as well as at least to devices to maintain vacuum in the condensers. Devices to maintain vacuum may be omitted in cases where they are considered unnecessary by the Society after taking into account the type of the main condenser.

2 Suitable measures deemed appropriate by the Society to be taken to prevent the internal pressure of the condensers for the condensation systems of steam turbines for driving devices such as cargo oil pumps from exceeding their design pressure in cases of cooling system failure.

13.15 Feed Water Systems for Boilers

13.15.1 Feed Water Systems for Main Boilers

1 Two feed water systems are to be provided for main boilers, each including a stop valve, a non-return valve specified in **9.9.5-1** and a feed pump. These feed water systems are to be capable of supplying feed water to the boiler in cases where any one of the systems being out of action.

2 Main boilers are to be provided with two or more feed water pumps which can supply feed water sufficient for maximum evaporation with any one of the pumps being out of action. These feed pumps are to be connected and ready for use.

3 Feed water pumps prescribed in **-2** are to be driven by independent prime movers.

4 Feed water systems are to be provided with feed water regulators capable of automatically controlling the feed water rate.

5 Feed pumps are not to be used for any purpose other than to feed the boilers

13.15.2 Feed Water Systems for Auxiliary Boilers

Every auxiliary boiler (including steam generating systems, hereinafter in 13.15.2) which provides services essential for the safety of the ship, or which could be rendered dangerous by the failure of its feed water supply, is to be provided with two separate feed water systems in accordance with 13.15.1, noting that a single penetration of the steam drum is acceptable.

13.15.3 Distilling Plants

In ships using distilled water as feed water, at least one distilling plant with a sufficient capacity is to be provided.

13.15.4 Pipes Passing through Tanks

Boiler feed water pipes are not to be led through tanks which contain oil or fuel, and oil or fuel pipes are not to be led through boiler feed water tanks.

13.16 Exhaust Gas Piping Arrangements

13.16.1 Exhaust Gas Pipes from Reciprocating Internal Combustion Engines and Gas Turbines

1 In principle, the exhaust gas pipes from two or more reciprocating internal combustion engines are not to be connected together except in the following **(1)** and **(2)** cases. In addition, the exhaust gas pipes from reciprocating internal combustion engines and gas turbines as well as the exhaust gas pipes from two or more gas turbines are, in principle, not to be connected together.

(1) In cases where exhaust gas pipes of two or more reciprocating internal combustion engines are connected to common silencers and effective means are provided to prevent any exhaust gas from returning into the cylinders of non-operating reciprocating internal combustion engines.

(2) In cases where exhaust gas pipes of two or more reciprocating internal combustion engines are connected to common exhaust gas cleaning systems which comply with **Chapter 22**.

2 Exhaust gas piping lines are to be arranged so that water does not enter the cylinders of reciprocating internal combustion engines or gas turbines. In particular, exhaust gas piping lines that are led overboard near the water line are to be so arranged to prevent water from being siphoned into the line.

3 Boiler uptakes and exhaust piping lines from reciprocating internal combustion engines are not to be connected together except in the following **(1)** and **(2)** cases. In addition, boiler uptakes and the exhaust gas pipes from gas turbines are not to be connected together except in case **(1)**.

(1) In cases where boilers or gas turbines are arranged to utilize waste heat from reciprocating internal combustion engines.

(2) In cases where boiler uptakes and exhaust piping lines from reciprocating internal combustion engines are connected to common exhaust gas cleaning systems which comply with **Chapter 22**.

13.16.2 Exhaust Gas Pipes from Boilers

In cases where dampers are installed in the funnels or uptakes of boilers, their degree of opening is not to be reduced to 2/3 or less of the flue area when closed. They are to be capable of locking in any open position and the degree of opening is to be clearly indicated.

13.16.3 Exhaust Gas Pipes from Incinerators

In cases where incinerator exhaust gas pipes are of a shape (e.g., u-shaped, etc.) which is susceptible to the accumulation of unburnt matter, a cleaning hole is to be provided for maintenance at the parts where said unburnt matter is expected to easily accumulate.

13.17 Tests

13.17.1 Shop Tests

Auxiliaries and piping are to be tested in accordance with the requirements in 12.6 after manufacture.

13.17.2 Tests On Board

1 Auxiliaries (excluding auxiliary machinery for specific use etc.) are to be subjected to running tests after installed on board. However, in the case of machinery having passed the running tests specified in 12.6.1-9, the test methods on board may be suitably modified at the discretion of the Society.

2 For piping systems for which welding between pipes or between pipes and pipe fittings is carried out on board the ship, all joints welded on board the ship are to be subjected to the non-destructive testing specified in 11.6.

3 In general, all the piping systems are, after assembly on board, to be checked for leakage under operational conditions and, if necessary, using special techniques other than hydrostatic testing. In particular, fuel oil piping systems, thermal oil piping systems and heating coils in tanks are, after installed on board, to be subjected to a leak test at a pressure of 1.5 *times* the design pressure or 0.4 *MPa*, whichever is greater.

4 Pneumatic leak testing may be carried out on water sensitive systems, in lieu of hydrostatic testing. In certain circumstances, a combined hydrostatic–pneumatic strength test may also be applied, where the system is partially filled with water and the free space above is pressurized with a test gas (typically air or nitrogen). When pneumatic tests cannot be avoided, the safety precautions in *IACS* Rec. 140, Part F, are to be observed.

Chapter 14 PIPING SYSTEMS FOR TANKERS

14.1 General

14.1.1 Scope*

1 The requirements of this Chapter apply to the piping systems for tankers which have all of the following features. The piping systems for other types of tankers will be considered by the Society on a case by case basis. The requirements given in this Chapter are to especially apply in lieu of the requirements given in **Chapters 12** and **13**.

(1) Crude oil, petroleum products having vapour pressures (absolute pressures) less than 0.28 *MPa* at 37.8℃ or other similar liquid cargo are carried.

(2) Machinery spaces and cargo oil tanks (including slop tanks; hereinafter, this definition applies throughout this Chapter) are arranged in accordance with the requirements given in **4.5.1-1**, **Part R**.

(3) Cargo loaded by land facilities and unloaded by cargo oil pumps onboard ships.

2 The piping systems for ships carrying dangerous chemicals in bulk are to comply with the requirements given in **Part S.** However, those items not specified in **Part S** are to comply with the requirements of this Chapter. In such cases, the term “cargo oil” is to be read as “cargo”.

14.1.2 Drawings and Data

Drawings and data to be submitted for approval are generally as follows:

(1) Piping diagrams of cargo oil pipes and instrumentation (with materials, dimensions, design pressures of pipes, valves, etc. and arrangements of devices to prevent any passage of flame).

(2) Control system diagrams (including safety and alarm systems) of integrated cargo and ballast systems driven by electrohydraulic power.

(3) Other drawings and data considered necessary by the Society.

14.2 Cargo Oil Pumps, Cargo Oil Piping Systems, Piping in Cargo Oil Tanks, etc.

14.2.1 Cargo Oil Pumps

1 Cargo oil pumps are to comply with the following requirements:

(1) Pumps are to be designed to minimize the risk of sparking and oil leakage at seals.

(2) Stop valves are to be provided on the delivery sides of pumps. However, such stop valves may be omitted, provided that cargo oil pipes on the delivery sides of pumps are provided with stop valves in proper positions.

(3) In cases where relief valves are provided on the delivery sides of pumps, arrangements are to be such that all escaped oil is led to the suction sides of pumps.

(4) Pressure measuring devices are to be fitted on the delivery sides of pumps. In cases where pumps are driven by prime movers which are installed in spaces other than pump rooms, additional pressure measuring devices are to be fitted at suitable positions visible from control positions.

(5) The requirements given in **4.5.10(1)**, **Part R**.

2 In cases where prime movers, other than steam engines or hydraulic motors, for driving cargo oil pumps are installed in cargo oil pump rooms, information regarding the construction of these prime movers and their driving systems are to be submitted for Society approval.

3 In cases where deep well pumps, submerged pumps, etc. are installed, information regarding the construction of these pumps and their driving systems are to be submitted for Society approval.

4 In general, cargo oil pumps are not to be used for purposes other than the transferring of cargo oil or ballast in cargo oil tanks, the transferring of tank cleaning water for cargo oil tanks, the discharge of bilge as stipulated in **14.3.1-2** or the discharge of ballast as specified in **14.3.2-2**.

14.2.2 Arrangement of Cargo Oil Piping Systems*

1 Cargo oil pipes are classified as Group III pipes, except in cases where considered necessary by the Society.

2 Cargo oil tanks are to be provided with cargo oil suction pipes arranged so that cargo unloading can be carried out in cases where one of the cargo oil pumps is out of use.

3 Cargo oil pipes are to be arranged so as to be capable of loading cargo oil to cargo oil tanks without passing through cargo oil pumps.

In cases where loading pipes are led directly into tanks from above deck, the opening ends of these pipes are to be led into the lower parts of tanks as far as practicable in order to prevent any accidents caused by static electricity.

4 In cases where sea suction pipes for ballast purposes are connected to cargo oil pipes, stop valves are to be provided between sea suction valves and cargo piping.

5 Slip-on joints used in cargo oil pipes are to comply with the requirements specified in **12.3.3**.

6 Sea suction pipes and the discharge pipes for permanent ballast tanks are not to be connected to sea suction pipes and the discharge pipes for cargo oil tanks.

7 Earthing and bonding of cargo tanks, piping systems, etc. for the control of static electricity are to comply with following requirements:

(1) The hazard of an incentive discharge due to the build-up of static electricity resulting from the flow of liquids/gases/vapours can be avoided if the resistance between the cargo tanks, piping systems, etc. and the hull of the ship is not greater than 1 $M\Omega$.

(2) This value of resistance will be readily achieved without the use of bonding straps where cargo tanks, piping systems, etc. are directly or via their supports, either welded or bolted to the hull of the ship.

(3) Bonding straps are required for cargo tanks, piping systems, etc. which are not permanently connected to the hull of the ship, e.g.

(a) Independent cargo tanks;

(b) Cargo tanks/piping systems which are electrically separated from the hull of the ship;

(c) Pipe connections arranged for the removal of spool pieces; and

(d) Wafer-style valves with non-conductive (e.g PTFE) gaskets or seals.

(4) Where bonding straps are required, they are to be:

(a) Clearly visible so that any shortcomings can be clearly detected;

(b) Designed and sited so that they are protected against mechanical damage and that they are not affected by high resistivity contamination e.g. corrosive products or paint; and

(c) Easy to install and replace.

14.2.3 Alternative Use of Tanks*

In cases where cargo oil tanks are designed so that they can also be used as ballast tanks or fuel oil tanks, such tanks are to be provided with any devices required by the Society, and approved drawings or documents including detailed operating manuals for these alternative uses are to be provided on board the ship.

14.2.4 Separation of Cargo Oil Pumps and Cargo Oil Pipes*

1 Cargo oil pipes are to be completely separated from other pipes, except in cases where permitted in **14.2.2**, **14.3.1** and **14.3.2**.

2 Cargo oil pipes are not to be led through fuel oil tanks, engine rooms, accommodation spaces and any spaces in cases where sources of vapour ignition are normally present. In addition, these pipes are not to be led to spaces forward of collision bulkheads or aft of the front bulkheads of engine rooms.

3 Cargo oil pipes on weather decks are to be arranged sufficiently apart from any accommodation spaces.

4 In cases where ships are equipped for bow and/or stern loading and the discharge of cargo oil outside cargo areas, the connections of all cargo lines leading to cargo hose connections therein are to be welded joints except in the case of valve connections and cargo lines are to be clearly identified and segregated by the following means of **(1)** or **(2)** situated in cargo areas. Open ends of cargo lines are to be provided with blank flanges at their bow and/or stern end connections.

(1) Two valves which can be secured in closed positions and provided that the efficiency of the segregation can be checked

(2) One valve together with another closing appliance providing equivalent standards of segregation such as removable spool pieces or spectacle flanges

5 Cargo oil pipes and similar pipes to cargo oil tanks are not to pass through ballast tanks. However, these pipes may pass through ballast tanks provided that the sections of these pipes in ballast tanks are short in length and the connections of these pipes are of welded joints or flanged joints which have no risk of leakage.

6 Notwithstanding the preceding -5, in the case of oil tankers other than double hull tankers, cargo oil pipes may pass through ballast tanks provided that the connections of these pipes are of welded joints or flanged joints which have no risk of leakage. Expansion bends only are permitted in these lines within ballast tanks.

14.2.5 Bulkhead Valves of Cargo Oil Piping Systems*

1 Cargo oil pipes passing through oiltight bulkheads between cargo oil tanks and pump rooms are to be provided with stop valves as close to bulkheads as practicable.

2 In cases where those valves prescribed in -1 above are located inside pump rooms, they are to be made of steel and to be capable of being closed at the positions of valves and from readily accessible positions outside compartments in which they are located. However, in the case of valves, operated at positions above decks, which are fitted on cargo oil branch pipes, any valves located inside pump rooms may be of cast iron without remote control devices.

3 In cases where those valves prescribed in -1 above are located inside tanks, they may be made of cast iron and need not be capable of being closed at the positions of valves. However, they are to be provided with remote control devices, and another valve is to be provided in pump rooms.

4 In cases where the valves are required to be remotely controlled according to those requirements given in -2 and -3 above, means are to be provided to show whether they are open or closed.

14.2.6 Valve Operation Rod Penetrating through Decks

Stuffing boxes are to be provided at positions in which operating rods from cargo valves pass through gastight or oiltight decks.

14.2.7 Piping in Cargo Oil Tanks*

1 Pipes other than cargo oil pipes, cargo oil heating pipes, ballast pipes of cargo tanks and pipes permitted in -2 to -4 are not to pass through cargo oil tanks, or to have any connections to these spaces.

2 Pipes used for the remote control of cargo oil piping systems as well as vapour discharge pipes, tank cleaning pipes and sounding devices of cargo oil tanks may be led to cargo oil tanks.

3 Scupper pipes, sanitary pipes, etc. may be led through cargo oil tanks subject to the Society approval.

4 Ballast pipes and other pipes, such as sounding and vent pipes to ballast tanks, are not to pass through cargo oil tanks. However, these pipes may pass through cargo oil tanks provided that the sections of these pipes in cargo oil tanks are short in length and the connections of these pipes are of welded joints or flanged joints which have no risk of leakage.

5 Notwithstanding the preceding -4, in the case of oil tankers other than double hull tankers, ballast pipes of ballast tanks adjacent to cargo oil tanks may pass through cargo oil tanks provided that the connections of these pipes are of welded joints or flanged joints which have no risk of leakage. Expansion bends only are permitted in these lines within cargo oil tanks.

14.2.8 Sounding Devices of Cargo Oil Tanks*

Suitable sounding devices approved by the Society are to be fitted onto all cargo oil tanks. These sounding devices are to be designed or arranged to prevent any outflow of flammable vapours into spaces such as engine rooms, accommodation spaces, etc. in cases where sources of vapour ignition are normally present.

14.2.9 Steam Pipes

1 Cargo oil heating steam supply and return pipes are not to penetrate cargo oil tank plating, other than at the tops of tanks, and main supply pipes are to be run above weather decks.

2 Isolating shut-off valves or cocks are to be provided at the inlet and outlet connections to the heating circuit(s) of each tank.

3 In order to detect any contaminated oil in steam drainage, cargo oil heating steam return pipes are to be led to observation tanks; or, other oil detectors installed in positions as far apart as possible from any hot surfaces such as boilers and ignition sources.

4 Steam temperatures in cargo areas are not to exceed 220°C.

5 In cargo oil pump rooms, drain pipes from steam or exhaust pipes or from the steam cylinders of pumps are to terminate well above bilge wells.

6 Branch connections of cleaning steam pipes of cargo oil tanks or other tanks to which cargo oil pipes are led are to be provided with screw-down non-return valves or two stop valves.

14.2.10 Thermal Oil Pipes

1 Thermal oil piping arrangements for cargo oil tanks are to comply with following requirements:

(1) All joints in cargo oil tanks are to be welded joints.

(2) Isolating shut-off valves or cocks are to be provided at the inlet and outlet connections to cargo oil tanks. In cases where thermal oil pipes penetrate oiltight bulkheads between cargo oil tanks and pump rooms, such shut-off valves or cocks may be installed as close to the bulkhead as practicable.

(3) Systems are to be arranged so that the pressure in coils is at least 3 *m* water head above the static heads of cargo in cases where circulating pumps are not operating.

(4) In the case of ships carrying oils having flashpoints below 60°C, the requirements given in **13.11.4** are also to be applied.

2 Thermal oil temperatures in cargo areas are not to exceed 220°C.

14.2.11 Integrated Cargo and Ballast Systems Driven by Electrohydraulic Power

Emergency stopping devices and control systems of integrated cargo and ballast systems driven by electrohydraulic power (hereinafter referred to as “integrated systems”) are to comply with the following requirements:

(1) Emergency stopping devices of integrated systems are to be independent from control systems. The failure of a single emergency stopping device or control system is not to render the integrated system inoperative.

(2) Manual emergency stops of the cargo pumps are to be arranged in a way that they are not to cause the stop of the hydraulic power source.

(3) Emergency stopping devices and control systems are to be provided with a backup power supply, which may be satisfied by a duplicate power supply from the main switch board. The failure of any power supply is to provide audible and visible alarms at each location where a control panel is fitted.

(4) Manual overriding or redundant arrangements are to be provided within any control systems made available for the operation of the integrated system in the event of the failure of any automatic or remote control system.

14.3 Piping Systems for Cargo Oil Pump Rooms, Cofferdams and Tanks adjacent to Cargo Oil Tanks

14.3.1 Bilge Piping Systems, etc. for Cargo Oil Pump Rooms and Cofferdams adjacent to Cargo Oil Tanks*

1 Bilge piping systems consisting of a power driven pump or eductor are to be provided to discharge bilge in cargo oil pump rooms and in cofferdams adjacent to cargo oil tanks. The bilge in these spaces is not to be led to the engine room.

2 Cargo oil pumps may be used for bilge drainage purposes specified in **-1** provided that each bilge suction is fitted with a screw-down non-return valve, and a stop valve or cock is fitted on the suction side of the pump. In addition, a stop valve is to be fitted between the cargo oil pipe and the overboard discharge valve.

3 Bilge pipes for cofferdams adjacent to cargo oil tanks are to be entirely separate from those for spaces not adjacent to cargo oil tanks. However, common bilge pumps (except cargo oil pump) may be used for the bilge drainage purposes of such spaces subject to Society approval, provided that the bilge pipes for such spaces not adjacent to cargo oil tanks have a non-return valves.

4 Sounding pipes of cofferdams adjacent to cargo oil tanks are not to be less than 38 *mm* in internal diameter and, unless otherwise approved by the Society, they are to be led to above the weather deck.

14.3.2 Ballast Tanks adjacent to Cargo Oil Tanks*

1 The requirements given in **14.3.2** are also applied to ballast tanks used as cofferdams at the fore and aft ends of cargo oil tanks in accordance with the requirements given in **2.1.1.1-1(3), Part 2-7, Part C**. However, other requirements will be applied, if the fore ends of these ballast tanks are located forward of the collision bulkhead.

2 Dangerous ballast pipes (*see*, Note 2 of **Table D12.6(1)**), such as those ballast pipes of ballast tanks adjacent to cargo oil tanks, are to be separated from other pipes and are not to be led to the engine room. For this purpose, an exclusive pump for ballasting and de-ballasting these tanks is, generally, to be provided in the pump room. However, where specially approved by the Society, these cargo pumps may be used for the purpose of only de-ballasting in an emergency.

3 Slip joints used in the ballast pipes of ballast tanks adjacent to cargo oil tanks are to comply with the requirements specified in **12.3.3**.

4 Air pipes to ballast tanks adjacent to cargo oil tanks are to be provided with easily renewable wire gauze to prevent any passage of flame at their outlets. In cases where approved by the Society, the requirements given in **13.6.3(1)** for the dimensions of air pipes will be properly modified.

5 Sounding pipes of ballast tanks adjacent to cargo oil tanks are to be led to above weather decks, unless otherwise approved by the Society.

14.3.3 Fuel Oil Tanks adjacent to Cargo Oil Tanks*

Sounding pipes of fuel oil tanks adjacent to cargo oil tanks are to be led to above the weather deck, unless otherwise approved by the Society.

14.3.4 Pump Arrangements of Forward Compartments*

Pumps used for bilge drainage or transfer of ballast water or fuel oil in compartments forward of cargo oil tanks are to be exclusive and, unless otherwise approved by the Society, to be installed in the forward parts of ships. However, in cases where approved by the Society, other suitable pumps than those specified above may be used for bilge drainage or the transfer of ballast water in compartments forward of the cargo oil tanks.

14.4 Ships Only Carrying Oils Having Flashpoints above 60°C

14.4.1 General

In the case of ships only carrying oils having flashpoints above 60°C, the requirements given in **14.1** to **14.3** will be partially modified in accordance with the following **(1)** to **(4)**:

(1) The requirements given in **14.1.2** to **14.2.9** may be properly modified.

(2) Bilges of cargo oil pump rooms and cofferdams adjacent to cargo oil tanks may be led to engine rooms (*see* **14.3.1**).

(3) Ballast pipes of ballast tanks adjacent to cargo oil tanks may be led to engine rooms (*see* **14.3.2-2**). Wire gauze, to prevent the passage of any flame, required for the outlets of air pipes to cargo oil tanks may be omitted (*see* **14.3.2-4**). Sounding pipes of these tanks may be arranged to have openings below weather decks (*see* **14.3.2-5**).

(4) Sounding pipes of fuel oil tanks adjacent to cargo oil tanks may not be led to above weather decks (*see* **14.3.3**).

14.5 Piping Systems for Combination Carriers

14.5.1 Scope

1 The requirements given in **14.5** apply to piping systems and venting systems of ships designed to alternate between carrying oil or solid cargo in bulk.

2 For all of the items especially mentioned in this **14.5**, the requirements given in **14.5** are to be applied in lieu of any of the requirements given in other Sections of this Part.

14.5.2 Terminology

The terms used in **14.5** are defined as follows:

(1) Combination carriers are defined as ore/oil carriers and as *B/O* carriers.

(2) Slop tanks are defined as any tank which is provided mainly for carrying tank washings and cargo oil and which is designed to be capable of loading oil whose flash point does not exceed 60°C in cases where ships are in dry cargo mode.

(3) Solid cargo/oil holds are defined as compartments which are used as solid cargo stowing holds in cases where ships are in dry cargo mode and which are used as cargo oil tanks in cases where ships are not in dry cargo mode.

(4) Ballast/solid cargo holds is defined as compartments which are used exclusively as ballast tanks adjacent to cargo oil tanks in cases where ships are not in dry cargo mode and which are used as solid cargo stowing holds in cases where ships are in dry cargo mode.

(5) Exclusive solid cargo holds are defined as compartments which are used as void spaces adjacent to cargo oil tanks in cases where ships are not in dry cargo mode and which are used as solid cargo stowing holds in cases where ships are in dry cargo mode.

(6) Oil/ballast tanks are defined as tanks which are used as cargo oil tanks in cases where ships are not in dry cargo mode and which are used as ballast tanks or void spaces in cases where ships are in dry cargo mode.

(7) Exclusive ballast tanks are defined as compartments which are adjacent to cargo oil tanks in cases where ships are not in dry cargo mode and which are used as exclusive tanks for ballast even in cases where ships are in or not in dry cargo mode.

(8) Cargo hold is a general term for any solid cargo/oil hold, ballast/solid cargo hold and exclusive solid cargo hold.

(9) Cargo oil tank is a general term for any solid cargo/oil hold, oil/ballast tank and slop tank.

14.5.3 Bilge Piping Systems*

1 Bilge piping systems for cargo holds are not to be led to engine rooms. Cargo oil pumps may be used for the purpose of bilge suction on the condition that any cargo oil piping systems in cargo oil pump rooms that are used for bilge suction comply with the requirements given in **13.5.3** and **13.5.4**.

2 Bilge suction pipes for the cargo holds are to comply with the following requirements:

(1) Cargo oil pipes may be used as bilge suction pipes for cargo holds in cases where two or more cargo oil piping systems (e.g. main and stripping lines) are provided or cargo oil piping systems are provided independently for oil/ballast tanks and cargo holds and in cases where these cargo oil piping systems are arranged so that any liquid in all or selected oil/ballast tanks and cargo holds can be discharged (for the oil/ballast tanks, include the filling of ballasting water) simultaneously in cases where ships are in dry cargo mode. The diameters of cargo oil pipes used as bilge suction pipes are not to be less than those specified for bilge suction pipes.

(2) In cases where bilge suction pipes are provided for exclusive use, exclusive pumps for bilge suction are to be provided in cargo oil pump rooms, or bilge suctions are to be connected to cargo oil pumps in cargo oil pump rooms. In cases where cargo oil pumps are used as bilge pumps, stop valves and screw-down non-return valves are to be provided at the connections between bilge pipes and cargo oil pumps.

3 Bilge suctions in cargo holds are to comply with the following requirements:

(1) In general, one bilge suction is to be arranged on each side of the aft ends of cargo holds. In cases where the lengths of cargo hold in ships having only one cargo hold exceeds 66 m, additional bilge suction is to be arranged in suitable positions in the forward half-lengths of holds.

(2) Bilge wells are to be arranged at suitable positions so as to protect any cover plates from being directly struck by any solid cargo, and to be provided with strum boxes, or other suitable means so that suction may not be choked by ore dust, etc.

(3) Bilge wells in solid cargo/oil holds and ballast/solid cargo holds, except in cases where these bilge wells are also used as cargo oil suction wells, are to be provided with cover plates to blank off these wells or to be provided with blank flanges to blank off the open ends of bilge suction pipes in cases where ships are not in dry cargo mode.

4 In the case of exclusive bilge suction pipes, branch bilge suction pipes are to comply with the requirements given in **13.5** in addition to those requirements given in **-3**. In calculating the inside diameters of branch bilge suction pipes for the draining of cargo hold bilge of ore/oil carriers, the mean widths of such cargo holds may be used in lieu of B. Bilge suction pipes which are also used as cargo oil pipes or which are connected to eductors are to, in addition to complying with the requirements given in **-2** and **-3,** be to the satisfaction of the Society.

14.5.4 Cargo Oil Piping Systems

1 Cargo oil suctions in solid cargo/oil holds, except in cases where these suctions are also used as bilge suctions, are to be provided with blank flanges to blank off the open ends of cargo oil suction pipes or to be provided with cover plates to blank off cargo oil suction wells in cases where ships are in dry cargo mode.

2 In addition to **14.5**, cargo oil piping systems for combination carriers are to comply with the requirements given in **1.2.4** and **4.5.1-4**, **Part R**.

14.5.5 Ventilating Systems

Ventilating systems for combination carriers are to comply with the requirements given in **4.5.4-2, Part R**.

14.6 Tests

14.6.1 Shop Tests

After the manufacture of any piping systems for tankers and ships carrying dangerous chemicals in bulk, tests are to be conducted in compliance with the requirements given in **12.6**.

14.6.2 Tests after Installation On Board

1 For piping systems for which welding between pipes or between pipes and pipe fittings is carried out on board the ship, all joints welded on the board ship are to be subjected to the non-destructive testing specified in **11.6**.

2 Cargo oil pipes, after the completion of their installation, are to be subjected to leak tests at a pressure not less than 1.25 *times* the design pressure.

3 Heating pipes inside cargo oil tanks are, after assembly on board, to be subjected to leak tests at a pressure not less than 1.5 *times* the design pressure or 0.4 *MPa*, whichever is greater.

4 For the leak tests in **-2** and **-3** above, either pneumatic testing, or a combined hydrostatic-pneumatic strength testing may be carried out in accordance with **13.17.2-4**.

5 After installation on board, auxiliaries and piping systems are to be subjected to the following tests:

(1) Function tests of cargo oil pumps.

(2) Function tests of various systems concerning the safety measures specified in this Chapter.

Chapter 15 STEERING GEARS

15.1 General

15.1.1 Scope*

1 The requirements in this Chapter apply to power-driven steering gears.

2 For those items mentioned in this Chapter, the requirements given in this Chapter are applied in lieu of the requirements in **Chapters 12** and **13**.

3 Electrical equipment and cables used for steering gears are to conform to the requirements of **Part H** in addition to those specified in this Part.

4 Manual steering gears will be considered by the Society in each case.

15.1.2 Terminology

The terms used in this Chapter are defined as follows:

(1) A "main steering gear" is defined as the machinery, rudder actuators, steering gear power units, if any, and ancillary equipment and the means of applying torque to the rudder stock (tiller, etc) necessary for effecting movement of the rudder for the purpose of steering the ship under normal service conditions.

(2) An "auxiliary steering gear" is defined as the equipment other than any part of the main steering gear necessary to steer the ship in the event of failure of the main steering gear but not including tiller, etc.

(3) A "steering gear power unit" (hereinafter referred to as "power unit") is:

(a) in the case of electric gear: an electric motor and its associated electrical equipment;

(b) in the case of electrohydraulic steering gear: a hydraulic pump, electric motor and its associated electrical equipment; and

(c) in the case of hydraulic steering gear other than those in **(b)**: a hydraulic pump and its driving engine.

(4) A "power actuating system" is defined as the hydraulic equipment provided for supplying power to turn the rudder stock, comprising a power unit or units, together with the associated hydraulic pipes and fittings, and a rudder actuator. The power actuating systems may share common mechanical components, i.e., tiller, etc.

(5) A "rudder actuator" is defined as the component which converts directly hydraulic pressure into mechanical action to move the rudder.

(6) A "control system" is defined as the equipment by which orders are transmitted from the navigating bridge to the power units. Steering gear control systems comprise transmitters, receivers, hydraulic control pumps and their associated motors, motor controllers, piping and cables. Steering gear control systems are also understood to cover "equipment required to control steering gear power actuating systems".

(7) "Maximum working pressure" means the maximum expected pressure in the system when the steering gear is operated under the conditions specified in **15.2.2(1)**.

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

15.1.3 Drawings and Data*

Drawings and data to be submitted are generally to be as follows:

(1) Drawings:

(a) General arrangements of steering gear

(b) Details of tiller, etc.

(c) Assembly and details of power units

(d) Assembly and details of rudder actuators

(e) Piping diagram of hydraulic pipes; Arrangements of control systems

(f) Diagram of hydraulic and electrical systems (including alarm devices and automatic steering gear)

(g) Arrangements and diagrams of an alternative source of power

(h) Diagram of the rudder angle indicator

(i) Other drawings considered necessary by the Society

(2) Data:

(a) Particulars

(b) Operating instructions (including drawings showing the change-over procedure for power units and control systems, drawings showing the sequence of automatic supply of power from an alternative source of power, data showing the kind, particulars and assembly of power sources in the case that the alternative source of power is an independent source of power and information about hydraulic fluid quality)

(c) Manuals for countermeasures to be taken at the time of a single failure of the power actuating system;

(d) Calculation sheet of the strength of essential parts.

(e) Other data considered necessary by the Society.

15.1.4 Display of Operating Instructions, etc. *

1 Simple operating instructions with block diagrams showing the change-over procedures for power units and control systems are to be permanently displayed on the navigating bridge and in the steering gear compartment for all ships equipped with power-operated steering gears.

2 In cases where system failure alarms are provided in accordance with to **15.3.1-6**, appropriate instructions for emergency procedures related to such alarms are to be permanently displayed on the navigation bridge.

15.1.5 Operating and Maintenance Instructions for Steering Gears

Operating and maintenance instructions and engineering drawings for steering gears are to be provided and written in a language understandable by any officers and crew members who are required to understand such information in the performance of their duties.

15.2 Performance and Arrangement of Steering Gears

15.2.1 Number of Steering Gears*

1 Unless expressly specified otherwise, every ship is to be provided with a main steering gear and an auxiliary steering gear. The main steering gear and the auxiliary steering gear are to be so arranged that the failure of one of them will not render the other one inoperative.

2 In cases where the main steering gear comprises two or more identical power units, an auxiliary steering gear need not be fitted, provided that:

(1) The main steering gear is capable of operating the rudder as required by **15.2.2(1)** while operating with all power units;

(2) 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. Steering gears, other than those of the hydraulic type, will be considered by the Society in each case.

15.2.2 Performance of Main Steering Gear

The main steering gear is to be:

(1) Capable of putting the rudder over from 35 *degrees* on one side to 35 *degrees* on the other side with the ship at its load draught and running ahead at the speed specified in **2.1.8, Part A** and, under the same conditions, from 35 *degrees* on either side to 30 *degrees* on the other side in not more than 28 *seconds*;

(2) Operated by power when the main steering gear has to meet the requirements in **(1)** or when the diameter of the upper stock is required in **Chapter 13, Part 1, Part C** to be over 120 *mm* (calculated with a material factor K_S =1 where K_S is less than 1, and excluding the increase required for ships which have strengthening for navigation in ice, the same being referred hereinafter); and

(3) So designed that they will not be damaged at maximum astern speed; however, this design requirement need not be proved by trials at maximum astern speed and maximum rudder angle.

15.2.3 Performance of Auxiliary Steering Gear*

The auxiliary steering gear is to be:

(1) Capable of putting the rudder over from 15 *degrees* on one side to 15 *degrees* on the other side in not more than 60 *seconds*

with the ship at its load draught and running ahead at one half of the speed specified in **2.1.8, Part A** or 7 *knots*, whichever is greater, and capable of being brought speedily into action in an emergency; and

(2) Operated by power where necessary to meet the requirement in **(1)** and in any case when the diameter of upper stock is required in **Chapter 13, Part 1, Part C** to be over 230 *mm*.

15.2.4 Piping

1 The hydraulic piping system is to be arranged so that transfer between power units can be readily effected.

2 Suitable arrangements to maintain the cleanliness of the hydraulic fluid are to be provided after taking into consideration the type and design of the power actuating system.

3 Arrangements for bleeding air from power actuating system are to be provided where necessary.

4 Relief valves are to be fitted to any part of the hydraulic system which can be isolated and in which pressure can be generated from the power source or from external forces. The setting pressure of the relief valves is not to be less than 1.25 times the maximum working pressure expected in the protected part. The minimum discharge capacity of the relief valves are not to be less than the total capacity of pumps which provide power for the actuator, increased by 10%. Under such conditions the rise in pressure is not to exceed 10% of the setting pressure. In this regard, due consideration is to be given to the extreme foreseen ambient conditions in respect of oil viscosity.

5 A low level alarm is to be provided for each hydraulic fluid reservoir to give the earliest practical indication of any hydraulic fluid leakage. This alarm is to be audible and visual and to be given on the navigating bridge and at a position from where the main engine is normally controlled.

6 A fixed storage tank having sufficient capacity to recharge at least one power actuating system including the reservoir, in cases where the main steering gear is operated by hydraulic power. This storage tank is to be permanently connected by piping in such a manner that the hydraulic system can be readily recharged from a position within the steering gear compartment and is to be provided with a contents gauge.

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

15.2.5 Re-start and Power-failure Alarm of Power Units

Main and auxiliary steering gear power units are to be:

(1) Arranged to re-start automatically when power is restored after a power failure; and

(2) Capable of being brought into operation from a position on the navigating bridge. In the event of a power failure to any one of the power units, an audible and visual alarm is to be given on the navigating bridge.

15.2.6 Alternative Source of Power

In cases where the diameter of upper stock is required in **Chapter 13, Part 1, Part C** to be over 230 *mm*, an alternative source of power is to be provided in accordance with the following:

(1) The alternative source of power is to be either:

  (a) An emergency source of electric power; or

  (b) An independent source of power located in the steering gear compartment and used only for this purpose.

(2) Any alternative source of power is to be capable of automatically supplying within 45 *seconds*, alternative power to the power unit and its associated control system and the rudder angle indicator. In this case the alternative source of power is to be capable of giving sufficient power to the power unit so that the steering capability required by **15.2.3(1)** can be regained. In every ship of 10,000 *gross tonnage* or more, alternative sources of power are to have the capacity for at least 30 *minutes* of continuous operation of the steering gear and for at least 10 *minutes* in any other ship.

(3) Automatic starting arrangements for generators or prime movers of pumps used as the independent source of power specified in **(1)(b)** are to comply with the requirements for starting devices and performance in **3.4.1, Part H**.

15.2.7 Electrical Installations for Electric and Electrohydraulic Steering Gear*

1 Cables used in power circuits required to be installed in duplicate by this Chapter are to be separated as far apart as practicable throughout their entire length.

2 Means for indicating that power units are running is to be installed on the navigating bridge and at the position from which the main engine is normally controlled.

3 Each electric or electrohydraulic steering gear comprising one or more power units is to be served by at least two exclusive circuits fed directly from the main switchboard. However, one of these circuits may be supplied through the emergency switchboard.

4 Any auxiliary electric or electrohydraulic steering gear associated with the main electric or electrohydraulic steering gear may be connected to one of the circuits supplying this main steering gear. These circuits are to have adequate rating for supplying all motors which can be simultaneously connected to them and may be required to be operated simultaneously.

5 Short circuit protection and overload alarms are to be provided for such circuits and motors. The overload alarm is to be both audible and visible and to be situated in a conspicuous position in the place from where the main engine is normally controlled.

6 Protection against any excess current, including the starting current, if provided, is to be for not less than twice the full load current of the motor or circuit so protected, and to be arranged to permit the passage of the appropriate starting currents.

7 In cases where a three-phase supply is used, an alarm is to be provided that will indicate failure of any one of the supply phases. The alarm is to be both audible and visible and to be situated in a conspicuous position in the place from where the main engine is normally controlled.

8 When any auxiliary steering gear, in ships of less than 1,600 *gross tonnage*, which is required by **15.2.3(2)** to be operated by power is not electrically powered or is powered by an electric motor primarily intended for other services, the main steering gear may be fed by one circuit from the main switchboard. However, in cases where such an electric motor, primarily intended for other services, is arranged to power such auxiliary steering gear, the requirements in **-5** to **-7** may be waived if the Society is satisfied that these protection arrangements are in accordance with the requirements in **15.2.5** and **15.3.1-1(3)** for auxiliary steering gear.

9 For ships with a gross tonnage less than 1,600 *tons* that are equipped with manual auxiliary steering gears, the power supply circuit from the main switchboard to the main steering gear may be one circuit.

15.2.8 Position of Steering Gears*

1 Steering gear is to be installed in an enclosed compartment readily accessible, and, as far as possible, separated from any machinery spaces.

2 Steering gear compartments are to be provided with suitable arrangements to ensure working access to steering gear machinery and controls. These arrangements are to include handrails and gratings or other non-slip surfaces to ensure suitable working conditions in the event of hydraulic fluid leakage.

15.2.9 Means of Communication*

A means of communication is to be provided between the navigating bridge and the steering gear compartment.

15.2.10 Rudder Angle Indictor

The angular position of rudder is to be:

(1) indicated on the navigating bridge. The rudder angle indicator is to be independent of the control system;

(2) recognizable in the steering gear compartment.

15.3 Controls

15.3.1 General*

1 Steering gear control is to be provided:

(1) For the main steering gear, both on the navigating bridge and in the steering gear compartment;

(2) In cases where the main steering gear is arranged in accordance with the requirements in **15.2.1-2**, by two independent control systems, both operable from the navigating bridge. This does not require duplication of the steering wheel or steering lever. In cases where the control system consists of a hydraulic telemotor, a second independent system need not be fitted.

(3) For any auxiliary steering gear, in the steering gear compartment; and, if power operated, it is also to be operable from the navigating bridge and to be independent of the control systems for main steering gear.

2 Any main and auxiliary steering gear control system operable from the navigating bridge is to comply with the following:

(1) If electric, it is to be served by its own separate circuit supplied from a steering gear power circuit from a point within the steering gear compartment, or directly from switchboard busbars supplying that steering gear power circuit at a point on the switchboard adjacent to the supply to the steering gear power circuit.

(2) Means are to be provided in the steering gear compartment for disconnecting any control system operable from the navigating bridge from the steering gear it serves.

(3) The system is to be capable of being brought into operation from a position on the navigating bridge.

(4) In the event of a failure of electrical power supply to the control system, an audible and visual alarm is to be given on the navigating bridge.

(5) Short circuit protection only is to be provided for steering gear control supply circuits.

3 For the control systems specified in the requirements of **-1(2)** above, at least the following most probable failures that may cause reduced or erroneous system performance are to be automatically detected and individual visible and audible alarms are to be given on the navigation bridge:

(1) Power supply failure

(2) Earth fault on AC and DC circuits

(3) Loop failure in closed loop systems, both command and feedback loops (normally short circuit, broken connections and earth faults)

(4) Data communication errors

(5) Programmable system failures (Hardware and software failures)

(6) In the case of closed loop systems, deviation between rudder order and feedback

Individual visible and audible deviation alarms are to be initiated on the navigation bridge when the rudder's actual position does not reach its set point within acceptable time limits (e.g. follow-up control and autopilot). The deviation alarm may be caused by mechanical, hydraulic or electrical failures.

4 For the control systems specified in the requirements of **-1(2)** above, the failures (as defined but not limited to those in **-3** above) likely to cause uncontrolled movement of the rudder are to be clearly identified. In the event of such a failure, the following response is to be implemented:

(1) the rudder is to stop in the angle when failure occurs without manual intervention, or

(2) the rudder is to return to the midship/neutral position.

For mechanical failures such as sticking valves and failure of static components (pipes, cylinders), system response without manual intervention is not mandatory, and operators may instead follow instructions permanently displayed in accordance with **15.1.4-2** in the case of such failures.

5 Cables and pipes of control systems required to be in duplicate by this Chapter are to be separated as far apart as is practicable throughout their entire length.

6 For the steering gears which are so arranged that more than one system (either power or control) can be simultaneously operated, where hydraulic locking, caused by a single failure, may lead to loss of steering, audible and visual alarms, which identifies the failed system, are to be provided on the navigation bridge.

15.3.2 Change-overs from Automatic to Manual Steering*

The steering gears of ships provided with automatic pilots are to be capable of immediate change-overs from automatic to manual steering.

15.4 Materials, Constructions and Strength of Steering Gears

15.4.1 Materials

1 Materials used in the steering gears are to be sound, flawless and adequate for their service conditions.

2 Materials used for cylinders and housings of rudder actuators, piping subjected to a hydraulic pressure and components transmitting mechanical forces to the rudder stock are not to have a minimum elongation of less than 12%, or a specified tensile strength in excess of 650 N/mm^2. This does not apply to those materials for valves and bolts where approved by the Society.

3 Materials used for tillers are to be forged steels or cast steels tested in accordance with the requirements in **Part K**.

4 Materials used for bosses and vanes of rotary vane type rudder actuators are to be forged steels, cast steels or nodular graphite cast irons tested in accordance with the requirements in **Part K**.

5 Materials used for bolts for assembling split type tillers and bolts for securing vanes to the bosses of rotary vane type rudder actuators are to be forged steels or rolled steels tested in accordance with the requirements in **Part K**.

6 Materials used for major parts other than those mentioned in **-3** to **-5** are to comply with the requirements in those recognized standards deemed appropriate by the Society.

7 Materials other than those mentioned in **-2** to **-6** may be used where approved by the Society.

15.4.2 Welding

1 All welded joints of parts used in power actuating systems are to be such that there is no incomplete penetration or any other injurious defects.

2 Welded joints in parts subjected to the internal pressure of power actuating systems are to have sufficient strength.

15.4.3 General Construction of Steering Gear

1 Steering gears are to be of sufficient strength and reliability.

2 Configurations of the major parts of steering gears are to be determined to avoid any local concentration of stress.

3 The design pressure for calculations to determine the scantlings of piping and other steering gear components subjected to internal hydraulic pressure is to be at least 1.25 times the maximum working pressure to be expected under the operational conditions specified in **15.2.2(1)**, taking into account any pressure which may exist in the low pressure side of the system. The design pressure is not to be less than the relief valve setting pressure.

4 Special consideration is to be given to the suitability of any essential component which is not duplicated. Any such essential component is, in cases where appropriate, to utilize anti-friction bearings such as ball bearings, roller bearings or sleeve bearings which are to be permanently lubricated or provided with lubrication fittings.

5 In cases where considered necessary, fatigue analysis is to be carried out on piping and components, taking into account any pulsating pressure due to dynamic loads. Both the cases of high cycle and cumulative fatigue are to be considered.

15.4.4 Strength of Rudder Actuators

1 The strength of all of the components of rudder actuators subjected to an internal pressure, except for the amount of allowable stress specified in this Chapter, is to comply with relevant requirements in **Chapter 10**.

2 In the strength calculations specified in **-1**, the allowable stress for any equivalent primary general membrane stress is to be not greater than the following values **(1)** or **(2)**, whichever is smaller:

(1) $\frac{\delta_B}{A}$

(2) $\frac{\delta_Y}{B}$

where

δ_B : Specified tensile strength of the material (N/mm^2)

δ_Y : Specified yield strength or 0.2% proof stress of the material (N/mm^2)

A and B : As given in **Table D15.1**.

Table D15.1 A and B

	Rolled or Forged Steel	Cast Steel	Nodular Cast Iron
A	3.5	4	5
B	1.7	2	3

15.4.5 Oil Seals in Rudder Actuators

1 Oil seals between any non-moving parts which form part of the external pressure boundary, are to be of the metal upon metal type or of an equivalent type.

2 Oil seals between any moving parts which form part of the external pressure boundary, are to be duplicated, so that the failure of one seal does not render the actuator inoperative. Alternative arrangements providing equivalent protection against leakage will be accepted in cases where approved by the Society.

15.4.6 Flexible Hose Assembly

Flexible hose assemblies specified in **12.3.4** are to be used in piping systems in cases where flexibility is required.

15.4.7 Tillers, etc. *

1 The scantlings of tillers, etc., made of forged steels or cast steels, which transfer power from the rudder actuator to the rudder stock, are to be so determined so that the bending stress does not exceeding $118/K$(N/mm^2) and the shearing stress does not exceeding $68/K$(N/mm^2) when the rudder torque T_R is applied.

where

T_R : Rudder torque specified in **13.2.3, Part 1, Part C** (N-m).

K : Material coefficient of the tiller, specified in **13.2.1.2, Part 1, Part C**

2 Notwithstanding the requirement specified in **-1**, the scantlings of rapson-slide type or trunk piston type tillers may be determined according to the following **(1)** to **(4)**:

(1) The vertical section of each side of tiller boss at the centre line of rudder stock is to comply with the following formulae:

$$(D^2 - d^2)H \geq 170T_RK$$

$$H/d \geq 0.75$$

where

D : Outer diameter of boss (mm).

d : Inner diameter of boss (mm).

H : Depth of boss (mm).

T_R : Rudder torque specified in **13.2.3, Part 1, Part C** (N-m).

K : Material coefficient of the tiller, specified in **13.2.1.2, Part 1, Part C**

(2) The section modulus of an arm about its vertical axis is to be not less than that obtained from the following formula:

$$Z_{TA} = 11\left(1 - \frac{r}{R_1}\right)T_RK$$

where

Z_{TA} : Required section modulus of the arm about its vertical axis (mm^3).

r : Distance from the centre of rudder stock to the section (mm).

R_1 : Length of the tiller arm measured from the centre of the rudder stock to the point of application of the driving force (mm). In cases where this length varies in accordance with rudder angle, R_1 is the maximum length within 35 *degrees* of rudder angle.

T_R : Rudder torque specified in **13.2.3, Part 1, Part C** (N-m).

K : Material coefficient of the tiller, specified in **13.2.1.2, Part 1, Part C**

(3) The sectional area of an arm at its outer end is to be not smaller than that obtained from the following formula:

$$A_R = 18.5\frac{T_R}{R_2}K$$

where

A_R : Required sectional area of the arm at its outer end (mm^2).

R_2 : Length of the tiller arm measured from the centre of the rudder stock to the point of application of the driving force (mm). In cases where this length varies in accordance with rudder angle, R_2 is the length at 0 *degrees* of rudder angle.

T_R : Rudder torque specified in **13.2.3, Part 1, Part C** (N-m).

K : Material coefficient of the tiller, specified in **13.2.1.2, Part 1, Part C**

(4) In cases where a tiller having two arms which have power units that are connected to each arm and these two power units are driven simultaneously, the scantlings of the arms may be reduced from those required in **(2)** and **(3)** to a value recognized by the Society.

3 Notwithstanding the requirement specified in **-1**, the scantlings of rotary vane type rudder actuators of forged steels or cast steels may be determined according to the following requirements, in addition to those requirements specified in **15.4.4**.

(1) Scantlings of the boss are to comply with the requirement specified in **-2(1)**.

(2) The section modulus about the vertical axis and the sectional area of vane is to be not less than that obtained from the following formulae:

$$Z_v = 11\left(\frac{B}{D+B}\right)\frac{T_R}{n}K$$

$$A_R = 37\left(\frac{1}{D+B}\right)\frac{T_R}{n}K$$

where

Z_v : Required section modulus of vane about the vertical axis (mm^3).

A_R : Required sectional area of vane (mm^2).

D : Outer diameter of boss (mm).

B : Height of vane measured from outer surface of boss (mm).

n : Number of vanes.

T_R : Rudder torque specified in **13.2.3, Part 1, Part C** (N-m).

K : Material coefficient of the vane, specified in **13.2.1.2, Part 1, Part C**

4 In cases where tillers which are separated into two pieces are bolted, there are to be at least two bolts on each side of the head. The diameter of bolts at bottom of thread is not to be less than that obtained from the following formula. In such case, the thickness of any coupling flange is to not less than three-fourth of the diameter of the bolts.

$$d_b = 1.45\sqrt{\frac{T_R}{nb}K}$$

where

d_b : Required diameter of bolts at bottom of thread (mm).

T_R : Rudder torque specified in **13.2.3, Part 1, Part C** (N-m).

K : Material coefficient of the bolt, specified in **13.2.1.2, Part 1, Part C**

n : Number of bolts on each side of the head.

b : Distance from the centre of rudder stock to the centre of bolt (cm).

5 Tillers are to be coupled, using a key, to rudder stocks by shrinkage fitting, force fitting or the bolted method. However, tillers may be coupled without a key, in cases where the fitting methods are in compliance to the satisfaction of the Society.

6 Scantlings of rotary vane type rudder actuators of nodular graphite cast iron are to be specified to not to be applied with bending stress exceeding $94/K$ (N/mm^2), or shearing stresses exceeding $54/K$ (N/mm^2) under the rudder torque T_R applied. Alternatively, the scantlings may be determined according to the requirements specified in **-3**, using 1.2 *times* the rudder torque T_R specified in **13.2.3, Part 1, Part C** as rudder torque for calculating.

15.4.8 Stoppers

1 Tillers are to be provided with rudder stoppers.

2 Steering gears are to be provided with positive arrangements, such as limit switches, for stopping the gear before the rudder stops are reached. These arrangements are to be synchronized with the gear itself and not with the steering gear control. However, these arrangements may be operated through mechanical links such as floating levers.

3 Suitable brake arrangements or ropes are to be provided to tillers to keep the rudder steady in the event of an emergency. In the case of hydraulic steering gear, where the rudder can be stopped safely by closing the oil pressure valves, this brake arrangement will not be required.

15.4.9 Buffers*

Steering gears other than those of a hydraulic type are to be provided with spring buffers or other suitable buffer arrangements to relieve the gears from any shock given off by the rudder.

15.5 Testing

15.5.1 Shop Tests

1 Pressure vessels and piping systems are to be subjected to tests in accordance with the requirements in **10.9**, **12.6** and **13.17** in addition to the tests specified in **15.5**.

2 All pressure parts are to be subjected to pressure tests with a pressure equal to 1.5 times the design pressure.

3 Each type of pump used as a power unit is to be subjected to a running test for the duration of not less than 100 *hours*. Test arrangements are to be such that the pump may run in idle condition, and at maximum delivery capacity at maximum working pressure. The passage from one condition to another is to occur at least as quickly as on board. During the test, idling periods are to be alternated with periods at maximum delivery capacity at maximum working pressure. During the whole test no abnormal heating, excessive vibration or other irregularities are permitted. After the test, the pump is to be disassembled to ascertain that there are no abnormalities. The test may be waived for a power unit which has been proved to be reliable in marine service.

15.5.2 Tests after Installation On Board*

1 Hydraulic piping systems are, after installed on board, to be subjected to a leak test at a pressure that is at least equal to their maximum working pressure.

2 The steering gear is, after installed on board, to be subjected to a running test.

3 In cases where the steering gear is designed to avoid any hydraulic locking, this feature is to be demonstrated.

15.6 Additional Requirements Concerning Tankers, Ships Carrying Liquefied Gases in Bulk or Ships Carrying Dangerous Chemicals in Bulk of 10,000 *Gross Tonnage* or More and Other Ships of 70,000 *Gross Tonnage* or More

15.6.1 Main Steering Gears

1 In tankers, ships carrying liquefied gases in bulk or ships carrying dangerous chemicals in bulk of 10,000 *gross tonnage* or more and in all other ships of 70,000 *gross tonnage* or more, the main steering gear is to be comprised of two or more equivalent power units complying with the requirements in **15.2.1-2**.

2 Steering gears in oil tankers, ships carrying liquefied gases in bulk or ships carrying dangerous chemicals in bulk of 10,000 *gross tonnage* or more are to comply with the following:

(1) The main steering gear is to be so arranged that in the event of any loss of steering capability due to a single failure in any part of one of the power actuating systems of the main steering gear, excluding failure in the tiller and seizure in the rudder actuator, steering capability is to be regained in not more than 45 *seconds* after the loss of one power actuating system;

(2) The main steering gear is to comprise either:

(a) Two independent and separate power actuating systems, each capable of meeting the requirements in **15.2.2(1)** ; or

(b) At least two equivalent power actuating systems which, acting simultaneously in normal operation, are to be capable of meeting the requirements in **15.2.2(1)**. In this case, the following requirements of **i)** and **ii)** are also to be met:

i) Any loss of hydraulic fluid from one system is to be capable of being detected and the defective system automatically isolated so that the other actuating system or systems are to remain fully operational.

ii) In cases where necessary to obtain steering capability, interconnection of hydraulic power actuating systems is to be provided.

3 Steering gears other than those of the hydraulic type will be considered by the Society in each case.

15.6.2 Controls

In the case of tankers, ships carrying liquefied gases in bulk or ships carrying dangerous chemicals in bulk of 10,000 *gross tonnage* or more, the modification for the hydraulic telemotor permitted in **15.3.1-1(2)** is not to be applied.

15.6.3 Number and Strength of Rudder Actuators

1 For tankers, ships carrying liquefied gases in bulk or ships carrying dangerous chemicals in bulk of 10,000 *gross tonnage* or more, but of less than 100,000 *tons deadweight*, a single rudder actuator may be permitted provided that

(1) Following any loss of steering capability due to a single failure of any part of the piping system or in one of the power units, steering capability is to be regained within 45 *seconds*;

(2) Special consideration is to be given to the stress analysis of the design including fatigue analysis and fracture mechanics analysis, as appropriate, to the material used, to the installation of sealing arrangements, to testing and inspection and to the provision of effective maintenance. In this case, both high cycle fatigue and cumulative fatigue are to be considered.

(3) Isolating valves are to be directly mounted on the rudder actuator so as to isolate the rudder actuator from the hydraulic oil in the piping systems; and

(4) Relief valves for protecting the rudder actuator against overpressure as required in **15.2.4-4** are to be provided.

2 For tankers, ships carrying liquefied gases in bulk or ships carrying dangerous chemicals in bulk of 10,000 *gross tonnage* or more, but less than 100,000 *tons deadweight* and equipped with a single rudder actuator, the strength of the rudder actuator is to comply with the following requirements in addition to those of **15.4.4** :

(1) A detailed calculation of all the major parts of the rudder actuator is to be carried out to confirm their strength.

(2) A detailed stress analysis of all the parts of rudder actuators subject to hydraulic pressure is to be carried out to confirm that their strength is sufficient to withstand design pressure.

(3) In cases where considered necessary because of design complexity or manufacturing procedures, a fatigue analysis and fracture mechanics analysis are to be carried out. In this case, both high cycle fatigue and cumulative fatigue are to be considered. In connection with this analysis, all foreseen dynamic loads are to be taken into account. In cases where considered necessary by the Society, an experimental stress analysis may be required in addition to, or in lieu of, the theoretical calculations.

(4) For the purpose of determining the general scantlings of the parts of rudder actuators subject to internal hydraulic pressure, the allowable stresses are not exceed:

(a) $\delta_m \leq f$

(b) $\delta_l \leq 1.5f$

(c) $\delta_b \leq 1.5f$

(d) $\delta_l + \delta_b \leq 1.5f$

(e) $\delta_m + \delta_b \leq 1.5f$

where

δ_m : Equivalent primary general membrane stress (N/mm^2)

δ_l : Equivalent primary local membrane stress (N/mm^2)

δ_b : Equivalent primary bending stress (N/mm^2)

f : Lesser of $\frac{\delta_B}{A}$ or $\frac{\delta_Y}{B}$

δ_B : Specified tensile strength of material (N/mm^2)

δ_Y : Specified minimum yield stress or 0.2% proof stress of material (N/mm^2)

A and B : As given in **Table D15.2**.

(5) In cases where the parts of rudder actuators subject to hydraulic pressure are subjected to a burst test at the minimum bursting pressure specified below and they are confirmed to withstand this test, the detailed stress analysis required in **(2)** may be omitted. However, when considered necessary because of design complexity or manufacturing procedures, the detailed stress analysis required in **(2)** is to be carried out notwithstanding the above.

$$P_b = PA\frac{\delta_{Ba}}{\delta_B}$$

where

P_b : Minimum bursting pressure (MPa)

P : Design pressure (MPa)

A : As given in **(4)**

δ_{Ba} : Actual tensile strength of the material (N/mm^2)

δ_B : Specified minimum tensile strength of the material (N/mm^2)

Table D15.2 *A* and *B*

	Rolled or Forged Steel	Cast Steel	Nodular Cast Iron
A	4	4.6	5.8
B	2	2.3	3.5

15.6.4 Shop Tests

For tankers, ships carrying liquefied gases in bulk or ships carrying dangerous chemicals in bulk of 10,000 *gross tonnage* or more, but less than 100,000 *tons deadweight* and equipped with a single rudder actuator, the rudder actuators are to be subjected to suitable and complete non-destructive testing in order to detect both surface flaws and volumetric flaws. The procedure and acceptance criteria for this non-destructive testing will be considered by the Society in each case. In cases where considered necessary, fracture mechanics analysis is to be used for determining the maximum allowable flaw size.

Chapter 16 WINDLASSES AND MOORING WINCHES

16.1 General

16.1.1 Scope

The requirements herein **Chapter 16** apply to windlasses and mooring winches.

16.1.2 Terminology

Terms used in this chapter are defined as follows:

(1) "Prime mover" means electric motors, hydraulic motors, steam turbines and so on which drive cable lifters.

(2) "Torque-transmitting components" means components which transmit power from the prime movers to cable lifters when anchors and chain cables are paid out or hoisted; for example, shafts, gears, clutches, couplings and coupling bolts, etc. (includes components which constitute prime movers)

(3) "Load-bearing components" means components which are loaded; this, however, excludes torque-transmitting components such as shaft bearings, cable lifters, sheaves, drums, bed-frames, brakes, chain cable stoppers and foundations, etc.

16.2 Windlasses

16.2.1 General*

1 Windlasses fitted to the ship in order to handle anchors are to be suitable for the size of chain cable being used.

2 The design, construction and testing of windlasses are to conform to a standard or code of practice recognized by the Society in addition to requirements in this chapter. The standard or code of practice is to specify criteria for stress, performance and testing.

16.2.2 Drawings and Data*

The following drawings and data showing design specifications, standards of compliance, engineering analyses and details of construction, are, in principle, to be submitted.

(1) Drawings and data for approval:

 (a) Windlass design specifications

 (b) Windlass arrangement plan

 (c) Dimensions, materials and welding details of torque-transmitting components and load-bearing components

 (d) Drawings and data concerning hydraulic systems

 (e) Control, monitoring and instrumentation arrangements

 (f) Procedures for shop tests

 (g) Other drawings and data considered necessary by the Society

(2) Drawing and data for reference:

 (a) Calculated strength for torque-transmitting components and load-bearing components

 (b) General arrangements and sectional assembly drawings of chain cable stoppers and documents which demonstrate the chain cable stoppers are in accordance with requirements specified in **16.2.4-2(6)** (in cases where chain cable stoppers are fitted)

 (c) Load calculations of prime movers (in cases where the load test specified in **16.2.5-1(3)** is not carried out)

 (d) Calculation sheets for cable lifter brake capacities (in cases where the cable lifter brake capacity test specified in **16.2.5-1(4)** is not carried out)

 (e) Operation and maintenance procedures

 (f) Other drawings and data considered necessary by the Society

16.2.3 Materials and Fabrication*

1 Materials

Materials used in the construction of torque-transmitting and load-bearing components of windlasses are to comply with the following requirements:

(1) The materials are to be approved by the Society in accordance with the requirements in **Part K of the Rules**, except in cases where the material is comply with standards recognized by the Society.

(2) The Proposed materials are to be indicated in the drawings and data specified in **16.2.2(1)**.

(3) All such materials are to be certified by the material manufacturers and are to be traceable to the manufacturers' certificates.

2 Welded fabrication

Welded fabrication is to comply with the following requirements:

(1) Weld joint designs, the degree of non-destructive examination of welds and post-weld heat treatment, if any, are to be indicated in the drawings and data specified in **16.2.2(1)**.

(2) Welding procedures and related specifications are to be qualified in accordance with requirements of standards recognized by the Society or the requirements of **Chapter 11**.

(3) Each welder to be engaged in the welding work is to pass the qualification tests specified in **Part M** (including initial and renewal tests) with respect to each required welder qualification depending on the applicable welding process and materials to be welded. In addition, each welder is to obtain a qualification certificate issued by the Society.

(4) Welding consumables are to be type-approved by the Society in accordance with the requirements in **Part M**. In cases where compliance with this requirement, however, is deemed impractical, welding consumables that satisfy the following **(a)** and **(b)** may be accepted.

(a) Welding consumables that conform to standards recognized by the Society; and

(b) Welding consumables subjected to deposited weld metal tests, the results of which are deemed appropriate by the Surveyor

16.2.4 Design*

1 Windlasses and their beds as well as any other accessories and facilities are to be installed effectively and securely onto the deck.

2 Mechanical designs of windlasses are to be according to the following requirements:

(1) Design loads are to comply with the following requirements:

(a) Holding loads

Calculations are to be made to show that, in the conditions specified in **i)** and **ii)** below, the maximum stress for each load bearing component do not exceed yield strength (or 0.2% proof stress) of the material.

i) The holding condition (single anchor, cable lifter brake fully applied and cable lifter declutched)

ii) Under a load equal to 80% of the specified breaking test load of the chain cable (For installations fitted with a chain cable stopper, 45% of the specified breaking test load of the chain cable may instead be used for the calculation.)

(b) Inertia loads

Designs for drive trains (including prime movers, reduction gears, bearings, clutches, shafts, cable lifters and bolting) are to consider the dynamic effects of the sudden stopping and starting of the prime movers or chain cables so as to limit inertial loads.

(2) The continuous duty pull is to be decided in accordance with the following requirements:

(a) Prime movers are to be able to exert, for at least 30 *minutes*, a continuous duty pull corresponding to the grade and diameter of chain cable as follows:

i) Maximum anchorage depth is to be not deeper than 82.5 *m* for the following windlasses:

1) those using Grade 1 chain cables: $Z_{cont1} = 37.5d^2$ (N) $(3.82\ d^2\ (kgf))$

2) those using Grade 2 chain cables: $Z_{cont1} = 42.5d^2$ (N) $(4.33\ d^2\ (kgf))$

3) those using Grade 3 chain cables: $Z_{cont1} = 47.5d^2$ (N) $(4.84\ d^2\ (kgf))$

where

Z_{cont1}: the continuous duty pull

d: the nominal diameter of chain cable (mm)

ii) Maximum anchorage depth is to be deeper than 82.5 *m* for the following windlasses:

$Z_{cont2}\ (N) = Z_{cont1}\ (N) + (D - 82.5) \times 0.27d^2$

$(Z_{cont2}\ (kgf) = Z_{cont1}\ (kgf) + (D - 82.5) \times 0.0275d^2)$

where

Z_{cont2}: the continuous duty pull

d: the nominal diameter of chain cable (mm)

D: the maximum anchorage depth (m)

(b) In general, the stresses in each torque-transmitting component are not to exceed 40% of the yield strength (or 0.2% proof stress) of the material when the continuous duty pull is loaded.

(3) Prime movers are to be able to provide the necessary temporary overload capacity for breaking out the anchor. This temporary overload capacity or "short term pull" is to be at least 1.5 times the continuous duty pull applied for at least 2 *minutes*. The speed in this period may be lower than that specified in **(4)**.

(4) The mean speed of the chain cable during hoisting of the anchor and chain cable is to be at least 0.15 m/s when the windlass hoists over two shots of chain cable and initially with at least three shots of chain cable (82.5 m) with the anchor submerged and hanging free.

(5) Windlasses are to be fitted with cable lifter brakes of capacities sufficient to stop the anchor and the chain cable when paying out the chain cable. Such brakes are to produce torques capable of withstanding the following loads without any permanent deformation of strength members and without brake slip.

(a) with a chain cable stopper: 0.45 × the breaking test load of chain cable

(b) without a chain cable stopper: 0.80 × the breaking test load of chain cable

(6) Chain cable stoppers, if fitted, along with their attachments are to be designed to withstand, without any permanent deformation, 80% of the specified minimum breaking strength of the chain cable.

(7) Hull supporting structures of windlasses and chain cable stoppers are to be according to the following requirements:

(a) Hull supporting structures of windlasses and chain cable stoppers are to comply with the requirements specified in **14.3.1.5, Part 1, Part C** or **Chapter 23, Part CS**.

(b) For those ships of 80 m or more in length L_C that are specified in **1.4.3.1-1, Part 1, Part C**, all windlass mounts on an exposed deck over the forward 0.25 L_C line are to be of sufficient strength in cases where the height of the exposed deck in way of the item is mounted is less than 0.1 L_C or 22 m above the designed maximum load line, whichever is lesser.

(c) The strength of any above deck framing and hull structure supporting a windlass and its securing bolt is to be according to the requirements in **10.4.2.3, Part 1, Part C** or **10.6.1, Part CS**.

3 Hydraulic systems where employed for driving windlasses are to comply with the requirements specified in other chapters of this Part in addition to those in this chapter.

4 Electrical systems (e.g., electric motors and electrical circuits) are to comply with the requirements specified in **Part H** in addition to those in this chapter.

5 The following protections are to be provided:

(1) Suitable protection systems which limit the speed and torque of the prime mover to protect mechanical parts, including component housings such as over pressure protection devices, slipping clutches between electric motors and gearing, torque limiting devices (for electrically driven windlasses only)

(2) Means to contain debris consequent to any severe damage to the prime mover due to over-speed in the event of uncontrolled rendering of the cable; for example, covers, particularly when an axial-piston-type hydraulic motor is used as the prime mover

(3) Devices or parts necessary for the safety of users; for example, covers for any exposed gearing, covers for any hot surfaces of steam cylinders, etc.

6 Windlasses are to be fitted with couplings which are capable of disengaging between cable lifters and drive shafts. Hydraulically or electrically operated couplings are to be capable of being disengaged manually.

7 Windlasses are to be permanently marked with the following information:

(1) Nominal size of chain cable; for example, 100/3/45 means the nominal diameter of 100 mm and grade 3, with a holding load of 45% of the breaking test load.

(2) Maximum anchorage depth (m)

16.2.5 Tests

1 Shop Tests

For conformance with approved plans, windlasses are to be inspected during fabrication at the manufacturer facilities and acceptance tests, as specified in the specified standard of compliance and including at least **(1)** to **(4)** below, are to be carried out in the presence of the Surveyor.

(1) Pressure tests

Before assembly hydrostatic pressure tests are to be carried out for the following components in accordance with the requirements in **12.6.1**. The test pressure is to be 1.5 times design pressure. However, the test pressure for steam cylinders may be 1.5 times working pressure.

(a) Housings with covers for hydraulic motors and pumps

(b) Hydraulic pipes

(c) Valves and fittings

(d) Pressure vessels

(e) Steam cylinders

(2) No-load tests

Windlasses are to be run without loads at nominal speed for 15 *minutes* in each direction (for a total of 30 *minutes*). In cases where the windlass is provided with a gear change, additional runs in each direction for 5 *minutes* at each gear change are required.

(3) Load tests

Windlasses are to be tested to verify that the continuous duty pull, overload capacity and hoisting speed as specified in **16.2.4-2** can be attained. In case where the manufacturing works does not have adequate facilities, the following **(a)** or **(b)** may be complied instead of the load tests:

(a) Submission of the documents specified in **16.2.2(2)(c)**.

(b) To carry out the load tests, including adjustments of overload protection, on board ship. In such cases, functional testing at manufacturer works is to be performed under no-load conditions.

(4) Cable lifter brake capacity tests

It is to be verified that the holding power of the cable lifter brake complies with **16.2.4-2(5)** either through testing or submission of the calculation sheet specified in **16.2.2(2)(d)**.

2 Tests after installation on board

Required tests in **2.1.7-7, Part B** for windlasses are to be carried out during sea trials.

16.3 Mooring Winches

16.3.1 Structure, etc.

1 Mooring winches are to comply with Japanese Industrial Standards or any other recognized standards.

2 Mooring winches and their beds as well as any other accessories and facilities are to be installed effectively and securely onto the deck.

3 Mounts of mooring winches which are integrated with windlasses are to be in accordance with the requirements in **16.2.4-2(7)(b)** and **(c)**.

16.3.2 Tests

All mooring winches are to be subjected to the following tests after their installation on board.

(1) Confirmation tests for abnormalities with all mooring winches being operated for 15 *minutes* in each direction at maximum speed under no loads.

(2) Functioning tests for confirming the proper operation of drum brakes.

(3) Notwithstanding the requirements specified in **(1)** and **(2)** above, in cases where there are multiple units of the same type, the period of testing and number of units to be tested may be reduced.

16.3.3 Selection of Mooring Winches

Selection of mooring winches is to be in accordance with **14.4.4.3-1, Part 1, Part C**.

Chapter 17 REFRIGERATING MACHINERY AND CONTROLLED ATMOSPHERE SYSTEMS

17.1 General

17.1.1 Scope*

1 The requirements in this chapter apply to refrigerating machinery using the primary refrigerants listed below and those forming refrigerating cycles used for refrigeration, air conditioning, etc., as well as any controlled atmosphere systems for cargo holds. However, any refrigerating machinery with compressors of 7.5 *kW* or less and any refrigerating machinery using primary refrigerants other than those listed below are to be as deemed appropriate by the Society.

*R*134*a* : CH_2FCF_3

*R*404*A* : *R*125/*R*143*a*/*R*134*a* (44/52/4 *wt*%) CHF_2CF_3 / CH_3CF_3 / CH_2FCF_3

*R*407*C* : *R*32/*R*125/*R*134*a* (23/25/52 *wt*%) CH_2F_2 / CHF_2CF_3 / CH_2FCF_3

*R*407*H* : *R*32/*R*125/*R*134*a* (32.5/15/52.5 *wt*%) CH_2F_2 / CHF_2CF_3 / CH_2FCF_3

*R*410*A* : *R*32/*R*125 (50/50 *wt*%) CH_2F_2 / CHF_2CF_3

*R*449*A* : *R*32/*R*125/*R*1234*yf*/*R*134*a* (24.3/24.7/25.7/25.3 *wt*%) CH_2F_2 / CHF_2CF_3 / CF_3CF=CH_2/CH_2FCF_3

*R*507*A* : *R*125/*R*143*a* (50/50 *wt*%) CHF_2CF_3 / CH_3CF_3

2 For those items mentioned in this Chapter, the requirements given in this Chapter are applied in lieu of the requirements in **Chapters 10**, **12** and **13**.

17.1.2 Drawings and Data*

Drawings and data to be submitted for approval are generally as follows:

(1) Drawings (with materials, scantlings, type, design pressure, design temperature, etc. of pipes, valves, etc.)
  (a) Piping diagrams of refrigerating systems for provision chamber and air conditioning installations
  (b) Drawings of pressure vessels exposed to primary refrigerant pressure
  (c) Drawings of controlled atmosphere systems
  (d) Other drawings considered necessary by the Society

(2) Data
  (a) Particulars of refrigerating machinery
  (b) Specifications of controlled atmosphere systems
  (c) Other drawings considered necessary by the Society

17.2 Design of Refrigerating Machinery

17.2.1 General

The design pressure of pressure vessels and piping systems and the class of pipes used for refrigerating machinery are to be as follows:

(1) The design pressure of the pressure vessels and piping systems used for the refrigerating machinery and exposed to the pressure of the refrigerant is not to be less than the pressure in **Table D17.1** depending on the kind of refrigerant.

(2) Pipes for the refrigerants specified in **Table D17.1** are to be classified into Group III.

17.2.2 Location

Refrigerating machinery compartments are to be provided with efficient arrangements for drainage and ventilation, and they are to be separated from any adjacent refrigerated chambers by gastight bulkheads.

17.2.3 Materials

1 Materials used for refrigerating machinery are to be suitable for the refrigerant used, with respect to design pressure, minimum working temperature, etc.

2 Materials used for primary refrigerant pipes, valves and their fittings are to comply with the requirements in **12.1.4** to **12.1.6**

according to the classes of pipes specified in **17.2.1(2)**.

3 Materials used for pressure vessels exposed to refrigerant pressure (condensers, receivers and other pressure vessels) are to comply with the requirements in **10.2** according to their respective pressure vessel classifications as specified in **10.1.3**.

4 The following materials are not to be used for any parts of refrigerating machinery.

(1) Aluminum alloys containing over 2% magnesium for any parts coming in contact with primary refrigerants.

(2) Pure aluminum less than 99.7% for any parts that usually come in contact with water and are without any corrosion protection.

5 The service limitations of valves made of iron castings are shown in **Table D17.2**. Although the utilization of iron castings is permitted by the Table, they are not to be used for valves in piping that has a design temperature below 0℃ or exceeding 220℃. However, in cases where the normal working pressure of the piping does not exceeding 1/2.5 times the design pressure, the temperature limitations may be lowered to -50℃.

Table D17.1 Design Pressure of Pressure Vessels and Piping Systems for Refrigerating Machinery

Refrigerants	High Pressure Side(1) (*MPa*)	Low Pressure Side(2) (*MPa*)
*R*134*a*	1.4	1.1
*R*404*A*	2.5	2.0
*R*407*C*	2.4	1.9
*R*407*H*	2.5	2.0
*R*410*A*	3.3	2.6
*R*449*A*	2.6	2.0
*R*507*A*	2.5	2.0

Notes:

(1) High Pressure side: The pressure part from the compressor delivery side to the expansion valve.

(2) Low Pressure side: The pressure part from the expansion valve to the compressor suction valve. In cases where a multistage compression system is adopted, the pressure part from the lower-stage delivery side to the higher-stage suction side is to be included.

Table D17.2 Service Limitation of Valves Made of Iron Castings

Kind of valves	Materials	Application
Stop valves	Gray iron castings with specified tensile strength not exceeding 200 N/mm^2 or the equivalent thereto	Not to be used
	Gray iron castings other than those specified above, spheroidal graphite iron castings, malleable iron castings or the equivalent thereto	(1) May be used for design pressures not exceeding 1.6 *MPa* (2) May be used for design pressures exceeding 1.6 *MPa*, but not exceeding 2.6 *MPa*, provided that the nominal diameter does not exceed 100 *mm* and the design temperature is 150℃ or below.
Relief valves	Any iron castings	Not to be used
Automatic control valves	Gray iron castings with a specified tensile strength not exceeding 200 N/mm^2 or the equivalent thereto	Not to be used
	Gray iron castings other than those specified above or the equivalent thereto	(1) May be used for design pressures not exceeding 1.6 *MPa* (2) May be used for design pressures exceeding 1.6 *MPa*, but not exceeding 2.6 *MPa*, provided that the nominal diameter does not exceed 100 *mm* and the design temperature is 150℃ or below.
	Spheroidal graphite iron castings, malleable iron castings or the equivalent thereto	Not to be used for design pressures exceeding 3.2 *MPa*

17.2.4 Pressure Relief Devices

1 Compressors are to be provided with pressure relief devices as required by **(1)** and **(2)**.

(1) Relief valves are to be provided between compressor cylinders and gas delivery stop valves with discharge being led to suction side of the compressor. In cases where devices which automatically stop compressors when pressures on high pressure sides of refrigerant piping systems become excessively high are installed, and alarm systems that activate visible and audible alarms when such refrigerant piping systems are in operation are also installed in refrigerating machinery compartments and at monitoring positions to provide an equivalent level of safety, the following **(a)** or **(b)** may be applied.

(a) Discharge from the relief valves specified in **(1)** above is to be led to the open air and openings are to be located at safe places.

(b) The relief valves specified in **(1)** above may be omitted on the condition that caution plates are provided that state all stop valves between the cylinders and pressure vessels specified in **-2** below be kept in the open position before compressors are started.

(2) Notwithstanding the requirement in **(1)** above, compressors of 11 *kW* or less for refrigerating installations may be provided with pressure control switches instead of relief valves.

2 Relief valves are to be fitted to pressure vessels which may be isolated and store primary refrigerants in a liquid condition. Discharged gases from relief valves are to be released into the atmosphere at a safe place above the weather deck or to the low pressure parts of the equipment.

3 In cases where any discharged gases from relief valves on the high pressure parts of primary refrigerants are led to low pressure parts before being released into the atmosphere, the operation of relief valves is not to be interrupted by any back pressure accumulation.

4 Relief valves are to be provided for the cooling liquid side of condensers and the brine side of evaporators except in cases where the connected pump is so constructed that the pressure does not exceed design pressure.

17.3 Controlled Atmosphere Systems*

17.3.1 General

Controlled atmosphere zones and all relevant installations are to be arranged as follows;

(1) Each controlled atmosphere zone is to be made as airtight as possible, and is to be arranged to keep the internal pressure normal.
(2) Gas freeing systems are to be provided to discharge gas from each controlled atmosphere zone, and ventilators are to be provided for enclosed spaces adjacent to any controlled atmosphere zone.
(3) Entrances to controlled atmosphere zones are to be so constructed as to be capable of prevented from being easily opened due to error, etc.
(4) Fixed nitrogen generators are to be installed in a dedicated room, airtight from any adjacent spaces. This nitrogen generator room is to be fitted with an exhaust mechanical ventilation system of a sufficient capacity.
(5) Each controlled atmosphere zone is to be provided with a warning alarm which will be activated before the injection of any nitrogen into the controlled atmosphere zone.
(6) Fixed oxygen alarm devices are to be provided in fixed nitrogen gas generation rooms and in each enclosed space adjacent to a controlled atmosphere zone in order to warn each place in the event of low level oxygen content.
(7) A means of two-way communication is to be provided between controlled atmosphere zones and nitrogen release control stations. A suitable number of portable oxygen measuring instruments with alarms are to be provided on board for the safe entrance to any dangerous zone. Furthermore, medical first-aid equipment, which includes oxygen resuscitation equipment, is to be provided on board.

17.4 Tests

17.4.1 Shop Tests

Refrigerating machinery is to be tested as follows:

(1) Pressure vessels exposed to the pressure of primary refrigerants are to be subjected to a hydrostatic test at a pressure of 1.5 times the design pressure and a tightness test at a pressure equal to the design pressure.
(2) Cylinders and crank cases of the compressors of refrigerators are to be subjected to a hydrostatic test at a pressure of 1.5 times the design pressure and a tightness test at a pressure equal to the design pressure.

17.4.2 Tests after Installation On Board*

1 Piping systems which are exposed to the pressure of primary refrigerants are to be subjected to a leak test at a pressure of 90% of the design pressure, after they are installed on board.

2 All installations and equipment connected with controlled atmosphere systems are to be tested by operation tests, etc. to verify that they operate normally.

Chapter 18 AUTOMATIC AND REMOTE CONTROL

18.1 General

18.1.1 Scope*

1 The requirements in this Chapter apply to automatic or remote control systems which are used to control the following machinery and equipment:

(1) Main propulsion machinery (in this Chapter, propulsion generating sets in electric propulsion ships are excluded),

(2) Controllable pitch propeller

(3) Steam generating sets

(4) Electric generating sets (in this Chapter, propulsion generating sets in electric propulsion ships are included)

(5) Auxiliary machinery associated with the machinery and equipment listed in **(1)** to **(4)**

(6) Fuel oil systems

(7) Bilge systems

(8) Deck machinery

2 In cases where considered necessary by the Society, the requirements in this Chapter are correspondingly applied to those automatic or remote control systems which are used for controlling machinery and equipment not listed in **-1(1)** to **(8)**.

18.1.2 Terminology

Terms used in this Chapter are defined as follows:

(1) A monitoring station (excluding control stations) is defined as a position where measuring instruments, indicators, alarms, etc. for machinery and equipment are centralized and all information necessary to grasp the operating condition of them can be obtained. However, in cases where a monitoring station is provided with the ship in addition to the control station mentioned in **(2)** below, the requirements of the Rules relating to monitoring stations do not apply to the monitoring station concerned.

(2) A control station is defined as a position which can function as a monitoring station and from which machinery and equipment can be controlled.

(3) A main control station is defined as a control station, provided with equipment necessary and sufficient to control the main propulsion machinery (this equipment will be referred to as “main control equipment” in **(3)** and **(4)**) and from which the main propulsion machinery is normally controlled, of those ships which provide main control equipment at a position outside of the navigation bridge.

(4) The main control station on the bridge is defined as a navigation bridge of the ship which provides main control equipment at the navigation bridge and the location the main propulsion machinery is normally controlled.

(5) A sub-control station is defined as a control station at which the main propulsion machinery is capable of being controlled, except for those local control stations for main propulsion machinery that are provided in the machinery spaces of the ship that has a main control station on its bridge.

(6) Bridge control devices are defined as remote control devices for main propulsion machinery or controllable pitch propellers provided on a navigation bridge or at the main control station on the bridge.

(7) Sequential control is defined as a pattern of control that can be carried out automatically in a predetermined sequence.

(8) Program control is defined as a pattern of control in which desired values can be changed in a predetermined schedule.

(9) Local control is defined as direct manual control of machinery and equipment performed at or near their locations, receiving necessary information from the measuring instruments, indicators and so on.

(10) A safety system is defined as a system which operates automatically, in order to prevent damage to machinery and equipment in cases where serious impediments to functioning should occur during their operation so that one of the following actions will take place:

  (a) Starting of stand-by machinery or equipment

  (b) Reduction of output of machinery or equipment

  (c) Shutting off fuel or power supplies, thereby stopping the machinery or equipment

18.1.3 Drawings and Data*

Drawings and data to be submitted are generally, as follows.

(1) Drawings and data for approval
 (a) Drawings and data concerning automation
  i) List of measuring points
  ii) List of alarm points
  iii) Control devices and safety devices
   1) List of controlled objects and controlled variables
   2) Kinds of control energy sources (self-actuated, pneumatic, electric, etc.)
   3) List of conditions for emergency stopping, speed reduction (automatic or demand for reduction), etc.
 (b) The following drawings and data for the automatic control devices and remote control devices for main propulsion machinery or controllable pitch propellers
  i) Operating instructions of main propulsion machinery such as starting and stopping, change-over of direction of revolution, increase and decreased of output, etc.
  ii) Arrangements of safety devices (including those attached to engines) and pilot lamps
  iii) Controlling diagrams
 (c) Following drawings and data for the automatic control devices and remote control devices for boilers:
  i) Operating instructions of sequential control, feed water control, pressure control, combustion control and safety devices
  ii) Diagrams for automatic combustion control devices and automatic feed water control devices
 (d) Diagrams and operating instructions for automatic control devices for electric generating sets (automatic load sharing devices, preference tripping devices, automatic starting devices, automatic synchronous making devices, sequential starting devices, etc.)
 (e) Panel arrangements of monitoring panels, alarm panels and control stands at respective control stations
 (f) Other drawings and data deemed necessary by the Society.

(2) Drawings and data for reference
 Other drawings and data deemed necessary by the Society.

18.2 System Design

18.2.1 System Design

1 Control systems, alarm systems and safety systems are to be so designed that one fault does not result in any other faults as far as practicable and the extent of any damage is kept to a minimum.

2 Control systems, alarm systems and safety systems are to be designed on the fail-to-safe principle. The characteristic of fail-safe is to be evaluated on the basis not only of the respective systems themselves and all associated machinery and equipment, but also on the total safety of the ship.

3 Automatic or remote control systems are to be sufficiently reliable under service conditions.

4 Signal cables are to be installed in such a manner that any harmful induced interference can be avoided.

18.2.2 Supply of Power

1 Supply of electrical power

The supply of electrical power is to be in accordance with the following:

(1) Electrical supply circuits to control systems, alarm systems and safety systems are not to branch off from power circuits and lighting circuits, except in cases where the electrical power to control systems, alarm systems and safety systems is supplied from power circuits to the machinery and equipment they serve.

(2) Electrical power to alarm systems and safety systems for electric generating sets is also to be supplied from an accumulator battery.

2 Supply of oil pressure

The supply of control oil pressure is to be in accordance with the following:

(1) Sources of oil pressure are to be capable of stably supplying all necessary pressure and quantities of purified oil.
(2) Overpressure preventive devices are to be provided for the delivery side of oil pressure pumps.
(3) Two or more sets of oil pressure pumps for control of the main propulsion machinery and the main shafting are to be provided and they are to be so arranged that in cases where one of the pumps in operation fails, stand-by pump(s) either start automatically or be readily started by remote control. In this case, oil pressure pumps are not to be used for the control of machinery and equipment other than the main propulsion machinery and the main shafting.

3 Supply of pneumatic pressure

The supply of control air is to be in accordance with the following:

(1) Control systems are to be provided with an air reservoir having a capacity capable of supplying air to control devices for at least 5 *minutes* in the event of the failure of the control air compressor.
(2) In cases where starting air reservoirs of reciprocating internal combustion engines used as main propulsion machinery are used as control air reservoirs, pressure reducing valves are to be duplicated or a spare pressure reducing valve is to be provided on board.
(3) There are to be two or more sets of air compressors which may be used as a source of control air. Each air compressor is to have redundant capacity even in the event of failure of either one of them.
(4) Control air is to pass through a filter and, if necessary, a drier so that any solids, oil and water may be removed as much as possible.
(5) Control air pipes are to be independent of general service air pipes and starting air pipes.

18.2.3 Environmental Conditions

Automatic or remote control systems are to be capable of withstanding the environmental conditions of the places where they are installed.

18.2.4 Control Systems

1 Independence of control systems

Control systems for main propulsion machinery or controllable pitch propellers, boilers, electric generating sets and auxiliary machinery essential for main propulsion of the ship are to be independent each other. However, when the propulsion generator plant and the main generating plant are connected to the same bus line, they may have common control systems.

2 Interconnection devices

In the case of multiple main propulsion machinery or controllable pitch propellers, electric generating sets, or auxiliary machinery (excluding auxiliary machinery for specific use, etc.) in which these multiple units are designed to be operated simultaneously under the same conditions, interconnection devices may be provided between the control devices of these installations.

3 Control characteristics

Remote control devices and automatic control devices are to have control characteristics that conform to the dynamic properties of the machinery and equipment they serve and consideration is to be given so that no malfunctions and hunting occurs due to any external disturbances.

4 Interlocks

Control devices are to be provided with suitable interlocking arrangements in order to prevent any damage to machinery and equipment due to anticipated malfunctions and mal-operation of such machinery and equipment.

5 Change-over to manual operation

Change-over to manual operation is to comply with the following requirements:

(1) Main propulsion machinery or controllable pitch propellers, boilers, electric generating sets and auxiliary machinery essential for main propulsion of the ship are to be so arranged as to be manually started, operated and controlled even in the event where automatic control devices becomes inoperative.
(2) Automatic control devices are generally to be provided with provisions for the manual stopping of the automatic functions of these devices.
(3) The provisions specified in **(2)** are to be capable of stopping the automatic functions of automatic control devices, even in cases where any part of the automatic control device becomes inoperative.

6 Cancellation of remote control functions

For remote control devices, remote control function is to be capable of being manually cancelled.

7 Indication of control locations

In cases where machinery and equipment are capable of being operated from more than one station, the following requirements in **(1)** and **(2)** are to be complied with. However, this requirement need not be complied with in cases where the safety of the machinery and equipment and the safety at the time of maintenance work can be guaranteed by other means considered appropriate by the Society.

(1) At each control station there is to be an indicator showing which station is in control of the machinery and equipment.

(2) Control of the machinery and equipment is to be possible only from one station at a time.

18.2.5 Alarm Systems

1 The function of alarm systems is to comply with the following requirements:

(1) In cases where an abnormal condition is detected, devices to issue a visual and audible alarm (hereinafter referred to as “alarm devices” in this Part) are to operate.

(2) In cases where arrangements are made to silence audible alarms they are not to turn off visual alarms.

(3) Two or more faults are to be indicated at the same time.

(4) Audible alarms for machinery and equipment are to be clearly distinguishable from other audible alarms such as general alarms, fire alarms, CO_2 flooding alarms, etc.

2 The function of the alarm systems provided in monitoring station for main propulsion machinery or controllable pitch propellers is to comply with the following requirements, in addition to those requirements in **1** above:

(1) Visual alarms are to remain on until all faults have been corrected.

(2) The acceptance of any one alarm is not to inhibit the activation of any other alarm.

(3) If an alarm has been activated and a second fault occurs prior to the first one being corrected, alarm devices go into operation again.

(4) In cases where an alarm has been manually stopped, clear indication of details of this stoppage is to be given.

3 Visual alarms are to be so arranged that each abnormal condition of machinery and equipment is readily distinguishable and that acknowledgement is clearly noticeable.

18.2.6 Safety Systems*

1 Independence of safety systems

Independence of safety systems is to comply with the following requirements:

(1) Safety systems are to be, as far as practicable, provided independently of control systems and alarm systems.

(2) Safety systems for main propulsion machinery, boilers, electric generating sets and auxiliary machinery essential for main propulsion of the ship are to be independent each other.

2 Function of safety systems

The function of the safety systems is to comply with the following requirements:

(1) Alarm systems which have functions prescribed in **18.2.5** are to operate when safety systems are put into action.

(2) In cases where safety systems are put into action and the operation of machinery or equipment has been stopped, the machinery or equipment is not to automatically restart before manual reset is made.

3 Override arrangements

In cases where arrangements are provided for overriding a safety system, the following requirements **(1)** and **(2)** are to be complied with:

(1) Visual indication is to be given at the relevant control stations of machinery and equipment when an override is operated.

(2) Override arrangements are to be such that any inadvertent operation is prevented.

18.2.7 Use of Computers*

1 The reliability and maintainability of computer based systems are not to be inferior to those of systems not relying upon computers.

2 Control systems, alarm systems and safety systems which constitute computer based systems are to comply with the following **(1)** to **(3)**:

(1) Requirements for computers

(a) The composition of computers is to be so planned that the extent of impact on the system as a whole of any failure in any part of a circuit or component is to be minimized as far as possible.

(b) Each component is to be protected against any possibility of overvoltage (electronic noise) which may originate from input or output terminals.

(c) Central processing units and important peripheral devices are to have self-monitoring functions.

(d) Important programs and data are to be ensured against loss in cases where an external electrical power supply may be temporarily interrupted.

(e) Computers are to be set up so they can be quickly re-started following planned procedures within a short period of time after electrical power has been restored after a power failure.

(f) Spare parts for all important elements which require special techniques for repair work are to be kept in ample supply for easy replacement.

(g) Change-over to back-up means is to be able to be performed easily and soundly.

(2) Back-up means

(a) In cases where one computer simultaneously performs fuel control (governor control, electronic injection control, etc.) and remote control of main propulsion machinery of ships in which reciprocating internal combustion engines, steam turbines or gas turbines are used as main propulsion machinery (excluding electric propulsion ships) or in cases where one computer simultaneously performs output control (rotational speed control, load control, etc.) and remote control of main propulsion machinery in electric propulsion ships, one of the following systems is to be provided in the case of computer failure. However, where this requirement is impracticable, relevant systems are to comply with requirements deemed appropriate by the Society.

i) Stand-by computer

ii) Governor-controlled back-up systems operated at the main control station

(b) Safety systems are to be provided with back-up means which can be used in a timely manner in the event of the failure of the computer in service.

i) Stand-by computer

ii) Safety systems that do not rely on computers

(c) In cases where visual display units (VDU) are adopted as indicators for the alarm systems stipulated in this chapter, at least two VDUs are to be installed, or other arrangements deemed appropriate by the Society are to be considered.

(3) Independence

Independence of computerized control systems and safety systems is to comply with the requirements in **18.2.4-1** and **18.2.6-1** respectively, except in cases where their constitution are comply with requirements specified below.

(a) In cases where secondary control systems or stand-by computers are installed for those control systems, the independence of such control systems may not be required for individual machinery or equipment. In such cases, local control equipment fitted to main propulsion machinery in accordance with the requirements given in **18.3.2-3(2)** are not regarded as secondary control systems.

(b) In cases where safety systems conform to the requirement given in **(2)(b)** above, the independence of individual machinery and equipment in systems, and their independence from other systems may not be required, notwithstanding the requirements in **18.2.6-1**.

(c) In cases where secondary systems or stand-by computers are installed in both control systems and safety systems, the independence of individual machinery and equipment in their systems including alarm systems, and their independence from the other systems may not be required.

18.3 Automatic and Remote Control of Main Propulsion Machinery or Controllable Pitch Propellers

18.3.1 General*

The devices for remote or automatic control by which the main propulsion machinery or controllable pitch propellers are controlled are to comply with the requirements in this **18.3**.

18.3.2 Remote Control Devices for Main Propulsion Machinery or Controllable Pitch Propellers*

1 General

Remote control devices for main propulsion machinery or controllable pitch propellers are to comply with the following requirements:

(1) Remote control devices for main propulsion machinery or controllable pitch propellers are to be capable of controlling the propeller speed and the direction of thrust (the blade angle of propellers in the case of controllable pitch propellers) by means of a simple operation.
(2) Remote control devices for main propulsion machinery or controllable pitch propellers are to be provided for each propeller. In cases where multiple propellers are designed to operate simultaneously, they may be controlled by one control device.
(3) In cases where the speed of the reciprocating internal combustion engines used as main propulsion machinery is controlled by governors, the governors are to be adjusted so that main propulsion machinery may not exceed 103 % of maximum continuous revolutions. These governors are to be capable of maintaining a safe minimum speed.
(4) In cases where a program control is adopted, programs for increasing and decreasing output is to be so designed that any undue mechanical stresses and thermal stresses does not occur in any parts of the machinery.
(5) In remote control stations or monitoring stations and at the manoeuvring platform for main propulsion machinery or controllable pitch propellers, the following instruments are to be provided:
(a) Indicators for propeller speed and direction of rotation in the case of solid propellers.
(b) Indicators for propeller speed and pitch position in the case of controllable pitch propeller.
(6) In remote control stations for main propulsion machinery or controllable pitch propellers, alarm devices necessary for the control of main propulsion machinery or controllable pitch propellers are to be provided.

2 Transfer of Control

Remote control devices for main propulsion machinery or controllable pitch propellers are to comply with the following requirements with respect to transfer of control:

(1) Each control station for main propulsion machinery or controllable pitch propellers is to be provided with means to indicate which of them is in control.
(2) Remote control of main propulsion machinery or controllable pitch propellers is to be only possible from one location at a time.
(3) Transfer of control is to be only possible with orders from the serving station and acknowledgement by the receiving station except for the following cases:
(a) Transfer of control between a local control station for main propulsion machinery or controllable pitch propellers and the main control station or sub-control station; and
(b) Transfer of control during a stoppage condition of the main propulsion machinery.
(4) In cases where the main propulsion machinery or controllable pitch propellers is controlled from the navigation bridge or the main control station on bridge, the transfer of control is to be possible from a local control station for main propulsion machinery or controllable pitch propellers to the main control station or the sub-control station even if no order of the transfer of control from the navigation bridge or the main control station on bridge has been given.
(5) Means are to be provided to prevent the propelling thrust from being significantly altered when control is transferred from one location to another.

3 Failure of remote control systems of main propulsion machinery or controllable pitch propellers

The following requirements are to be complied with in case of a failure of any of the remote control devices for main propulsion machinery or controllable pitch propellers:

(1) In remote control stations for main propulsion machinery or controllable pitch propellers, alarm devices which operate in the event of a failure of any of the remote control devices for main propulsion machinery or controllable pitch propellers are to be provided.
(2) In the event of a failure of any remote control devices for main propulsion machinery or controllable pitch propellers, the main propulsion machinery or the controllable pitch propellers are to be able to be locally controlled. It is also to be possible to control any of the auxiliary machinery, essential for the propulsion and safety of the ship, at or near the machinery concerned.
(3) In the event of a failure of any of the remote control devices for main propulsion machinery or controllable pitch propellers, the preset speed and direction of the propeller thrust are to be maintained until the control is in operation at the main control station, the sub-control station or the local control station for the main propulsion machinery or controllable pitch propellers, unless this is considered impracticable by the Society.
(4) In the event of a failure of any of the remote control devices for main propulsion machinery or controllable pitch propellers, transfer of control to the main control station, the sub-control station or the local control station for the main propulsion

machinery or controllable pitch propellers is to be possible by a simple operation.

(5) Remote control stations for main propulsion machinery or controllable pitch propellers are to be provided with independent emergency stopping devices for the main propulsion machinery, which are effective in the event of a failure of any of the remote control devices for the main propulsion machinery or the controllable pitch propellers.

4 Remote starting of main propulsion machinery in ships in which reciprocating internal combustion engines are used as main propulsion machinery (excluding electric propulsion ships)

Starting by means of remote control devices for main propulsion machinery is to comply with the following:

(1) The number of times of starting main propulsion machinery is to satisfy the number specified in **2.5.3**.

(2) Remote control devices for main propulsion machinery arranged to automatically start are to be so designed that the number of automatic consecutive attempts which fail to produce a start is limited to three *times*. In the event of a failure of starting, a visual and audible alarm is to be issued at the relevant control station as well as the main control station on bridge, the main control station or monitoring stations (in cases where a main control station on the bridge and a main control station are not provided) for the main propulsion machinery or the controllable pitch propellers.

(3) In cases where compressed air is used for starting the main propulsion machinery, alarm devices to indicate any low starting air pressure are to be provided at the remote control station and the monitoring station for the main propulsion machinery.

(4) The low starting air pressure mentioned in **(3)** for the operation of alarm devices is to be set at a level to permit further main propulsion machinery starting operations.

18.3.3 Bridge Control Devices*

Bridge control devices are to comply with the following **(1)** through **(6)** as well as requirements in **18.3.2**.

(1) Even in cases where main propulsion machinery or controllable pitch propellers is controlled from the navigation bridge or the main control station on the bridge, telegraphed orders from the navigation bridge or the main control station on the bridge are to be indicated in the main or sub-control stations respectively and at any manoeuvring platforms which are capable of controlling main propulsion machinery or controllable pitch propellers.

 (a) Sub-control stations or local control stations for main propulsion machinery or controllable pitch propellers for ships provided with a main control station on bridge; or

 (b) Main control stations for ships not provided with main control station on bridge.

(2) Bridge control devices are to be provided with either one of the following devices in order to prevent any prolonged running of main propulsion machinery in its critical speed range:

 (a) Devices to make main propulsion machinery pass automatically and rapidly through its critical speed range:

 (b) Alarm devices which operate in cases where the main propulsion machinery that is operating exceeds a predetermined period in its critical speed range.

(3) Bridge control devices are to be provided with visual and audible alarms which give the officer in charge of the navigational watch enough time to assess navigational circumstances in an emergency before the safety systems of main propulsion machinery specified in **18.1.2(10)(b)** or **(c)** go into effect, except in cases in which total failure of main propulsion machinery will occur within a short period of time.

(4) Bridge control devices are to be provided with the override arrangement specified in **18.2.6-3** for the following safety systems of main propulsion machinery:

 (a) Safety systems which perform as specified in **18.1.2(10)(b)**

 (b) Safety systems which perform as specified in **18.1.2(10)(c)**, except in cases in which total failure of main propulsion machinery will occur within a short period of time.

(5) Operations following any setting of the bridge control device including reversing from the maximum ahead service speed in case of emergency are to take place in an automatic sequence and with time intervals acceptable to the machinery.

(6) For steam turbines, a slow-turning device is to be provided which operates automatically or manually to prevent any risk of rotor distortion due to propulsion turbines being stopped for long periods of time. Discontinuation of this automatic turning from the bridge must be possible.

18.3.4 Safety Measures*

1 Safety measures for main propulsion machinery or controllable pitch propellers

Safety measures for main propulsion machinery or controllable pitch propellers are to comply with the following requirements:

(1) The following safety measures are to be taken regarding remote control devices for main propulsion machinery or controllable pitch propellers:
   (a) Necessary interlocking devices are to be provided to prevent any serious damage due to operational error.
   (b) In cases where any auxiliary machinery essential for the main propulsion of the ship are driven by electric motors, the main propulsion machinery is to be so designed as to automatically stop in the event of a failure of the main source of electrical power or it is to be capable of being stopped.
   (c) Main propulsion machinery is to be so arranged as to not restart automatically when electrical power is restored after a failure of the main source of electrical power whereas the main propulsion machinery was stopped.
   (d) Remote control devices for main propulsion machinery or controllable pitch propellers are to be so designed that the engine may not be abnormally overloaded in the event of any failure of them.
(2) Stopping devices for main propulsion machinery are to be provided at monitoring stations for main propulsion machinery or controllable pitch propellers.
(3) With respect to safety measures for main propulsion machinery driven by reciprocating internal combustion engines, the requirements specified in **2.4.5-1** are to be applied.

2 Safety systems of main propulsion machinery

Safety systems of main propulsion machinery are to comply with the following requirements:

(1) Devices to shut off the fuel or steam supply (this device hereinafter being referred to as a “safety device”) for main propulsion machinery are not to be automatically activated except in cases which could lead to complete breakdown, serious damage or explosion.
(2) Safety systems for the main propulsion machinery are to be so designed as to not lose their function or fail-safe capability, even in the event of a failure of their main electrical source or their air source.

3 Self-reversing reciprocating internal combustion engines

Remote control devices for self-reversing reciprocating internal combustion engines are to be at least provided with the following safety measures:

(1) Starting operations are to be only possible when the camshaft is definitely at the position of “Ahead” or “Astern”.
(2) Fuel is not to be injected during reversing operations.
(3) Reversing operations are to be conducted after the “Ahead” revolution is reduced to a predetermined value.

4 Main propulsion machinery of multi-engines coupled to a single shaft ship.

Remote control devices for multi-engines coupled to a single shaft are to be at least provided with the following safety measures:

(1) Each main propulsion machinery is to be provided with overload preventive devices.
(2) Each main propulsion machinery is not to be subjected to abnormally unbalanced loads.

5 Main propulsion machinery with clutches

Remote control devices for engines with clutches are to be at least provided with the following safety measures:

(1) Clutches equipped to main propulsion machinery in multi-engines coupled to single shafts are to be disengaged when the main propulsion machinery is stopped in an emergency. While multi-engines are operating in different directions of rotation, their clutches are not to be engaged simultaneously.
(2) Engaging and disengaging of clutches are to be carried out below a predetermined number of revolutions of the main propulsion machinery.
(3) Overspeed protective devices specified in **2.4.1-2**, **3.3.1-1** or **4.3.1-1** are to be provided.
(4) In cases where there is fear that the speed of a propulsion motor would exceed 125% of the rated revolutions when the clutch is disengaged, overspeed protective devices, deemed appropriate by the Society, are to be provided.

6 Main propulsion machinery driving controllable pitch propellers

Remote control devices for engines driving controllable pitch propellers are to be at least provided with the following safety measures:

(1) Overload preventive devices are to be provided.
(2) Starting of engines or engaging of clutches is to be performed while the propeller blades are in a neutral position.
(3) Overspeed protective devices as specified in **2.4.1-2**, **3.3.1-1** or **4.3.1-1** are to be provided.
(4) In cases where there is fear that the speed of the propulsion motor would exceed 125% of the rated revolutions when the

propeller pitch is altered, overspeed protective devices, deemed appropriate by the Society, are to be provided.

18.4 Automatic and Remote Control of Boilers

18.4.1 General

1 Automatic control systems for both combustion and feed water of oil-fired, dual-fuel-fired, gas-fired and multi-fuel-fired boilers are to comply with the requirements in **18.4.2** to **18.4.5** respectively.

2 Automatic control systems for either combustion or feed water of oil-fired, dual-fuel-fired, gas-fired and multi-fuel-fired boilers are to comply with the relevant requirements in **18.4.2** or **18.4.3** as well as the requirements in **18.4.4** and **18.4.5**.

3 Automatic control of boilers other than oil-fired, dual-fuel-fired, gas-fired and multi-fuel-fired boilers or those having special features is to be deemed appropriate by the Society.

4 In cases where boilers are remotely controlled, control devices and monitoring devices necessary for the operation of such boilers are to be provided at all relevant control stations.

5 Remote water level indicators are to comply with the requirements in **9.9.8**.

18.4.2 Automatic Combustion Control Systems

1 General

Automatic combustion control systems are to comply with the following requirements:

(1) Automatic combustion control systems are to be able to obtain planned steam amount, steam pressure and steam temperature as well as be able to secure stable combustion.

(2) Devices to control the fuel supply to adjust according to the load imposed and are to be capable of ensuring stable combustion in the controllable range of fuel supply.

(3) In cases where combustion control is carried out according to the pressure of the boiler, the upper limit of this pressure is to be lower than the set pressure of the safety valves.

2 Combustion control devices for intermittent operation

The combustion control devices for intermittent operation are to comply with the following requirements and they are to operate according to a planned sequence:

(1) Before ignition of the pilot burner or before ignition of the main burner if a pilot burner is not fitted, the combustion chamber and the flue are to be prepurged by air of not less than 4 times the volume of the combustion chamber and the flue up to the boiler uptake. For small boilers with only one burner, a prepurge for not less than 30 *seconds* will be accepted.

(2) In the case of direct ignition, a method of ignition in which the main burner is fired by ignition spark, the opening of the fuel valve is not to precede the ignition spark.

(3) In the case of indirect ignition, a method of ignition in which the main burner is fired by a pilot burner, the opening of the fuel valve for the pilot burner (hereinafter referred to as "ignition fuel valve" in this part) is not to precede the ignition spark, and the opening of the fuel valve for the main burner (hereinafter referred to as "main fuel valve" in this part) is not to precede the opening of the ignition fuel valve.

(4) Firing is to definitely be carried out within the planned period. If the firing of the main burner has failed, main fuel valves are to be so designed as to close after being opened within 10 *seconds* in the case of direct ignition and 15 *seconds* in the case of indirect ignition.

(5) Firing on the main burners is to be carried out at their low firing position.

(6) After closure of the main fuel valve, post-purge is to be carried out for not less than 20 *seconds* to ensure an adequate supply of air to completely burn off all remaining fuel oil between the fuel oil valve and the burner nozzle. Auxiliary boilers need not to be comply with this requirement in cases where deemed appropriate by the Society.

3 Combustion control devices for controlling the number of firing burners

The combustion control devices for controlling the number of firing burners are to comply with the following requirements:

(1) Each burner is to be fired and extinguished according to a planned sequence. However, the base burner may be fired by manual operation and other burners may be fired by a flame from burner(s) already lit.

(2) Any remaining fuel in extinguished burners is to be automatically burnt up in order not to interfere with any restarting of the burner. However, while the pilot burner is not ignited, any remaining fuel in the base burner is not to be removed by steam or

air when it is in place.

(3) The burners for main boilers are to be capable of being fired and extinguished from main control stations or the main control station on the bridge, except for the firing of base burner.

4 Other combustion control devices

Other combustion control devices are to be deemed appropriate by the Society. They are also to comply with the relevant requirements in **-2** and **-3**.

18.4.3 Automatic Feed Water Control Devices

1 Automatic feed water control devices are to be capable of automatically controlling the feed water in order to maintain a water level in the boilers within a predetermined range.

2 Main boilers are to be provided with not less than three water level detectors used for a feed water control devices, remote water level indicators, low-water level safety devices and low-water level alarm devices.

18.4.4 Safety Measures

1 Safety devices

Safety devices are to comply with the requirements in **9.9.10-1**.

2 Heating of fuel oil

In cases where heated fuel oil is used, an automatic temperature control device is to be provided for the heater; and, the boiler is to be provided with a device to automatically shut off the fuel supply to the burners or an alarm device which operates when the temperature of fuel oil falls below a predetermined value.

18.4.5 Alarms

Alarm devices are to comply with the requirements in **9.9.10-2**.

18.5 Automatic and Remote Control of Electric Generating Sets

18.5.1 General*

1 Electric generating sets arranged to be automatically or remotely started are to be provided with interlocking devices necessary for safe operation.

2 Electric generating sets (other than those used as emergency sources of electrical power) arranged to be automatically started are to be so designed that the number of automatic consecutive attempts which fail to produce a start is limited to two times; and, they are to be provided with an alarm device which operates at times of starting failure.

3 In cases where reciprocating internal combustion engines used to drive propulsion generators are remotely started, the number of starts is to conform to the required number specified in **2.5.3**.

4 In cases where the automatic starting of stand-by generating sets with automatic connections to switchboard busbars is provided, automatic closure on to the busbars is to be limited to one attempt in the event of any original power failure being caused by a short circuit.

5 Automatic control and remote control systems for electric generating sets, whose generators are driven by the main propulsion machinery; which supplies electrical power to electrical installations relating to the services specified in **3.1.2(1), Part H**; and, which is operated while the main propulsion machinery is being controlled by bridge control devices, are to comply with the requirements in **3.2.1, Part H**, in addition to those in this **18.5**.

6 With respect to safety measures for electric generating set driven by reciprocating internal combustion engines, the requirements specified in **2.4.5-1** are to be applied.

18.5.2 Emergency Source of Electric Power

Automatic or remote control devices for reciprocating internal combustion engines driving emergency generators are to comply with the following requirements:

(1) Alarm devices, to be activated in the event of any of the abnormal conditions given in **Table D18.2**, are to be provided.

(2) Devices referred to in **(1)** are to provide alarms at both local and navigation bridge. Visual alarms at navigation bridge may be of group indication.

(3) Each reciprocating internal combustion engine with a maximum continuous output of 220 kW or over is to be provided with an overspeed protective device specified in **2.4.1-4**.

(4) When devices, other than overspeed protective devices, are provided to shutdown reciprocating internal combustion engines, means are to be provided to override those devices automatically during navigation.

(5) The silencing of the audible alarms from navigation bridge is not to cause the silencing of the audible alarms at local positions.

Table D18.2 Alarms for Reciprocating Internal Combustion Engines to Drive Emergency Generators

Monitored Variables		Alarms	Remarks
Temperature	L.O. inlet	H	Applicable to engines with maximum continuous output of 220 kW or over.
	Cooling water or air outlet	H	
Pressure	L.O. inlet	L	
	Cooling water inlet	L	Applicable to engines with maximum continuous output of 220 kW or over. Low flow may be accepted.
Others	F.O. Leakage from high pressure pipes	○	Fuel injection pipes and common rails
	Overspeed	○	Applicable to engines with maximum continuous output of 220 kW or over.

Note: "H" and "L" mean high and low. "○" means abnormal condition has occurred.

18.6 Automatic and Remote Control of Auxiliary Machinery

18.6.1 Automatic Operation of Air Compressors

In cases where air compressors for starting and air compressors for controlling are automatically operated, alarm devices are to be provided to indicate any pressure drop in air reservoirs.

18.6.2 Automatic Starting and Stopping of Bilge Pumping Arrangements

In cases where bilge pumps are capable of being started and stopped automatically, alarm devices are to be provided to indicate any high level of bilge in relevant bilge wells and the running of pumps for a long time.

18.6.3 Thermal Oil Installations

Thermal oil installations arranged to be automatically controlled are to comply with the following:

(1) Control devices

Control devices are to comply with **18.4.2-1** and **-2**, also with **9.12.2-1** and **-2**.

(2) Safety devices

Safety devices are to comply with **9.12.1** and **9.12.2-5**.

(3) Alarm devices

Thermal oil installations are to be provided with alarm devices which operate in the following cases:

(a) When the safety devices required in **(2)** have operated.

(b) When the temperature of the fuel at the inlet of burner has fallen in cases where heated fuel is used.

18.6.4 High Temperature Alarm for Oil Heaters

In cases where the temperature for fuel oil and lubricating oil is automatically controlled, high temperature alarm devices are to be provided, except in cases where oils are not heated above their flashpoints.

18.6.5 Opening and Closing Devices for Sea Valves

In cases where sea valves to be fitted on the shell plating below the load water line are remotely or automatically controlled, other opening and closing devices which can be easily operated even in the event of failure of such automatic or remote control devices are to be provided.

18.6.6 Liquid Level Alarm Systems for Fuel Oil Tanks

In cases where fuel transfer to fuel oil tanks is automatically controlled, the receiving tanks are to be provided with high and low level alarms.

18.6.7 Mooring Arrangements

In cases where mooring arrangements are provided with remote control devices, these mooring arrangements are to be capable of being locally operated.

18.6.8 Fuel Filling Arrangements

In cases where arrangements for filling fuel into their respective fuel tanks from outside of the ships (hereinafter referred to as "fuel filling arrangements" in this Part) are provided with remote control devices, the fuel filling arrangements are to be such as not to interfere with the filling of fuel, even in the event of failure of any of the remote control devices.

18.6.9 Reciprocating Internal Combustion Engines

1 With respect to the safety measures for auxiliary machinery driven by reciprocating internal combustion engines, the requirements specified in **2.4.5-1** are to be applied.

2 The requirements in **18.5.2** apply correspondingly to the automatic or remote control devices for emergency reciprocating internal combustion engines other than those mentioned in **18.5.2**.

18.7 Tests

18.7.1 Shop Tests*

After being constructed, automatic or remote control systems of machinery and equipment, considered necessary by the Society, are to be subjected to the following tests:

(1) Environmental tests

Devices, units and sensors (hereinafter referred to as "automatic devices" in this Part) and automatic equipment composed of automatic devices are to be subject to the following tests at the manufacturing site. The procedures for these tests are to be deemed appropriate by the Society.

(a) External examination
(b) Operation test and performance test
(c) Electrical power supply failure test (to be applied to electrical/electronic devices, etc.)
(d) Electrical power supply fluctuation test (to be applied to electrical/electronic devices, etc.)
(e) Power supply fluctuation test (to be applied to hydraulic/pneumatic devices, etc.)
(f) Insulation resistance test (to be applied to electrical/electronic devices, etc.)
(g) High voltage test (to be applied to electrical/electronic devices, etc.)
(h) Pressure test (to be applied to hydraulic/pneumatic devices, etc.)
(i) Dry heat test
(j) Damp heat test
(k) Vibration test
(l) Inclination test (to be applied to equipment with moving parts)
(m) Cold test
(n) Salt mist test (to be applied to devices installed in unenclosed spaces such as open decks)
(o) Electrostatic discharge immunity test (to be applied to electronic devices)
(p) Radiated radio frequency immunity test (to be applied to electronic devices)
(q) Conducted low frequency immunity test (to be applied to electronic devices)
(r) Conducted high frequency immunity test (to be applied to electronic devices)
(s) Burst/Fast transient immunity test (to be applied to electronic devices)
(t) Surge immunity test (to be applied to electronic devices)
(u) Radiated emission test (to be applied to electronic devices that emit the electromagnetic wave)
(v) Conducted emission test (to be applied to electronic devices that emit the electromagnetic wave)
(w) Flame retardant test (to be applied to flammable enclosures of equipment)
(x) Other tests considered necessary by the Society

(2) Completion tests of automatic equipment

All automatic devices which have passed the environmental tests specified in **(1)** are to be subjected to the following tests after completion of their assembly as automatic equipment. The procedures of these tests are to comply with the requirements deemed appropriate by the Society.

(a) External examination

(b) Operation tests and performance tests

(c) Insulation resistance tests and high voltage tests (to be applied to electric/electronic devices etc.)

(d) Pressure tests (to be applied to hydraulic/pneumatic devices etc.)

(e) Other tests deemed necessary by the Society.

18.7.2 Approval of Use

1 In cases where automatic devices and automatic equipment have passed the environmental tests specified in **18.7.1**, they will receive approval of use from the Society; and, upon request from the manufacturer, the Society will make this information public.

2 With respect to all automatic devices and automatic equipment which have already received approval of use from the Society, a part or all of the environmental test specified in **18.7.1(1)** may be omitted.

18.7.3 Tests after Installation On Board

After being installed onboard, the automatic or remote control systems of machinery and equipment are to be confirmed that they operate effectively, under practical conditions as far as possible. However, parts of these tests may be carried out during sea trials.

Chapter 19 WATERJET PROPULSION SYSTEMS

19.1 General

19.1.1 Application

1 The requirements in this Chapter apply to waterjet propulsion systems intended for high speed engines used for main propulsion and steering (hereinafter referred to as "propulsion systems" in this Chapter).

2 The prime movers used for driving propulsion systems are to comply with the following requirements:

(1) Reciprocating internal combustion engines: **Chapter 2**

(2) Gas turbines: **Chapter 4**

3 The following requirements need not be applied to those propulsion systems without deflectors.

(1) **19.5.2-1**

(2) **19.5.3**

(3) **19.5.4-3**

(4) **19.6.2**

(5) **19.6.3(1)**, **(2)**, **(5)**, **(6)** and **(7)**

(6) **19.7.1-1**, **-5** through **-10**

4 Special consideration will be given to propulsion systems of unconventional designs to which the requirements in this Chapter are not applicable.

19.1.2 Terminology

The terms used in this Chapter are defined as follows:

(1) "Propulsion systems" are systems, including the following **(a)** through **(d)** components, which receive water through inlet ducts and discharge water through nozzles at an increased velocity to produce propulsive thrust and steering.

(a) Shafting (main shafts, bearings, shaft couplings, coupling bolts and sealing devices)

(b) Water intake ducts

(c) Waterjet pump units

(d) Steering systems

(2) "Pump units" are made up of impellers, impeller casings, stators, stator casings, nozzles, bearings, bearing housing and sealing devices.

(3) "Impellers" are rotating assemblies provided with blades to give energy to the water.

(4) "Main shafts" are shafts that impellers are connected to.

(5) "Water intake ducts" are portions that lead water drawn from water intakes to impeller inlets.

(6) "Nozzles" are portions that inject rectified water from impellers.

(7) "Deflectors" are devices serving as rudders by leading water injected from nozzles either to port or to starboard.

(8) "Reversers" are devices to thrust ships so as to go astern by reversing flow directions of water injected from nozzles.

(9) "Stators" are assemblies composed of rows of stationary vanes that reduce any swirl added to the water by impellers.

(10) "Steering system" is a ship's directional control system, including the steering gear, steering gear control system and rudder (including the rudder stock) if any, or any equivalent system (including deflectors, reversers and steering actuating systems driving defectors and/or reversers) for applying force on the ship hull to cause a change of heading or course. (*See* **Fig. D19.1** and **Fig. D19.2**)

(11) "Steering actuating system" consists of a steering gear power unit, a steering actuator and, for hydraulic or electrohydraulic steering gear, hydraulic piping.

(12) "Steering actuator" is a component which converts power into mechanical action to control the propulsion system as follows.

(a) In the case of electric steering: electric motor and driving pinion.

(b) In the case of electro hydraulic steering: hydraulic motor and driving pinion.

(13) "High speed engines" are diesel engines complying with the following condition or gas turbines:

$(S \cdot n^2)/(1.8 \times 10^6) \geq 90$

$(\pi \cdot d_j \cdot n)/(6 \times 10^4) \geq 6$

S : Length of stroke (mm)

n : Number of revolutions of an engine at maximum continuous output (rpm)

d_j : Diameter of journal (mm)

(14) "Declared steering angle limits" are the operational limits in terms of maximum steering angle or equivalent, according to manufacturer guidelines for safe operation, also taking into account ship speed, propeller torque/speed or other limitations; furthermore, "declared steering angle limits" are to be declared by the directional control system manufacturer for each ship specific non-traditional steering means, and ship manoeuvrability tests, such as those in the Standards for Ship Manoeuvrability (*IMO* resolution *MSC*.137(76)) are to be carried out with steering angles not exceeding the declared steering angle limits.

Fig. D19.1 Definition of steering system (in cases where two or more identical steering actuating systems are provided)

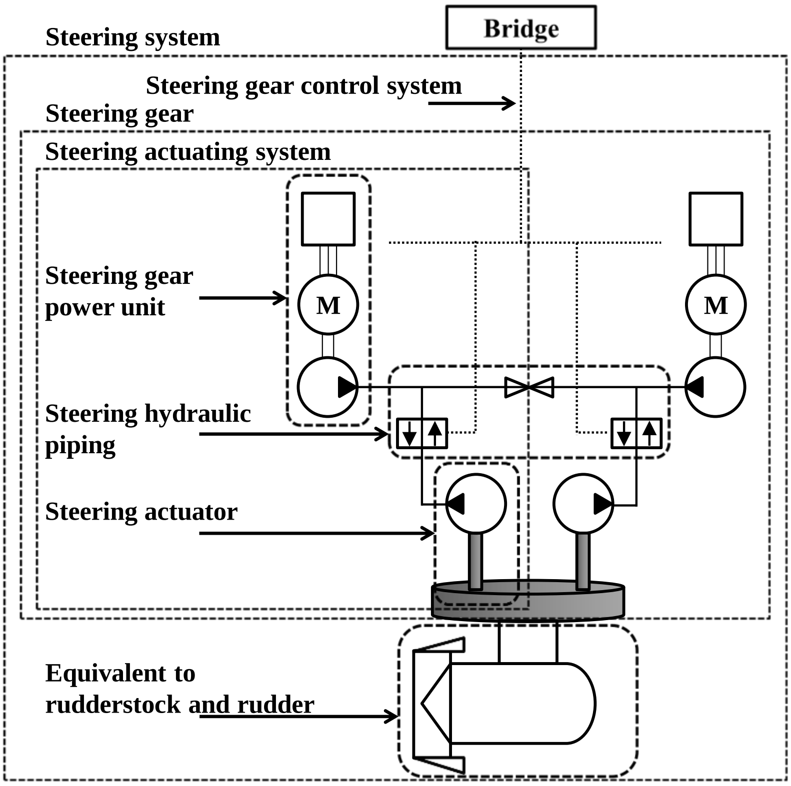
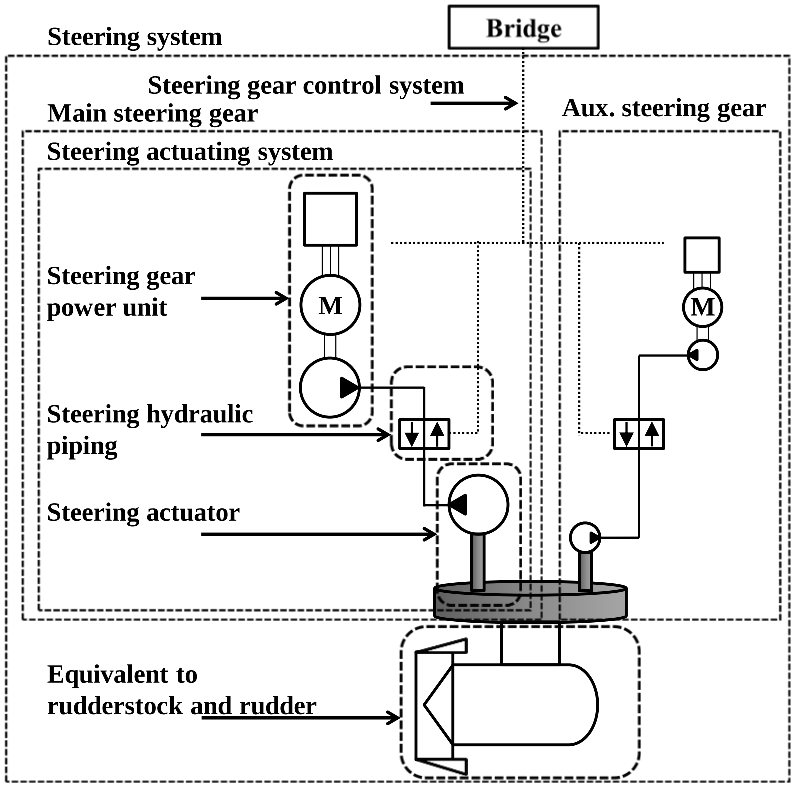


Fig. D19.2 Definition of steering system (in cases where a main steering gear and an auxiliary steering gear are provided)



19.1.3 Drawings and Data

Drawings and data to be submitted are generally as follows.

(1) Particulars

(2) Specifications

(3) Material specifications

(4) Details of welding procedures

(5) General arrangements and sectional assembly drawings (showing the materials and dimensions of various parts, including water intake ducts)

(6) Shafting arrangements (showing arrangements, shapes and construction of main engines, gears, clutches, couplings, main shafts, main shaft bearings and thrust bearings, sealing devices and impellers)

(7) Details of water intake ducts

(8) Construction of impellers (showing detailed blade profiles, the maximum diameter of blades from the centre of main shafts, number of blades, and material specifications)

(9) Details of bearings (including thrust bearings), in the case of roller bearings, together with specifications of such bearings and the calculation sheets for the life times of roller bearings

(10) Details of sealing devices (including waterjet pump unit sealing devices)

(11) Details of deflectors

(12) Details of reversers

(13) Details of steering actuators

(14) Piping diagrams (hydraulic systems, lubricating systems, cooling water systems, etc.)

(15) Arrangements of control systems and diagram of hydraulic and electrical systems (including safety devices, alarm devices and automatic steering)

(16) Arrangements and diagrams of alternative sources of power

(17) Diagram of indication devices for deflector positions

(18) Torsional vibration calculation sheets and calculation sheets for bending natural frequencies when bending vibrations due to self-weight are expected

(19) Strength calculation sheets for deflectors and reversers, etc.
(20) Others items considered to be necessary by the Society

19.1.4 Display of Operating Instructions

Simple operating instructions with block diagrams showing the change-over procedures for propulsion systems and control systems are to be permanently displayed on navigation bridges and at auxiliary steering stations.

19.1.5 Operating and Maintenance Instructions for Propulsion Systems, etc.

Operating and maintenance instructions and engineering drawings for propulsion systems are to be provided and written in a language understandable by officers and crew members who are required to understand such information in the performance of their duties.

19.2 Number of Propulsion Systems and Auxiliary Steering Station

19.2.1 Number of Propulsion Systems

1 In general, a minimum of two propulsion systems are to be provided for ships.

2 Each steering system for a propulsion system is to be provided with a main steering gear and an auxiliary steering gear. The main steering gear and the auxiliary steering gear are to be so arranged that the failure of one of them will not render the other one inoperative.

3 Notwithstanding the requirements of **-2** above, in cases where each main steering system comprises two or more identical steering actuating systems, an auxiliary steering gear need not be fitted provided that each steering gear:

(1) is capable of satisfying the requirements in **19.5.1-1(2)** while operating with all steering gear steering actuating systems;
(2) is arranged so that after a single failure in its piping or in one of the steering actuating systems, steering capability can be maintained or speedily regained;

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

4 In special cases, a single propulsion system installation may be considered, notwithstanding the requirements specified in **-1** to **-3**, provided that the ship in question is not engaged in international voyages. In such cases, the functions of propulsion and steering are to be designed with redundancy in the following arrangements:

(1) A minimum of two prime movers are to be provided.
(2) A minimum of two steering actuating systems are to be provided.
(3) Electric supply is to be maintained or restored immediately in cases where there is a loss of any one of the main generators in service so that the functioning of at least one of the prime mover and steering system is maintained by the arrangements specified in **19.6.2-1(1)** or **(2)**.

19.2.2 Auxiliary Steering Stations

1 Auxiliary steering stations, in cases where deflectors are to be operable other than from navigation bridges, are to be provided.

2 Auxiliary steering stations are to comply with the following requirements:

(1) Auxiliary steering stations are to be enclosed compartments that are readily accessible, and, as far as possible, separated from machinery spaces.
(2) Auxiliary steering stations are to be provided with adequate space so as to permit propulsion systems to be operated effectively.
(3) Auxiliary steering stations are to be provided with suitable arrangements to ensure working access to steering positions. These arrangements are to include handrails and gratings or other non-slip surfaces to ensure suitable working conditions in the event of any hydraulic fluid leakage.

19.3 Materials and Welding

19.3.1 Materials

1 Materials used for the following components are to be adequate for their service conditions and are to comply with the requirements in **Part K**.

(1) Shafting (excluding bearings and sealing devices)
(2) Impellers

(3) Impeller casings, stator casings and bearing housings
(4) Water intake ducts which are a part of the shell plating (including shaft covers)
(5) Mounting flanges and bolts of waterjet pump units
(6) Deflectors and reversers (including pins)
(7) Hydraulic piping

2 The materials used for the following components are also to comply with the requirements below:
(1) Shafting: **6.2.1**
(2) Impellers: **7.1.3**
(3) Hydraulic pumps, hydraulic piping and steering actuators: **10.2.1**, **12.1.4** and **15.4.1**

19.3.2 Welding

In cases where principal components of propulsion systems are of welded construction, they are to comply with the requirements specified in **Chapter 11**.

19.4 Construction and Strength

19.4.1 Main Shaft

The minimum diameter of main shafts is to be not less than the value determined by the following formula:

$$d_s = k \cdot \sqrt[3]{\frac{H}{N_o}}$$

where

d_s : Required diameter of main shaft (mm)
H : Maximum continuous output of main engine (kW)
N_o : Number of revolutions of main shaft at the maximum continuous output (rpm)
k : Values shown in **Table D19.1**

Table D19.1 Values of k according to Fitting Method

<table>
<tr><td colspan="2" rowspan="4">Shaft material</td><td colspan="5">Position</td></tr>
<tr><td colspan="4">Fitting parts of shafts with impellers and shaft couplings</td><td rowspan="3">Other parts</td></tr>
<tr><td colspan="4">Fitting method</td></tr>
<tr><td>Keyway</td><td>Spline</td><td>Flange Coupling</td><td>Force Fitting</td></tr>
<tr><td rowspan="2">Carbon steel or low alloy steel</td><td>Shafts of Kind 2</td><td>105</td><td>108</td><td colspan="2">102</td><td>105</td></tr>
<tr><td>Shafts of Kind 1</td><td rowspan="2">The value in the Note below where $a_1 = 100$, or $a_2 = 80$</td><td rowspan="2">The value in the Note below where $a_1 = 102$, or $a_2 = 82$</td><td colspan="2" rowspan="2">The value in the Note below Where $a_1 = 98$, or $a_2 = 78$</td><td rowspan="2">The value in the Note below where $a_1 = 100$, or $a_2 = 80$</td></tr>
<tr><td colspan="2">Austenitic stainless steel</td></tr>
<tr><td colspan="2">Martensite precipitation hardened type stainless steel</td><td>80</td><td>82</td><td colspan="2">78</td><td>80</td></tr>
</table>

Note:

$200 \leq \sigma_y \leq 400: \ k = a_1 - 0.1(\sigma_y - 200)$

$\sigma_y > 400: \ k = a_2$

where

σ_y : Yield point or 0.2 % of proof stress of main shaft material (N/mm^2)

19.4.2 Shaft Couplings and Coupling Bolts

1 The minimum diameter of shaft coupling bolts at joining faces of couplings is to be not less than the value determined by the following formula:

$$d_b = 15300\sqrt{\frac{H}{N_o}\left(\frac{1}{nDT_b}\right)}$$

where

d_b : Required diameter of shaft coupling bolt (*mm*)

n : Number of bolts

D : Pitch circle diameter (*mm*)

T_b : Specified tensile strength of bolt material (N/mm^2)

Other symbols used here are the same as those used in **19.4.1**

2 The thickness of shaft coupling flanges at pitch circles is not to be less than the required diameter of shaft coupling bolts determined by the formula in **-1** above. However, such a value is not to be less than 0.2 *times* the required diameter of the corresponding shaft.

3 Fillet radii at the base of flanges are not to be less than 0.08 *times* the diameter of their respective shafts in cases where fillets are not to be recessed in way of nuts and bolt heads.

19.4.3 Impeller Blades

The strength of impeller blades at their root is to be determined so that the following formula is satisfied. In such cases, the allowable stress value of the material is, in principle, to be 1/1.8 of the specified yield point (or 0.2 % of proof strength).

$$S \geq \frac{5.8\times10^7 H}{Lt^2ZN_o} + 2.2\times10^{-11}D^2N_o^2$$

where

S : Allowable stress of impeller material (N/mm^2)

Z : Number of impeller blades

L : Width of impeller blade at root (*mm*)

t : Maximum thickness of impeller blade at root (*mm*)

D : Diameter of impeller (*mm*)

Other symbols used here are the same as those used in **19.4.1**

19.4.4 Water Intake Ducts, etc.

Suction water intake ducts, impeller casings and nozzles are to have strength enough to handle their design pressure, and consideration is to be given to any corrosion.

19.4.5 Sealing Devices

The materials, constructions and arrangements of sealing devices (excluding gland packing type sea water sealing devices) for shafting and waterjet pump units are to be approved by the Society.

19.4.6 Torsional Vibration and Bending Vibration of Main Shaft

1 General

(1) Notwithstanding the requirements specified in **19.1.3** concerning the submission of torsional vibration calculation sheets for main shafting systems, the submission of such sheets may be omitted in cases where shafting systems are of the same type as one that has been previously approved or it can be readily assumed that such shafting systems will not cause any excessive vibration stress.

(2) Measurements of torsional vibrations to confirm the correctness of estimated values are to be carried out. However, in cases where the submission of torsional vibration calculation sheets is omitted according to the requirements in **(1)**, or the Society considers that there is no critical vibration within the service speed range, the measurement of torsional vibrations may be omitted.

2 Allowable Limits

Torsional vibration stresses of main shafting systems are to be in accordance with the following **(1)** and **(2)** requirements within the service speed range of such shafting systems:

(1) Torsional vibration stresses within the range from 80% up to and including 105% of the number of maximum continuous

revolutions are not to exceed τ_1 given in the following:

$\tau_1 = A - B\lambda^2 \quad (0.8 < \lambda \leq 0.9)$

or

$\tau_1 = C \quad (0.9 < \lambda \leq 1.05)$

where

τ_1 : Allowable limit of torsional vibration stresses for the range of $0.8 < \lambda \leq 1.05$ (N/mm^2)

λ : Ratio of the number of revolutions to the number of maximum continuous revolutions

A, B and C : Values shown in **Table D19.2**

In cases where the specified tensile strength of materials of carbon steel shafts or low alloy steel shafts of Kind 1 exceeds 400 N/mm^2, the value of τ_1 may be increased by multiplying the factor k_m given in the following formula:

$$k_m = \frac{T_s + 160}{560}$$

where

k_m : Correction factor

T_s : Specified tensile strength of main shaft material (N/mm^2)

(2) Torsional vibration stresses of within the range of 80 % and below of the number of maximum continuous revolutions of engines are not to exceed τ_2 given below. In cases where torsional vibration stresses exceed the value calculated by the formula of τ_1 shown in **(1)**, barred speed ranges are to be imposed. In this case, the formula for τ_1 is the one for the range of $\lambda \leq 0.9$.

$\tau_2 = 2.3\tau_1$

where

τ_2 : Allowable limit of torsional vibration stresses for the range of $\lambda \leq 0.8$ (N/mm^2)

3 Bending Vibrations

For main shafting systems of propulsion systems, consideration is to be given to natural vibrations due to the bending of shafting systems.

Table D19.2 Values of A, B and C

	Carbon steels or low alloy steels		Austenitic stainless steels	Martensite precipitation hardened type stainless steels
	Shafts of Kind 1	Shafts of Kind 2		
A	24.3	9.0	26.4	39.6
B	24.1	6.2	26.4	37.1
C	4.8	4.0	5.0	9.6

19.5 Steering Systems

19.5.1 Capability of Steering Gear

1 The main steering gear is to be:

(1) of adequate strength and capable of steering the ship at the maximum ahead service speed specified in **2.1.8, Part A,** which is to be demonstrated;

(2) capable of changing direction of the propulsion system from one side to the other at declared steering angle limits at an average turning speed of not less than 2.3 °/s with the ship running ahead at the speed specified in **2.1.8, Part A**;

(3) for all ships, operated by power; and

(4) so designed that they will not be damaged at maximum astern speed; this design requirement need not be proved by trials at maximum astern speed and declared steering angle limits.

2 The auxiliary steering gear is to be:

(1) of adequate strength and capable of steering the ship at navigable speed and of being brought speedily into action in an emergency;

(2) capable of changing direction of the propulsion system from one side to the other at declared steering angle limits at an average

turning speed of not less than 0.5 °/s with the ship running ahead at one half of the maximum ahead service speed specified in **2.1.8, Part A** or 7 *knots*, whichever is the greater; and

(3) for all ships, operated by power where necessary to meet the requirements of Regulation 29.4.2, Chapter II-1, *SOLAS* and in any ship having power of more than 2,500 *kW* propulsion power per propulsion system.

3 Reversers are to be such that they enable the ship to go astern with sufficient steering under normal circumstances, and they are to have astern power to provide effective braking for ships when changing from ahead to astern runs.

19.5.2 General Construction

1 Design pressures of the scantlings of piping and other components of steering actuating systems subject to internal hydraulic pressure are to be at least 125 % of the maximum working pressure expected under the worst permissible operating condition, taking into account any pressure which may exist in the low pressure side of the system. Design pressures are not to be less than relief valve setting pressures.

2 Reversers are to have sufficient strength against any thrusts at maximum astern power output.

3 The construction and strength of hydraulic pumps and hydraulic systems are to comply with the requirements in **10.5**, **12.2.1**, **12.3**, **12.4.2** through **12.4.4** and **12.5.1**.

4 The arrangements of piping, relief valves and measuring devices for hydraulic systems and the construction of liquid level indicators are to comply with the requirements in **13.2.1** and **13.8.4**.

19.5.3 Steering Actuators

1 The strength of steering actuators is to comply with the requirements specified in **15.4.4**.

2 The construction of oil seals in steering actuators is to comply with the requirements specified in **15.4.5**.

19.5.4 Steering Actuating Systems

1 Suitable arrangements to maintain the cleanliness of hydraulic fluid are to be provided after taking into consideration the type and design of the steering actuating system.

2 Arrangements for bleeding air from steering actuating systems are to be provided where necessary.

3 Relief valves are to be fitted to any parts of steering actuating systems which can be isolated and in which pressure can be generated from power sources or from external forces. Setting pressures of relief valves are not to be less than 125 % of the maximum working pressure expected in the protected part. Minimum discharge capacities of relief valves are not to be less than 110 % of the total capacity of pumps which provide power for steering actuators; under such conditions, however, no rise in pressure is to exceed 10 % of the setting pressure. In this regard, due consideration is to be given to any anticipated extreme ambient conditions in respect of oil viscosity.

4 Low level alarms are to be provided for hydraulic fluid tanks to give the earliest practicable indication of fluid leakage from steering actuating systems. These alarms are to be audible and visual, and are to be given on navigation bridges and at other positions from which main engines are normally controlled.

5 In cases where flexible hoses are used for steering actuating systems, the construction and strength of such flexible hoses are to comply with the requirements specified in **15.4.6**.

19.5.5 Stoppers

1 Propulsion systems are to be provided with stoppers for deflectors in order to limit steering angles.

2 Propulsion systems are to be provided with positive arrangements, such as limit switches, for stopping deflectors before coming into contact with any stoppers. These arrangements are to be activated by the actual movements of deflectors and not through control systems for steering. Mechanical links may be used for this purpose.

19.6 Electric Installations

19.6.1 General

For items not specified in this section **19.6**, the requirements specified in **Part H** are to apply.

19.6.2 Maintenance of Electric Supply

1 Main sources of electric power are to be so arranged that electric supplies to relevant equipment are maintained or restored immediately in the case of a loss of any one of the generators in service so as to ensure the functions of propulsion and steering of at least one of the propulsion systems, its associated control systems and its indication devices of steering gear by the following arrangements:

(1) In cases where electrical power is normally supplied by one generator, adequate provisions are to be made for automatic starting and connecting to main switchboards of standby generators of sufficient capacity to maintain the functions of the above with automatic restarting of important auxiliaries including sequential operations in cases where there is a loss of electrical power of the generator in operation.

(2) If electrical power is normally supplied by more than one generator simultaneously in parallel operations, provisions are to be made to ensure that, in cases where there is a loss of electrical power of one of generating sets, the remaining ones are kept operational so as to maintain the functions required by **1** above. (See **2.3.6, Part H**)

2 In cases where the propulsion power exceeds 2,500 *kW* per thruster unit, an alternative source of power is to be provided in accordance with the following:

(1) The alternative source of power is to be either:

(a) An emergency source of electric power; or

(b) An independent source of power located in the steering gear compartment and used only for this purpose.

(2) Any alternative source of power is to be capable of automatically supplying alternative power within 45 *seconds* to the steering gear (including its associated control system) and the indication devices for the steering gear. In such cases, the alternative source of power is to be capable of changing direction of the ship's directional control system from one side to the other at declared steering angle limits at an average turning speed of not less than 0.5 °/*s* with the ship running ahead at one half of the speeds specified in **2.1.8, Part A** or 7 *knots*, whichever is greater. Alternative sources of power are to have enough capacity for the continuous operation of such systems for at least 30 *minutes* in every ship of 10,000 *gross tonnage* or more, and for at least 10 *minutes* in every other ship.

(3) Automatic starting arrangements for generators or prime movers of pumps used as the independent source of power specified in **(1)(b)** are to comply with the requirements for starting devices and performance in **3.4.1, Part H**.

19.6.3 Electrical Installations for Steering Systems

In cases where hydraulic pumps for steering actuating systems are driven by electric motors, electrical installations for steering and reversing systems are to comply with the following requirements:

(1) Each steering system is to be served by at least two exclusive circuits fed directly from main switchboards. One of these circuits, however, may be supplied through the emergency switchboard.

(2) Cables used in those exclusive circuits required in **(1)** are to be separated, as far as practicable, throughout their length.

(3) Audible and visual alarms are to be given on navigation bridges in the event of any power failure to electric motors for hydraulic pumps.

(4) Means for indicating that electric motors for hydraulic pumps are running are to be installed on navigation bridges and positions from which main engines are normally controlled.

(5) Short circuit protection and overload alarms are to be provided for such circuits and motors respectively. Overload alarms are to be both audible and visible and are to be situated in conspicuous positions in places from which main engines are normally controlled.

(6) Protection against excess current, including starting currents, if provided, is to be for not less than twice the full load current of those motors or circuits so protected, and is to be arranged to permit the passage of the appropriate starting currents.

(7) In cases where a three-phase supply is used, alarms are to be provided that will indicate failure of any one of the supply phases. Such alarms are to be both audible and visible and are to be situated in conspicuous positions in places from which main engines are normally controlled.

19.7 Controls

19.7.1 General

1 Propulsion systems are to be capable of being brought into operation and being controlled on navigation bridges.

2 Steering systems are to be capable of being controlled from the auxiliary steering stations specified in **19.2.2**. Means are to be provided in such auxiliary steering stations for disconnecting any control systems, operable from navigation bridges, from the steering systems they serve.

3 Reversing systems are to be controlled in local control stations for main propulsion or in auxiliary steering stations. Means are to be provided in local control stations for main propulsion or in auxiliary steering stations for disconnecting any control systems, operable from navigation bridges, from reversing systems they serve.

4 In the event of any failure of remote control devices for reversing systems, preset positions of reversers are to be maintained until control over of such systems can be established at local control stations for main propulsion or at auxiliary steering stations.

5 Independent control devices are to be provided for propulsion systems. In cases where multiple propulsion systems are designed to operate simultaneously, they may be controlled by a single device such as a joystick.

6 Those control devices specified in **-5** are to be so designed that the failure of one such control device does not result in the failure of another control device.

7 Cables and pipes of control systems are to be separated, as far as practicable, throughout their length.

8 In cases where control systems are electric, they are to be served by its own separate circuit supplied directly from switchboard busbars supplying that power circuits for the propulsion systems at a point on the switchboard adjacent to the supply to the power circuits for the propulsion systems.

9 Short circuit protection only is to be provided for control supply circuits.

10 Audible and visual alarms are to be given on navigation bridges and at positions from which main engines are normally controlled in the event of any failure of control systems or electrical power supplies to such control systems.

11 Means of communication are to be provided between navigation bridges and all control stations, including auxiliary steering stations.

12 Propulsion systems for ships provided with automatic steering are to be capable of immediate change-overs from automatic to manual steering.

13 For those items concerned with safety, alarms and control devices for propulsion systems not specified in **19.7.1**, the requirements specified in **18.1** through **18.3** and **18.7** are to apply.

19.7.2 Indication Devices

1 Indication devices for deflector positions

(1) Deflector positions are to be indicated on navigation bridges and in auxiliary steering stations.

(2) Indication devices for deflector positions are to be independent of control systems.

2 Indication devices for reverser positions

Reverser positions are to be indicated on navigation bridges, at control stations (including auxiliary steering stations) and at monitoring stations for propulsion systems.

3 Indication devices for impeller speed

Impeller speeds are to be indicated on navigation bridges, at control stations (including auxiliary steering stations) and at monitoring stations for propulsion systems.

19.8 Piping

19.8.1 Lubricating Oil Systems

1 Lubricating oil systems for propulsion systems are to comply with relevant requirements specified in **13.10**.

2 Lubricating oil arrangements for propulsion systems are to be provided with alarm devices which give visible and audible alarms on navigation bridges and at positions from which main engines are normally controlled in the event of any failure of the supply of lubricating oil or an appreciable reduction of lubricating oil pressure.

19.9 Tests

19.9.1 Shop Tests

1 For impeller casings, stator casings and bearing housings, hydrostatic tests at pressures 1.5 *times* design pressure are to be carried out.

2 For impellers, dynamic balancing tests are to be carried out.

3 For forward bearing tubes of main shafts and sealing device tubes, hydrostatic tests of at pressures of at least 0.2 *MPa* or 1.5

times design pressure, whichever is higher, are to be carried out.

4 For steering actuating systems, the tests specified in **15.5.1** are to be carried out.

5 For control, safety and alarm devices, performance tests are to be carried out.

19.9.2 Tests after Installation On Board

1 For hydraulic piping systems, leak tests at pressures at least equal to the maximum working pressure are to be carried out after installation on board.

2 For sealing devices for waterjet pump units, leak tests at working oil pressure are to be carried out.

3 Operation tests of propulsion systems are to be carried out.

Chapter 20 AZIMUTH THRUSTERS

20.1 General

20.1.1 Application

1 The requirements in this Chapter apply to azimuth thrusters intended for main propulsion (hereinafter referred to as “thrusters” in this Chapter).

2 The prime movers for driving thrusters are to comply with the following requirements:

(1) Reciprocating internal combustion engines: **Chapter 2**

(2) Gas turbines: **Chapter 4**

(3) Electric motors: **Chapter 2** and **5, Part H**

3 Special consideration will be given to those thrusters of unconventional design to which the requirements in this Chapter are not applicable.

20.1.2 Terminology

The terms used in this Chapter are defined as follows:

(1) “Thrusters” are propulsion units which control ship direction through steering functions enabled by their own capability of azimuthing. Thrusters include the following components:

(a) Propellers;

(b) Propeller shafts;

(c) Gears, clutches and gear shafts for transmission of propulsion torque (when integrated in thrusters);

(d) Azimuth thruster casings; and

(e) Steering system.

(2) “Azimuth thruster casings” are watertight structures that include steering columns (or struts), propeller pods, propeller nozzles and nozzle supports.

(3) “Azimuth steering gear” is a device for applying steering torque to thrusters, and include electric motors, hydraulic pumps, hydraulic systems, hydraulic motors and gear assemblies for azimuth steering gear.

(4) “Steering system” is a ship’s directional control system, including the steering gear, steering gear control system and rudder (including rudder stocks) if any, or any equivalent system (including azimuth steering gear) for applying force on the ship hull to cause a change of heading or course.(*See* **Fig. D20.1** and **Fig. D20.2**)

(5) “Steering actuating system” consists of a steering gear power unit, a steering actuator and, for hydraulic or electrohydraulic steering gear, hydraulic piping.

(6) “Steering actuator” is a component which converts power into mechanical action to control the propulsion system as follows.

(a) In the case of electric steering: electric motor and driving pinion.

(b) In the case of electro hydraulic steering: hydraulic motor and driving pinion.

(7) “Declared steering angle limits” are the operational limits in terms of maximum steering angle or equivalent, according to manufacturer guidelines for safe operation, also taking into account ship speed, propeller torque/speed or other limitations; furthermore, “declared steering angle limits” are to be declared by the directional control system manufacturer for each ship specific non-traditional steering means, and ship manoeuvrability tests, such as those in the Standards for Ship Manoeuvrability (*IMO* resolution *MSC*.137(76)) are to be carried out with steering angles not exceeding the declared steering angle limits.

Fig. D20.1 Definition of steering system (in cases where two or more identical steering actuating systems are provided)

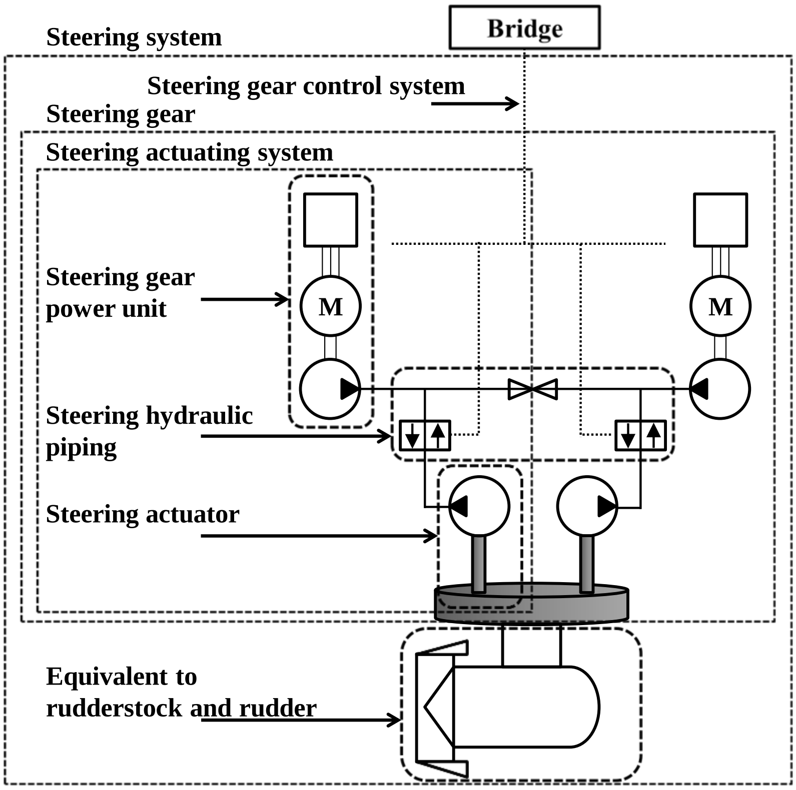
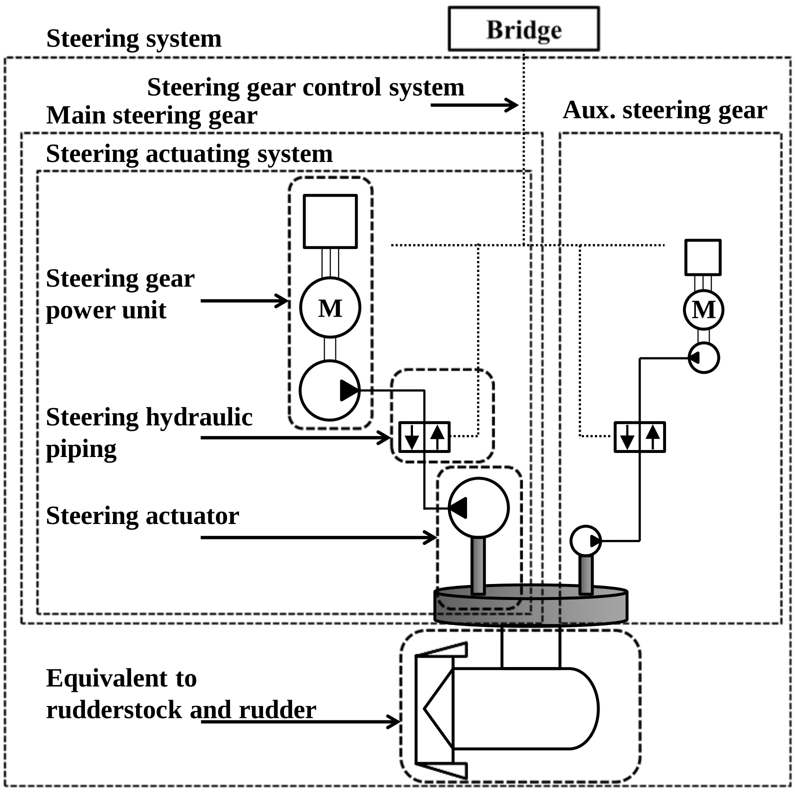


Fig. D20.2 Definition of steering system (in cases where a main steering gear and an auxiliary steering gear are provided)



20.1.3 Drawings and Data

Drawings and data to be submitted are generally as follows.

(1) Particulars
(2) Specifications
(3) Material specifications
(4) Details of welding procedures for principal components
(5) General arrangements and sectional assembly drawings
(6) Shafting arrangements (details of propeller shafts, gears, clutches, gear shafts, shaft couplings, bearings and sealing devices and propellers, together with specifications and service life calculations of roller bearings, torsional vibration calculations and propeller pull-up length calculation sheets)
(7) Details of azimuth thruster casings
(8) Drawings of azimuth steering gear (details of actuating systems, gear assemblies, bearings and sealing devices for azimuth steering gear)
(9) Piping diagrams (hydraulic systems, lubricating systems, cooling water systems, etc.)
(10) Arrangements of control systems and diagram of hydraulic and electrical systems (including safety devices, alarm devices and automatic steering)
(11) Arrangements and diagrams of alternative sources of power
(12) Diagrams of indication devices for azimuth angles
(13) Strength calculations
(14) When a vibration measurement system specified in **20.4.3(1)** is being used, the following **(a)** and **(b)** documents:
  (a) Function descriptions for vibration measurement systems
  (b) Management manuals including the following **i)** through **iii)**
    i) List of the bearings for vibration measurements and measurement points
    ii) Guidance for measurements (including the way for taking signals from casings)
    iii) Guidance for the analysis and the evaluation of measurement results
(15) When a Fe-density measurement system specified in **20.4.3(2)** is being used, the following **(a)** and **(b)** documents:
  (a) Function descriptions for Fe-density measurement systems
  (b) Management manuals including the following **i)** through **iii)**
    i) Guidance for lubricating oil sampling
    ii) Guidance for Fe-density measurements
    iii) Guidance for the analysis and the evaluation of measurement results
(16) Other plans and documents considered necessary by the Society

20.1.4 Display of Operating Instructions

Simple operating instructions with block diagrams showing change-over procedures for thrusters and control systems are to be permanently displayed on navigation bridges and in azimuth thruster compartments.

20.1.5 Operating and Maintenance Instructions for Thrusters, etc.

Operating and maintenance instructions and engineering drawings for thrusters are to be provided and written in a language understandable by officers and crew members who are required to understand such information in the performance of their duties.

20.2 Number and Position of Thrusters

20.2.1 Number of Thrusters

1 In general, a minimum of two thrusters is to be provided for ships.

2 Each of the steering systems of thrusters is to be provided with a main steering gear and an auxiliary steering gear. The main steering gear and the auxiliary steering gear are to be so arranged that the failure of one of them will not render the other one inoperative.

3 Notwithstanding the requirements of **-2** above, in cases where each main steering system comprises two or more identical steering actuating systems, an auxiliary steering gear need not be fitted provided that each steering gear:

(1) is capable of satisfying the requirements in **20.5.1-1(2)** while operating with all steering gear steering actuating systems;

(2) is arranged so that after a single failure in its piping or in one of the steering actuating systems, steering capability can be maintained or speedily regained;

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

4 In special cases, a single thruster installation may be subject to consideration and deemed acceptable, notwithstanding the requirements specified in **-1** to **-3**. In such cases, the functions of propulsion and steering are to be designed with redundancy as in the following arrangements:

(1) A minimum of two prime movers are to be provided.

(2) A minimum of two independent azimuth steering gear are to be provided. However, such azimuth steering gear may have only one gear wheel.

(3) Electric supply is to be maintained or restored immediately in the cases where there is a loss of any one of the main generators in service so that the functioning of at least one of the prime mover and steering system, is maintained by the arrangements specified in **20.6.2-1(1)** and **(2)**.

20.2.2 Position of Thrusters

1 Thrusters are to be installed in readily accessible enclosed compartments and be separated, as far as possible, from any machinery spaces.

2 Azimuth thruster compartments are to be of sufficient space as to permit thrusters to be operated effectively.

3 Azimuth thruster compartments are to be provided with suitable arrangements to ensure working access to azimuth steering gear machinery and controls. These arrangements are to include handrails and gratings or other non-slip surfaces to ensure suitable working conditions in the event of any hydraulic fluid leakage.

4 The locations where the following equipment are provided and which comply with the requirements specified in **20.7.1-2** may be deemed as azimuth thruster compartments.

(1) Instruments specified in **20.7.1-9**

(2) Communication means specified in **20.7.1-11**

(3) Gyro repeaters required in Regulation 19.2.5.2, Chapter V, *SOLAS*

20.3 Materials and Welding

20.3.1 Materials

1 The materials used for the following components are to be adequate for their service conditions and are to comply with the requirements in **Part K**.

(1) Gears, clutches, gear shafts and all principal components of shafting

(2) Propellers and blade fixing bolts of controllable pitch propellers

(3) Hydraulic piping of controllable pitch propellers and azimuth steering gear

(4) Azimuth thruster casings

(5) Gears and gear shafts for azimuth steering gear

(6) Bedplates for supporting thrusters

2 The materials used for the following components are also to comply with the requirements below:

(1) Gears, clutches, gear shafts and all principal components of shafting: **5.2.1** and **6.2.1**

(2) Propellers and blade fixing bolts of controllable pitch propellers: **7.1.3**

(3) Hydraulic piping of controllable pitch propellers and azimuth steering gear: **12.1.4**

20.3.2 Welding

In cases where the principal components of thrusters are of welded construction, they are to comply with the requirements specified in **Chapter 11**.

20.4 Construction and Strength

20.4.1 General

1 The installation and construction of thrusters are to be such that ship stability is not adversely affected even when sea water enters azimuth thruster casings and floods compartments where they are installed.

2 Sealing devices are to be provided in cases where thrusters penetrate hull structures to prevent any sea water from entering ships.

20.4.2 Gears, Clutches, Gear Shafts, etc.

The construction and strength of gears, clutches, gear shafts and etc. for propulsion are to comply with the requirements specified in **Chapter 5**. The construction and strength of bevel gears and gears for azimuth steering gear are to comply with recognised standards.

20.4.3 Propeller Shafts, Bearings and Sealing Devices of Propeller Shafts

The construction and strength of propeller shafts, bearings and sealing devices of propeller shafts are to comply with the requirements specified in **Chapter 6**. When roller bearings are used for the propeller shaft bearings and where a propeller shaft Kind 1*C* is being used, the system specified in the following **(1)** or **(2)** may be used instead of the temperature sensors and the temperature recorder specified in **6.2.11**. In this case, the executive management is to use their experience and knowledge to determine the criteria for each parameter for the ship (including the criteria for alarm and abnormal conditions).

(1) Vibration measurement system to measure vibration of power transmission system in the azimuth thrusters specified in the following **(a)** through **(c)**. Where the system is fixed type, the environmental tests specified in **18.7.1(1)** are to be carried out.
  (a) The measurement is to be carried out regularly at intervals not exceeding 3 *months*.
  (b) Measurement points and the relevant data described in the guidance for measurement in the management manual specified in **20.1.3(14)(b)** are to be recorded appropriately.
  (c) A trend display and frequency analysis of the measurement data are to be provided.

(2) Fe-density measurement system of lubricating oil in the azimuth thruster casings specified in the following **(a)** and **(b)**. Where the system is fixed type, the environmental tests specified in **18.7.1(1)** are to be carried out. In principle, sampling is to be carried out during navigation. Where the sampling can only be conducted at anchor, the sampling is to be carried out in 30 *minutes*, in principle, after the thrusters stop.
  (a) Sampling is to be carried out regularly at intervals not exceeding 3 *months*.
  (b) The measurement data is to be the amount of Fe per hour, considering the change of new lubricating oil. A trend display of the data is to be provided.

20.4.4 Propellers

The construction and strength of propellers are to comply with the requirements specified in **Chapter 7**.

20.4.5 Torsional Vibration of Shafting

Calculations for torsional vibration of shafting are to comply with the requirements specified in **Chapter 8**.

20.4.6 Strengthening for Navigation in Ice

Thrusters in ships intended to be registered with the ice-strengthened class notation are to comply with the requirements specified in **Chapter 8, Part I**.

20.5 Steering Systems

20.5.1 Capability of Steering Gears

1 The main steering gear is to be:

(1) of adequate strength and capable of steering the ship at the maximum ahead service speed specified in **2.1.8, Part A** which is to be demonstrated;

(2) capable of changing direction of the thruster from one side to the other at declared steering angle limits at an average turning speed of not less than 2.3 °/*s* with the ship running ahead at the speed specified in **2.1.8, Part A**;

(3) In addition to the requirements specified in **(2)** above, the rate of turning for the azimuth steering gear is to be not less than 1.0 *rpm* in static conditions of ships if astern power is obtained by turning thrusters.

(4) for all ships, operated by power; and

(5) so designed that they will not be damaged at maximum astern speed; this design requirement need not be proved by trials at maximum astern speed and declared steering angle limits.

2 The auxiliary steering gear is to be:

(1) of adequate strength and capable of steering the ship at navigable speed and of being brought speedily into action in an emergency;

(2) capable of changing direction of the thruster from one side to the other at declared steering angle limits at an average turning speed of not less than 0.5 °/*s* with the ship running ahead at one half of the maximum ahead service speed specified in **2.1.8, Part A** or 7 *knots*, whichever is the greater; and

(3) for all ships, operated by power where necessary to meet the requirements of Regulation 29.4.2, Chapter II-1, *SOLAS* and in any ship having power of more than 2,500 *kW* propulsion power per thruster.

20.5.2 Construction

1 Design pressures for calculations to determine the scantlings of piping and other components of steering actuating systems of azimuth steering gear subject to internal hydraulic pressure are to be at least 125 % of the maximum working pressure expected under the worst permissible operation conditions after taking into account any pressure which may exist in low pressure sides of such systems. Design pressures are not to be less than relief valve setting pressures.

2 The construction and strength of the hydraulic pumps and hydraulic systems are to comply with the requirements in **10.5**, **12.2.1**, **12.3**, **12.4.2** through **12.4.4** and **12.5.1**.

3 The installation of piping and arrangements of relief valves as well as measuring devices for hydraulic systems and the construction of liquid level indicators are to comply with the requirements in **13.2.1** and **13.8.4**.

20.5.3 Hydraulic Systems

Hydraulic power-actuated azimuth steering gear is to be provided with the following arrangements:

(1) Suitable arrangements to maintain the cleanliness of hydraulic fluids are to be provided after taking into consideration the types and designs of such hydraulic systems.

(2) Arrangements for bleeding air from hydraulic systems are to be provided where necessary.

(3) Relief valves are to be fitted to any part of hydraulic systems which can be isolated and in which pressure can be generated from power sources or from external forces. Setting pressures of such relief valves are not to be less than 125 % of the maximum working pressure expected in such protected parts. Minimum discharge capacities of relief valves are not to be less than 110 % of the total capacity of pumps which provide power for hydraulic motors; under such conditions, however, no rise in pressure is to exceed 10 % of the setting pressure. In this regard, due consideration is to be given to any anticipated extreme ambient conditions in respect of oil viscosity.

(4) Low level alarms are to be provided for hydraulic fluid tanks in order to give the earliest practicable indication of any hydraulic fluid leakage. These alarms are to be audible and visual, and are to be given on navigation bridges and at other positions from which main engines are normally controlled.

(5) In cases where flexible hoses are used for hydraulic systems, the construction and strength of such flexible hoses are to comply with the requirements specified in **15.4.6**.

20.5.4 Sealing Devices

Sealing devices for steering parts of azimuth steering gear are to be approved by the Society in their materials, construction and arrangement.

20.6 Electric Installations

20.6.1 General

1 Each steering system is to be served by at least two exclusive circuits fed directly from main switchboards. One of these circuits, however, may be supplied through the emergency switchboard.

2 Cables used in those exclusive circuits required in **-1** are to be separated, as far as practicable, throughout their length.

3 Audible and visual alarms are to be given on navigation bridges and at positions from which main engines are normally controlled in the event of any power failure to electric motors for propulsion and steering.

4 For items not specified in this section **20.6**, those requirements specified in **Part H** are to apply.

20.6.2 Maintenance of Electric Supply

1 In cases where any generators in service are lost, main sources of electric power are to be so arranged that electric supplies to any relevant equipment are maintained or restored immediately so as to ensure the functions of propulsion and steering of at least one thruster, its associated control systems and indication devices for azimuth angles by the following arrangements:

(1) In cases where electrical power is normally supplied by one generator, adequate provisions are to be made for the automatic starting and the connecting to main switchboards of standby generators of sufficient capacities to maintain the functions of the above with automatic restarting of important auxiliaries, including sequential operations, in cases of loss of electrical power to generators in operation.

(2) If electrical power is normally supplied by more than one generator simultaneously in parallel operations, provisions are to be made to ensure that, in cases of loss of electrical power to one of such generating sets, the remaining ones are kept operational so as to maintain the functions required by **-1** above. (See **2.3.6, Part H**)

2 In cases where propulsion power exceeds 2,500 *kW* per thruster, an alternative source of power is to be provided in accordance with the following:

(1) The alternative source of power is to be either:

(a) An emergency source of electric power; or

(b) An independent source of power located in the steering gear compartment and used only for this purpose.

(2) Any alternative source of power is to be capable of automatically supplying alternative power within 45 *seconds* to the steering gear (including its associated control system) and the indication devices for the steering gear. In such cases, the alternative source of power is to be capable of changing direction of the ship's directional control system from one side to the other at declared steering angle limits at an average turning speed of not less than 0.5 °/*s* with the ship running ahead at one half of the speeds specified in **2.1.8, Part A** or 7 *knots*, whichever is greater. Alternative sources of power are to have enough capacity for the continuous operation of such systems for at least 30 *minutes* in every ship of 10,000 *gross tonnage* or more, and for at least 10 *minutes* in every other ship.

(3) Automatic starting arrangements for generators or prime movers of pumps used as the independent source of power specified in **(1)(b)** are to comply with the requirements for starting devices and performance in **3.4.1, Part H**.

20.6.3 Electrical Installations for Azimuth Steering Gear

Electrical installations for azimuth steering gear are to comply with the following requirements:

(1) Means for indicating that electric motors for steering are running are to be installed on navigation bridges and positions from which main engines are normally controlled.

(2) Short circuit protection and overload alarms are to be provided for such circuits and motors respectively. Overload alarms are to be both audible and visible and are to be situated in conspicuous positions in places from which main engines are normally controlled.

(3) Protection against excess current, including starting currents, if provided, is to be for not less than twice the full load current of those motors or circuits so protected, and is to be arranged to permit the passage of the appropriate starting currents.

(4) In cases where a three-phase supply is used, alarms are to be provided that will indicate failure of any one of the supply phases. Such alarms are to be both audible and visible and are to be situated in conspicuous positions in places from which main engines are normally controlled.

20.7 Controls

20.7.1 General

1 Thrusters are to be capable of being brought into operation and being controlled from navigation bridges.

2 Azimuth steering gear is to be capable of being controlled from azimuth thruster compartments. Means are to be provided in azimuth thruster compartments for disconnecting any control system operable from navigation bridges from the steering system it serves.

3 Independent control devices are to be provided for thrusters. In cases where multiple thrusters are designed to operate simultaneously, they may be controlled by a single device such as a joystick.

4 Those control devices specified in **-3** are to be so designed that the failure of one such control device does not result in the failure of another control device.

5 Cables and pipes of control systems are to be separated, as far as practicable, throughout their length.

6 In cases where control systems are electric, they are to be served by its own separate circuit supplied from a power circuit for the thrusters from a point within the azimuth thruster compartment, or directly from switchboard busbars supplying that power circuits for thrusters at a point on the switchboard adjacent to the supply to the power circuits for the thrusters.

7 Short circuit protections are only to be provided for control supply circuits.

8 Audible and visual alarms are to be given on navigation bridges and at positions from which main engines are normally controlled, in the event of any failure of control systems or of electrical power supplies to such control systems.

9 The following instruments are to be provided on navigation bridges and at all control stations of thrusters

(1) Indication devices for propeller speed and direction of rotation in the cases of solid propellers

(2) Indication devices for propeller speed and pitch position in the case of controllable pitch propellers

(3) Indication devices for azimuth angle

10 Indication devices for azimuth angle specified in **-9(3)** are to be independent of control systems.

11 Means of communication are to be provided between navigation bridges and all control stations for thrusters.

12 Thrusters for ships provided with automatic steering are to be capable of immediate change-overs from automatic to manual steering.

13 For those items concerned with safety, alarms and control devices for thrusters not specified in **20.7.1**, the requirements specified in **18.1** through **18.3** and **18.7** are to apply.

20.8 Piping

20.8.1 Lubricating Oil Systems

1 Lubricating oil systems for thrusters are to comply with relevant requirements specified in **13.10**.

2 Lubricating oil arrangements of thrusters are to be provided with alarm devices which give visible and audible alarms on navigation bridges and at positions from which main engines are normally controlled in the event of any failure of the supply of lubricating oil or any appreciable reduction of lubricating oil pressure.

20.8.2 Cooling Systems

Cooling systems for thrusters are to comply with the requirements specified in **13.12** (in this case the term “main propulsion machinery” is to be read as “thrusters”).

20.9 Additional Requirements for Thrusters which Incorporate Electric Motors in Propeller Pods

20.9.1 General

1 Means to detect the ingress of sea water into propeller pods is to be provided, and audible and visual alarms are to be given on navigation bridges and at positions from which main engines are normally controlled. Means for discharging sea water from propeller pods is to be provided.

2 Fire detection and alarm systems are to be provided in propeller pods in cases where the pods can be accessed.

3 In cases where cooling fans are provided for propulsion motors, main cooling fans with sufficient capacities at the maximum output of propulsion motors as well as auxiliary cooling fans with sufficient capacities at the normal output of propulsion motors are to be provided. These cooling fans are to be arranged so that they can be easily changed over. However, such auxiliary fans may be omitted provided that exclusive cooling fans are provided for thrusters.

4 In cases where cooling fans are provided for propulsion motors, a means of control is to be provided for stopping such fans and closing any inlets and outlets of air for such fans from safe positions in the case of fire.

20.10 Tests

20.10.1 Shop Tests

1 For gears, the tests specified in **5.5.1** are to be carried out.

2 For propeller shaft sleeves, the tests specified in **6.3.1(2)** are to be carried out.

3 For propellers, the tests specified in **7.4.1** are to be carried out.

4 For azimuth steering gear, the following tests are to be carried out.

(1) The tests specified in **15.5.1**.

(2) The tests specified in **5.5.1** for gears.

5 For azimuth thruster casings, after assembly, pressure tests at the larger of 0.2 *MPa* and the following pressure of a water head equivalent to $1.5D$ or $2d$, whichever is smaller, are to be carried out. However, airtight tests at pressures of 0.05 *MPa* for propeller nozzles may be acceptable.

where

D : The depth of ship (m)

d : The design maximum load draught (m)

6 For safety and alarm devices, performance tests are to be carried out.

20.10.2 Tests after Installation On Board

1 For sealing devices for propeller shafts and azimuth steering gear, leak tests for the sealing devices are to be carried out at working oil pressure after installation on board.

2 For azimuth steering gear, leak tests for hydraulic systems are to be carried out at pressures at least equal to maximum working pressure after installation on board. However, when it is difficult to carry out such tests after installation on board, such tests may be carried out as shop tests.

3 Operation tests of thrusters are to be carried out.

4 Function tests on those arrangements specified in **20.9.1** are to be carried out (excluding those discharging devices specified in **20.9.1-1**).

Chapter 21 SELECTIVE CATALYTIC REDUCTION SYSTEMS AND ASSOCIATED EQUIPMENT

21.1 General

21.1.1 Application

1 The requirements in this chapter apply to selective catalytic reduction systems (hereinafter referred to as “SCR systems”) and associated equipment.

2 Urea based ammonia (e.g. AUS 40 (a 40% urea and 60% water aqueous urea solution) specified in *ISO* 18611-1:2014) is to be used as reductant agent in SCR systems. In cases where another reductant agent is used, however, special consideration is to be given to such systems in accordance with their respective designs as well as the following **(1)** and **(2)**:

(1) Aqueous ammonia (28% or less concentration of ammonia by weight) is not to be used as a reductant agent in SCR systems except in cases where it can be demonstrated that it is not practicable to use a urea based reductant agent.

(2) Anhydrous ammonia (99.5% or greater concentration of ammonia by weight) is not to be used as a reductant agent in SCR systems except in cases where the flag administration agrees to its use and the following **(a)** and **(b)** can be demonstrated:

  (a) It is not practicable to use an aqueous urea solution.

  (b) It is not practicable to use an aqueous ammonia.

3 In cases where a reductant agent specified in **(1)** or **(2)** of **-2** above is used, arrangements for its loading, carriage and use are to be derived from a risk based analysis.

4 When reductant agent tanks with volume below of 500 *litres* and urea based ammonia (e.g. AUS 40 (a 40 % urea and 60 % water aqueous urea solution) specified in *ISO* 18611-1:2014) specified in **-2** above is used as a reductant agent, the requirements for the reducing agent tanks are to be as deemed appropriate by the Society.

5 In addition to the requirements in this chapter, the Society may apply special requirements as instructed by the flag administration of the ship or the governments of sovereign nations whose waters the ship navigates.

21.1.2 Terminology

The terms used in this Chapter are defined as follows:

(1) “SCR system” means a system consisting of a SCR chamber and a reductant agent injection system.

(2) “SCR chamber” means an integrated unit containing one or more catalyst blocks into which flows exhaust gas from diesel engines without outflow and which receives its supply of the reductant agent from a reductant agent injection system.

(3) “Catalyst block” means a block of certain dimensions through which exhaust gas passes and which contains catalysts on its inside surfaces which reduce the NOx content of exhaust gas.

(4) “Reductant agent injection system” means a system which consists of equipment such as pumps for supplying reductant agents to nozzles, nozzles for injecting reductant agents and device(s) for controlling the flow rates of the reductant agents injected by the nozzles.

21.1.3 Drawings and Data to be Submitted

Drawings and data to be submitted are generally as follows:

(1) Plans and documents for approval

  (a) Particulars

  (b) Specifications

  (c) Material specifications

  (d) General arrangement

  (e) SCR chamber construction, including the arrangement of catalyst blocks

  (f) Reductant agent storage tank construction and their arrangements

  (g) Ventilation systems for compartments installed with equipment for using or handling reductant agent, such as its storage tanks, or for the compartments specified in **21.4.2-3**.

  (h) Detailed arrangements of injection nozzles of reductant agent injection systems

(i) Piping diagram
(j) Arrangements of control systems and diagram of hydraulic and electrical systems, including safety systems and alarm systems
(k) Plans and documents concerning automation
i) List of measuring points
ii) List of alarm points
iii) Control systems and safety systems (list of controlled objects and controlled variables, list of conditions for safety systems, and kinds of control energy sources such as self-actuated, pneumatic and electric)
(l) The construction, arrangement and diagrams of electrical systems, including safety systems and alarm systems, of exhaust gas heating devices, if fitted
(m) Plans and documents for the control and monitoring systems of SCR systems, if the ships are provided with monitoring and control systems for periodically unattended machinery spaces.
(n) Other drawings considered necessary by the Society

(2) Plans and documents for reference
(a) Operation manual for SCR systems
(b) Operation manual for automatic control and safety systems
(c) Documents related to allowable back pressure
(d) Documents related to any studies and corresponding results explaining cases where bypass pipes are not fitted for SCR systems in accordance with **21.3.1-2(1)**
(e) Engineering analysis such as Failure Mode Effect Analysis (FMEA)
(f) Other drawings considered necessary by the Society

21.2 Design

21.2.1 General Requirements

1 In addition to the requirements in this Chapter, pipes, valves, pipe fittings and auxiliaries are to satisfy the requirements in **Chapter 12**. In such cases, the term “sea water” is to be read as “reductant agent”. However, when applying **Table D12.1** and when piping materials are selected according to *ISO* 18611-3:2014, “urea in SCR systems” is to be applied as “type of medium”.

2 In addition to the requirements in this Chapter, air pipes and sounding pipes are to satisfy the requirements in **13.6** and **13.8**. In such cases, the term “fuel oil” is to be read as “reductant agent”.

3 In addition to the requirements in this Chapter, the control systems, safety systems and alarm systems of reductant agent injection systems are to satisfy the requirements in **Chapter 18**.

4 Appropriate means are to be provided to allow continuous proper operation of diesel engines which are connected to SCR systems in case where a single component of the system or associated equipment fails or becomes otherwise inoperable.

21.2.2 Material

1 Reductant tanks are to be of steel or other equivalent material with a melting point above 925 °C.

(Note)

The wording “to be of steel or other equivalent material” is not applicable for integral tanks on FRP vessels such as those listed below, provided that the integral tanks are coated and/or insulated with a self-extinguishing material.

(1) FRP vessels complying with Regulation 17 of *SOLAS* Chapter II-2 based upon its associated IMO guidelines (*MSC*.1/*Circ*.1574), and

(2) FRP vessels exempted from the application of *SOLAS* e.g., yachts, fast patrol, navy vessels, etc., generally of less than 500 gross tonnage, subject to yacht codes or flag regulations.

2 Pipes/piping systems (including pumps, valves, vents, other parts and their joints) are to be of steel or other equivalent material with melting point above 925 °C, except downstream of the tank valve, provided this valve is metal seated and arranged as fail-to-closed or with quick closing from a safe position outside the space in the event of fire; in such cases, approved plastic piping in accordance with **Annex 12.1.6** may be accepted even if it has not passed a fire endurance test.

3 Reductant tanks and pipes /piping systems (including pumps, valves, vents, other parts and their joints) are to be made with a material compatible with reductant or coated with appropriate anti-corrosion coating.

4 Material used for exhaust gas heating devices is to be deemed appropriate by the Society.

21.3 SCR systems

21.3.1 SCR chamber

1 Consideration of Exhaust Gas Allowable Back Pressure and Temperature

SCR chambers suitable for diesel engines are to be installed, and the systems are to be arranged on exhaust gas pipes so that the back pressure and temperature do not exceed the allowable limits specified by the diesel engine manufacturer.

2 Changeover of Exhaust Gas Pipes

(1) In cases of SCR system failure as well as any blocking or clogging of SCR chambers, bypass pipes are to be provided except for such diesel engines connected to systems that can be satisfactory operated under the possible operating ranges of the engines without bypass pipes in the event of back pressure increases due to such a failure or blocking or clogging.

(2) For diesel engines with changeover arrangements from exhaust gas pipes in which a SCR chamber is installed to bypass pipes, changeover devices for those pipes are to be fitted at the branch positions of the pipes.

(3) The devices specified in **(2)** above are to be fitted with appropriate means to prevent the simultaneous closing of the exhaust pipes in which the SCR chamber is installed and bypass pipes, such as interlock devices so that the proper operation of the diesel engines emitting exhaust gas will be maintained.

(4) The devices specified in **(2)** above are to be provided with indicators which show which exhaust gas pipe is being used. These indicators are to be fitted at both local positions and control stations of SCR systems.

3 Maintenance Considerations

(1) Catalyst blocks are to be arranged so that they can be easily replaced.

(2) Sufficient space for replacing catalyst blocks is to be provided on board ship.

4 Maintaining the Quality of Catalytic Reactions

Consideration is to be given to SCR chambers so that any degradation of catalytic reactions due to the adherence of soot, etc. is prevented.

21.3.2 Reductant Agent Injection Systems

1 Injection Control

Reductant agent injection systems are to be fitted with interlock devices so that the reductant solution cannot be injected in cases where the temperature of exhaust gas at the inlet of the SCR chamber is below the design temperature specified by the manufacturer.

2 Injection Amount Monitoring

Arrangements are to be provided to monitor the amount of reductant agent injected during use of the SCR system at control stations.

3 Injection Position

The reductant agent is to be injected so that hydrolysis is achieved after the injection and the appropriate denitration reaction is produced in the chamber.

4 Safety Devices and Alarm Devices

The reductant agent injection system is to be fitted with alarm devices and safety devices to stop the injection of reductant agent when the temperature at the outlets of engines or the inlets of SCR chambers exceed preset levels in order to avoid any self-ignition of ammonia gas caused by abnormal increases in exhaust gas temperatures.

21.4 Requirements for Construction and Arrangements, etc.

21.4.1 Construction and Arrangement

1 Reductant agent storage tanks may be located within the engine room.

2 Reductant agent storage tanks are to be protected from excessively high or low temperatures applicable to the particular concentration of the solution. Depending on the operational area of the ship, this may necessitate the fitting of heating and/or cooling systems. The physical conditions recommended by applicable recognized standards (such as *ISO* 18611-3:2014) are to be taken into account to ensure that the contents of the reductant agent tank are maintained to avoid any impairment of the reductant agent during storage.

3 The reductant agent storage tank is to be arranged so that any leakage will be contained and prevented from making contact with heated surfaces. All pipes or other tank penetrations are to be provided with manual closing valves attached to the tank.

4 Storage tanks for reductant agents as well as any equipment using or handling reductant agents, such as reductant agent injection systems, are to be so arranged to prevent the spread of any spillage in the compartments where they are installed. For example, drip trays of a sufficient size are to be provided under such tanks and equipment.

5 Where reductant agent is stored in tanks which form part of the ship's hull, the following **(1)** to **(5)** are to be considered during the design and construction:

(1) These tanks may be designed and constructed as integral part of the hull, (e.g. double bottom, wing tanks).

(2) These tanks are to be coated with appropriate anti-corrosion coating.

(3) These tanks are to be designed and constructed as per the structural requirements applicable to hull and primary support members for deep tank construction after taking into account the specific gravity of reductant agent.

(4) These tanks are to be segregated by cofferdams, void spaces, pump rooms, empty tanks or other similar spaces so as to not be located adjacent to accommodation or service spaces, cargo spaces containing cargoes which react with reductant agent in a hazardous manner as well as any food stores, oil tanks and fresh water tanks.

(5) These tanks are to be included in the ship's stability calculation.

6 Piping for reductant agent and venting systems are to be independent of other ship service piping and/or systems.

7 Piping systems for reductant agent are not to pass through or to extend into accommodation, service spaces, or control stations.

8 Piping systems for reductant agents are not to pass through or extend into any storage tanks for other liquids, except in cases where deemed appropriate by the Society.

9 The piping systems for reductant agents, excluding those near reductant agent injection nozzles, are not to be located immediately above or near equipment operating at high temperatures such as boilers, steam pipelines and exhaust gas pipes, etc. which are required to be insulated. As far as practicable, such piping systems are to be arranged far from hot surfaces, electrical installations and other sources of ignition.

10 In cases where a reductant agent is produced from solid matter on board, the solid matter is to be stored at an appropriate location in consideration of the storage conditions specified by the manufacturer.

21.4.2 Closing Devices and Shut-down Systems

1 Reductant agent supply piping, which, if damaged, would allow reductant agent to escape from storage tanks situated above the double bottom, is to be fitted with a cock or valve directly on the tank capable of being closed from a safe position outside the space in which such tanks are situated in the event of a fire occurring in such a space. In the case of storage tanks situated in any shaft or pipe tunnel or similar space, valves on the tank are to be fitted, but an additional valve on the pipe or pipes outside the tunnel or similar space may be so fitted as to prevent the reductant agent from escaping in the event of fire. If such an additional valve is fitted in a machinery space, the valve is to be capable of being operated from a position outside said machinery space.

2 Reductant agent supply pumps are to be provided with stopping devices installed inside the space in which they are installed and, in addition, in a location outside such a space which will not be cut off in the event of fire in said space.

3 In cases where exhaust gas heating devices fitted with burners and blowers are installed, stopping devices for the burners and blowers are to be installed inside the space in which they are installed and, in addition, in a location outside such a space which will not be cut off in the event of fire in said space.

21.4.3 Ventilation Systems

1 If storage tanks for reductant agent or equipment for using or handling reductant agent, such as reductant agent injection systems, is installed in a closed compartment, the area is to be served by an effective ventilation system of extraction type providing not less than 6 air changes per hour which is independent from the ventilation system of accommodation, service spaces, or control stations. The ventilation system is to be capable of being controlled from outside the compartment. A warning notice requiring the use of such ventilation before entering the compartment is to be provided outside the compartment adjacent to each point of entry.

2 Notwithstanding the requirements specified in **-1** above, where storage tanks for reductant agent or equipment for using or handling reductant agent, such as the reductant agent injection systems are located within an engine room a separate ventilation system is not required when the general ventilation system for the space is arranged so as to provide an effective movement of air in the vicinity of the storage tank and equipment and is to be maintained in operation continuously except when the storage tank is empty and has been thoroughly ventilated.

3 The requirements specified in **-1** and **-2** above also apply to closed compartments normally entered by persons in accordance with the following **(1)** or **(2)**:

(1) When they are adjacent to the urea integral tanks and there are possible leak points (e.g. manhole, fittings) from these tanks; or

(2) When the urea piping systems pass through these compartments, unless the piping system is made of steel or other equivalent material with melting point above 925 °C and with fully welded joints.

21.4.4 Venting Systems of Reductant Agent Storage Tank

1 Reductant agent storage tanks are to be arranged so that they can be emptied of urea and ventilated by means of portable or permanent systems.

2 The vent pipes of reductant agent storage tanks are to terminate in a safe location on the weather deck in consideration of the emission of ammonia gas from the vent outlets in the event of fire near the tanks. Tank venting systems are to be arranged to prevent entrance of water into reductant agent storage tanks.

21.4.5 Safety Devices and Alarm Devices

1 In cases where changeover devices for exhaust gas pipes are fitted, devices which automatically open bypass sides of the changeover devices in the event of any of the following **(1)** and **(2)** failures are to be fitted. The above changeover devices are also to be operated within allowable limits of engine back pressure.

(1) Abnormal increases of the exhaust gas pressures at the inlet or the differential pressures across the catalyst blocks

(2) Abnormal increase of the exhaust gas temperature at the outlet (However, alarms may be omitted in cases where means are provided to prevent damage by soot fire.)

2 Alarm devices, to be activated in the event of any of the abnormal conditions given in **Table D21.1**, are to be provided at control stations of SCR systems.

3 SCR systems are to be fitted with monitoring devices at control stations of SCR systems, and these devices are to be capable of indicating the information listed in the following **(1)** to **(4)**:

(1) Liquid levels in tanks for reductant agent

(2) Temperatures in tanks for reductant agent

(3) Exhaust gas temperatures at inlets

(4) Pressures at inlets or differential pressures across catalyst block

4 In addition to the requirements given in **-1** to **-3** above, additional safety, alarm and monitoring systems may be required to be fitted based upon engineering analysis results, such as Failure Mode Effect Analysis (FMEA), for SCR systems.

Table D21.1 Alarm points for SCR system(1)

Monitored Variables	
Liquid levels in tank for reductant agent	H L
Temperature in tank for reductant agent	H L
Exhaust gas pressure at inlet (2)	H
Exhaust gas temperature at inlet	H
Exhaust gas temperature at outlet(3)	H
Power loss of control, alarm, monitoring or safety devices	○

Notes:

(1) "H" and "L" mean "high" and "low". "○" means abnormal condition occurred.

(2) Differential pressure across catalyst block may be accepted in lieu.

(3) Alarms may be omitted in cases where means are provided to prevent damage by soot fire.

21.5 Electrical Installations

21.5.1 General

1 Capacities of main sources of electrical power are to cover maximum electric demand during SCR system operation, including normal seagoing conditions, cargo loading and unloading conditions, and departure and arrival conditions.

2 For items not specified in **-1** above, electrical installations are to comply with relevant requirements specified in **Part H**.

21.6 Exhaust Gas Heating Device

21.6.1 General

1 In cases where exhaust gas heating devices equipped with burners are installed for the purpose of raising the temperatures of the exhaust gas from engines, the requirements in **21.6.2**, **21.6.3** and **21.6.4** are to be complied with.

2 Exhaust gas heating devices which are not equipped with burners are to conform to requirements deemed appropriate by the Society.

21.6.2 Construction and Arrangement

1 Exhaust gas heating devices are to be so arranged that the pressure in exhaust gas pipes does not exceed the exhaust gas allowable back pressure specified by the engine manufacturer.

2 Appropriate measures are to be taken to prevent the frames of burners from coming in direct contact with the exhaust gas from the engines.

3 Appropriate measures are to be taken to prevent any unburnt fuel from engines from entering into exhaust gas heating devices when the SCR system is not in use. In cases where an on-off damper is installed in the flue gas line of the exhaust gas heating device, an indicator which shows the condition of the damper is to be provided.

4 Temperature measurement devices for the combustion gas at the outlets of exhaust gas heating devices or the exhaust gas at the inlets of SCR chambers are to be provided.

5 A blower of adequate capacity is to be so provided that the temperature of the exhaust gas rises to the required level.

6 Combustion chambers and gas flue lines of exhaust gas heating device are to comply with the following **(1)** and **(2)**:

(1) Main parts of combustion chambers are to be constructed with appropriate materials.

(2) Means to inspect and clean combustion chambers and flue lines are to be provided.

7 The construction and control of burners are to comply with the following **(1)** to **(5)**:

(1) The fuel supply is to be appropriately controlled so that the temperature of the exhaust gas from engines is heated to a temperature in which the catalysis is able to effectively function.

(2) They are to be so arranged that the combustion chamber is capable of being pre-purged before ignition.

(3) They are to be so arranged that the fuel supply does not precede the operation of the ignition system in cases where an automatic ignition system is adopted.

(4) They are to be capable of controlling the amount of fuel supplied in cases where an automatic fuel supply system is provided.

(5) The ignition of the main burner and pilot burner, etc. is to follow their planned sequence in cases where an automatic combustion control device is provided.

21.6.3 Installation Considerations

1 Exhaust gas heating devices are to be so installed as to minimize the effects of the following loads or external forces:

(1) ship motions or any vibrations caused by machinery installations;

(2) external forces caused by the piping or any other supports fitted onto the exhaust gas heating device; and

(3) thermal expansions due to temperature fluctuation.

21.6.4 Safety Devices and Alarm Devices

1 Each exhaust gas heating device is to be fitted with a safety device which automatically shuts off the fuel supply to all burners in any of the following **(1)** and **(2)** cases:

(1) when the temperature of combustion gas at the outlet of the exhaust gas heating device or exhaust gas temperature at the inlet of SCR chamber is above or below the preset temperature for normal operation of the SCR system; or

(2) when the flame is extinguished.

2 Each exhaust gas heating device is to be fitted with an alarm device which operates in any of the following **(1)** to **(6)** cases:

(1) when the temperature of combustion gas at the outlet of the exhaust gas heating device or exhaust gas temperature at the inlet of SCR chamber is above or below the preset temperature for normal operation of the SCR system;

(2) when the flame is extinguished;

(3) when the power supply to the alarm device is stopped;

(4) when the fuel injection pressure to the furnace falls, in the case where fuel supply is of pressure injection type;

(5) when the blowers stop; or

(6) other cases deemed necessary by the Society.

21.7 Safety and Protective Equipment

21.7.1 General

For the protection of crew members, the ship is to have on board at least the following suitable protective equipment and installations. Locations and numbers of the equipment and installations are to be derived from the detailed installation arrangements. Locations where such equipment is stored or installed are to be clearly marked so as to be easily identifiable.

(1) Personnel protective equipment

(a) Large apron of chemical-resistant material

(b) Special gloves with long sleeves

(c) Suitable footwear

(d) Suitable protective equipment consisting of coveralls and tight-fitting goggles or face shields or both

(2) Self-contained breathing apparatus (capable of functioning for at least 30 minutes)

(3) Eyewash

(4) Stretcher

21.8 Tests

21.8.1 Tests at Facilities (Shop tests)

1 Reductant agent independent storage tanks are to be subjected to hydrostatic tests at a pressure corresponding to a water head of 2.5 *m* above the top plate.

2 After completion of the fabrication process, piping, valves and pipe fittings, containing reductant agent, the design pressure of which exceeds 0.35 *MPa* are to be subjected to hydrostatic tests together with the welded fittings at a pressure equal to 1.5 times the design pressure.

3 The pressure parts of reductant agent supply pumps are to be subjected to hydrostatic tests at a pressure equal to 1.5 times the design pressure or 0.2 *MPa*, whichever is greater. Tests carried out in the presence of the Surveyor may be replaced by manufacturer's tests. In such cases, submission or presentation of test records may be required by the Society.

4 For reductant agent supply pumps, shop trials are to be carried out according to test procedures deemed appropriate by the Society. Tests carried out in the presence of the Surveyor may be replaced by manufacturer's tests. In such cases, submission or presentation of test records may be required by the Society.

5 Electrical motors and their corresponding control gears used for pumps fitted on SCR systems are to be tested in accordance with relevant requirements in **Part H**. Shop tests for electrical motors whose continuous rated capacities are less than 100 *kW* and their corresponding control gears may be replaced by manufacturer tests. In such cases, submission or presentation of test records may be required by the Society.

21.8.2 Tests after Installation On Board

1 In cases where reductant agent is carried in tanks which form part of the ship's hull, the tanks are to be subjected to hydrostatic tests in accordance with item10(1), **Table B2.7, Part B**. Where the specific gravities of the liquids used for the tests are less than those of the reductant agent, an appropriate additional head is to be considered.

2 After installation on board, SCR systems are to be tested in accordance with the following **(1)** to **(4)**:

(1) Piping systems for reductant agent are to be subjected to leak tests at pressures equal to 1.5 times the design pressure or 0.4 *MPa*, whichever is greater.

(2) Operation tests of SCR systems are to be carried out at maximum quantities of emitted exhaust gas.

(3) Performance tests for control, safety and alarm devices are to be carried out

(4) Operation tests for changeover devices of exhaust gas pipes and the corresponding indicators are to be carried out.

Chapter 22 EXHAUST GAS CLEANING SYSTEMS AND ASSOCIATED EQUIPMENT

22.1 General

22.1.1 Application

1 The requirements in this chapter apply to exhaust gas cleaning systems and associated equipment installed to reduce sulphur oxides and particular matter emitted from fuel oil combustion units such as reciprocating internal combustion engines and boilers, and which use sodium hydroxide solutions or calcium hydroxide (hereinafter referred to "chemical treatment fluid" in this chapter) that has corrosive properties or which are otherwise considered to represent a hazard to personnel.

2 In cases where exhaust gas cleaning systems which use chemical agents other than those specified in **-1** above are used, safety measures are to be taken according to the result of a risk assessment to be conducted to analyse the risks, in order to eliminate or mitigate the hazards to personnel brought by the use of such exhaust gas cleaning systems, to an extent equivalent to systems complying with this chapter.

3 In cases where exhaust gas cleaning systems which do not use chemical agents are used, the term "liquids containing chemical treatment fluid" is to be read as "liquids which have passed through scrubber chambers"; this, however, does not apply to **22.4.1-4**, **-9** and **-10**, **22.7.1-2** and **22.7.2-2(1)**.

4 Exhaust gas cleaning systems and associated equipment used in exhaust gas recirculation systems are to comply with **Chapter 23**.

5 In addition to the requirements in this Chapter, the Society may apply special requirements as instructed by the flag administration of the ship or the governments of sovereign nations whose waters the ship navigates.

22.1.2 Terminology

The terms used in this Chapter are defined as follows:

(1) "Exhaust gas cleaning system" means a system which consists of storage tanks for residues, etc., washwater supply pumps, chemical treatment fluid supply pumps, washwater injection systems and scrubber chambers.

(2) "Scrubber chamber" means an integrated unit which discharges the washwater, into which flows exhaust gas from fuel oil combustion units and which receives the washwater supply from the washwater injection system.

(3) "Washwater" means freshwater or sea water (including cases where sodium hydroxide or calcium hydroxide is added) which is injected into scrubber chambers or exhaust gas inlets, and includes liquids which have passed through scrubber chambers.

(4) "Washwater injection systems" means a system which consists of equipment such as pumps for supplying washwater to nozzles, nozzles for spraying washwater and devices for controlling the flow rates.

(5) "Residue" means a substance generated by exhaust gas cleaning systems resulting from the cleaning of exhaust gas, except for any liquids allowed to be discharged overboard.

22.1.3 Drawings and Data to be Submitted

Drawings and data to be submitted are generally as follows:

(1) Plans and documents for approval
   - (a) Particulars
   - (b) Specifications
   - (c) Material specifications
   - (d) General arrangement
   - (e) Construction of scrubber chamber
   - (f) Construction of storage tanks for chemical treatment fluid or liquid containing chemical treatment fluid and their arrangements.
   - (g) Ventilation systems for compartments installed with equipment for using or handling chemical treatment fluid, such as storage tanks, or for the compartments specified in **22.4.2-3**
   - (h) Piping diagrams (including details of watertight bulkheads and penetrations of fire-resisting divisions)

(i) Arrangements of control systems and diagrams of hydraulic and electrical systems, including safety systems and alarm systems
(j) Plans and documents concerning automation
i) List of measuring points
ii) List of alarm points
iii) Control systems and safety systems (list of controlled objects and controlled variables, list of conditions for safety systems, and kinds of control energy sources such as self-actuated, pneumatic and electric)
(k) Plans and documents for the control and monitoring systems of exhaust gas cleaning systems, if the ships are provided with monitoring and control systems for periodically unattended machinery spaces
(l) Other drawings considered necessary by the Society
(2) Plans and documents for reference
(a) Operation manual for exhaust gas cleaning systems
(b) Operation manual for automatic control and safety systems
(c) Documents related to allowable back pressure
(d) Documents related to any studies and corresponding results explaining cases where bypass pipes are not fitted for exhaust gas cleaning systems in accordance with **22.3.1-3(1)**
(e) Engineering analysis such as Failure Mode Effect Analysis (FMEA)
(f) The results of risk assessments conducted to analyse the risks specified in **22.1.1-2**
(g) Other drawings considered necessary by the Society

22.2 Design

22.2.1 General Requirements

1 In addition to the requirements in this chapter, pipes, valves, pipe fittings and auxiliaries are to satisfy the requirements in **Chapter 12**. In such cases, the term "sea water" is to be read as "liquids containing chemical treatment fluid". However, regardless of design pressure and temperature, piping systems containing chemical treatment fluids only are to comply with the requirements applicable to Class I piping systems specified in **Chapter 12**. As far as practicable, e.g. except for the flange connections that connect to tank valves, the piping systems are to be joined by welding.

2 In addition to the requirements in this chapter, air pipes and sounding pipes are to satisfy the requirements in **13.6** and **13.8** (excluding **13.6.1-6** and **13.6.2-3**). In such cases, the term "fuel oil" is to be read as "liquids containing chemical treatment fluid".

3 In addition to the requirements in this Chapter, the control systems, safety systems and alarm systems of exhaust gas cleaning systems are to satisfy the requirements in **Chapter 18**.

4 Appropriate means are to be provided to allow continuous proper operation of fuel oil combustion units such as reciprocating internal combustion engines and boilers which are connected to exhaust gas cleaning systems in case where a single component of the system or associated equipment fails or becomes otherwise inoperable.

22.2.2 Material

1 Materials used for exhaust gas cleaning systems are to be selected in consideration of notch ductility at operating temperatures and pressures, their corrosive effects and the possibility of hazardous reactions.

2 Storage tanks, pipes/piping systems and drip trays for chemical treatment fluids which transfer undiluted chemical treatment fluids are to be of steel or other equivalent material with a melting point above 925 °C.

3 Storage tanks and pipes/piping systems for chemical treatment fluids are to be made with a material compatible with chemical treatment fluids, or coated with appropriate anti-corrosion coating.

Note:
Several metals are incompatible with the chemical treatment fluids, e.g. NaOH is incompatible with zinc, aluminum, etc.

22.3 Exhaust Gas Cleaning Systems

22.3.1 Construction of Exhaust Gas Cleaning Systems

1 Considerations for exhaust gas allowable back pressure and temperature

Exhaust gas cleaning systems suitable for fuel oil combustion units are to be installed, and the systems are to be arranged so that the back pressure and temperature do not exceed the allowable limits specified by the fuel oil combustion unit manufacturer.

2 Considerations for exhaust gas heating

Exhaust gas cleaning systems are to be provided with suitable means to ensure the systems do not suffer any damage caused by exhaust gas heating even when the exhaust gas cleaning system is not cleaning exhaust gas with washwater, or are to be provided with devices at their exhaust gas inlets to shut down the exhaust gas supply.

3 Changeover of exhaust gas pipes

(1) In cases of exhaust gas cleaning system failure as well as any blocking or clogging of scrubber chambers, bypass pipes are to be provided except for such fuel oil combustion units connected to systems that can be satisfactory operated under the possible operating ranges of the units without bypass pipes in the event of back pressure increases due to such a failure or blocking or clogging.

(2) For fuel oil combustion units with changeover arrangements from exhaust gas pipes in which a scrubber chamber is installed to bypass pipes, changeover devices for those pipes are to be fitted at the branch positions of the pipes.

(3) The devices specified in **(2)** above are to be fitted with appropriate means to prevent the simultaneous closing of the exhaust pipes in which the scrubber chamber is installed and bypass pipes, such as interlock devices so that the proper operation of the fuel oil combustion units emitting exhaust gas will be maintained.

(4) The devices specified in **(2)** above are to be provided with indicators which show which exhaust gas pipe is being used. These indicators are to be fitted at both local positions and control stations.

4 Prevention of reverse flow of washwater

Exhaust gas cleaning systems are to be fitted with appropriate means to prevent the reverse flow of washwater from scrubber chambers to fuel oil combustion units.

5 Arrangement of pipes for overboard discharge~~s~~

(1) Overboard discharge from exhaust gas cleaning systems are not to be interconnected to other systems.

(2) Due consideration is to be given to the location of overboard discharge with respect to vessel propulsion features, such as thrusters, propellers or as in **13.3.2-1** to prevent discharge water from falling onto survival craft (lifeboats and liferafts) when abandoning ship.

6 Prohibition of connection of exhaust gas pipes

In principle, exhaust gas pipes of fuel oil combustion units, such as reciprocating internal combustion engines and boilers, are not to be connected to common exhaust gas cleaning systems except where exhaust pipes of more than one fuel oil combustion units are required to be connected to common exhaust gas cleaning systems and the systems satisfy the following requirements in addition to **-3**.

(1) The exhaust gas cleaning systems are to be fitted with appropriate devices to prevent the reverse flow of exhaust gas to fuel oil combustion units such as other engines and boilers.

(2) The devices specified in **(1)** above are to be fitted with appropriate means to prevent the simultaneous closing of the bypass pipes and the exhaust pipes in which the scrubber chamber is installed, such as interlock devices so that the proper operation of the fuel oil combustion units, such as engines and boilers, emitting exhaust gas will be maintained.

(3) The devices specified in **(1)** above are to be provided with indicators which show which exhaust gas pipe is being used. These indicators are to be fitted at both local positions and control stations.

(4) Safety measures are to be provided for preventing the propagation of fire between fuel oil combustion units, such as reciprocating internal combustion engines and boilers, connected to common exhaust gas cleaning systems.

22.4 Requirements for Construction and Arrangements, etc.

22.4.1 Construction and Arrangement

1 Chemical treatment fluids storage tanks may be located within the engine room.

2 Chemical treatment fluids storage tanks are to be protected from excessively high or low temperatures applicable to the particular concentration of the fluids. Depending on the operational area of the ship, this may necessitate the fitting of heating and/or cooling systems.

3 Drip trays of a sufficient size are to be provided under storage tanks for liquids containing chemical treatment fluids as well as any equipment using or handling such liquids, such as pumps, to prevent the spread of any spillage in the compartments where they are installed.

4 The drip trays specified in **-3** above are to be fitted with drain pipes which lead to appropriate tanks, such as residue tanks, which are fitted with high level alarm, or are to be fitted with alarms for leak detection.

5 The storage tank for chemical treatment fluids is to be arranged so that any leakage will be contained and prevented from making contact with heated surfaces. All pipes or other tank penetrations are to be provided with manual closing valves attached to the tank. In cases where such valves are provided below top of tank, they are to be arranged with quick acting shutoff valves which are to be capable of being remotely operated from a position accessible even in the event of chemical treatment fluid leakages.

6 The storage tanks are to have sufficient strength to withstand a pressure corresponding to the maximum height of a fluid column in the overflow pipe, with a minimum of 2.4 *m* above the top plate taking into consideration the specific density of the treatment fluid.

7 Where chemical treatment fluids is stored in tanks which form part of the ship's hull, the following **(1)** to **(4)** are to be considered during the design and construction:

(1) These tanks may be designed and constructed as integral part of the hull, (e.g. double bottom, wing tanks).

(2) These tanks are to be coated with appropriate anti-corrosion coating and are to be segregated by cofferdams, void spaces, pump rooms, empty tanks or other similar spaces so as to not be located adjacent to accommodation, cargo spaces containing cargoes which react with chemical treatment fluids in a hazardous manner as well as any food stores, oil tanks and fresh water tanks.

(3) These tanks are to be designed and constructed as per the structural requirements applicable to hull and primary support members for deep tank construction after taking into account the specific gravity.

(4) These tanks are to be included in the ship's stability calculation.

8 The chemical treatment fluid piping and venting systems are to be independent of other ship service piping and/or systems.

9 The chemical treatment fluid piping systems are not to pass through or to extend into accommodation, service spaces, or control stations.

10 Piping systems for liquids containing chemical treatment fluids are not to pass through or to extend into any storage tanks for other liquids, except where deemed appropriate by the Society.

11 Piping systems for liquids containing chemical treatment fluids, excluding those near nozzles spraying washwater, are to be so arranged to prevent any outflows or leakage from the piping system from coming into contact with any high temperature equipment surfaces. Such piping systems are especially not to be located immediately above or near equipment such as boilers, steam pipes or exhaust gas pipes.

12 Storage tanks for liquids containing chemical treatment fluids are to satisfy the following requirements:

(1) The tanks are to be so arranged to prevent liquids containing chemical treatment fluids escaping or leaked from the tanks from coming into contact with high temperature equipment surfaces. Such tanks are especially not to be located immediately above or near equipment such as boilers, steam pipes or exhaust gas pipes.

(2) In cases where shore connections with standard couplings are fitted onto filling-up pipe lines, proper protection against any spraying of chemical treatment fluids, such as effective enclosures, is to be provided in consideration of the sodium hydroxide solution spraying out during filling-up operations.

13 Discharge pipes from storage tanks for liquids containing chemical treatment fluids are to be fitted with stop valves directly on the tank.

14 Tanks for residues generated from the exhaust gas cleaning process are to satisfy the following requirements:

(1) The tanks are to be independent from other tanks, except in cases where these tanks are also used as the over flow tanks for chemical treatment fluids storage tank.

(2) Manholes or access holes in a sufficient size are to be provided at such locations that each part of the tank can be cleaned without difficulties.

(3) Tank capacities are to be decided in consideration of the number and kinds of installed exhaust gas cleaning systems as well as the maximum number of days between ports where residue can be discharged ashore. In the absence of precise data, a figure of 30 *days* is to be used.

(4) Where residue tanks used in closed loop chemical treatment systems are also used as the overflow tanks for chemical treatment fluids storage tank, the requirements for storage tanks apply.

15 Piping systems for washwater used in scrubber chambers are to be constructed of corrosion resistance materials or are to be otherwise appropriately protected, taking into account the corrosive effects of the water.

16 For piping systems for washwater used in scrubber chambers, where materials other than hull construction materials are used and where two or more kinds of different metallic materials are arranged adjacent to each other, appropriate measures are to be taken to prevent bimetallic corrosion (galvanic corrosion).

17 In case distance piece is fitted to the piping system specified in **-15** above, it is to be made of corrosion resistant steel material or be coated with an anti-corrosive material suitable for the operating environment. In addition, the thickness of the distance piece is to be at least the minimum values specified in the following **(1)** or **(2)**. If the values specified in the following **(1)** and **(2)** do not exist as standardised values, the thickness specified in piping standard Schedule 160 (Sch.160) are, as far as practicable, to be used instead.

(1) 12 *mm* in cases where complete pipe is made of corrosion resistant material steel.

(2) 15 *mm* of mild steel in cases where the inside the pipe is treated with an anticorrosive coating or fitted with a sleeve of corrosion resistant material.

18 The following connections on piping systems only for chemical treatment fluids are to be screened or provided with other appropriate means, and fitted with drip trays to prevent the spread of any spillage where they are installed:

(1) Detachable connections between pipes (flanged connections, mechanical joints, etc.);

(2) Detachable connections between pipes and equipment such as pumps, strainers, heaters, valves; and

(3) Detachable connections between equipment mentioned in **(1)** and **(2)** above.

19 The drip trays specified in **-18** above are to be fitted with drain pipes which lead to appropriate tanks, such as the residue tanks specified in **-14** above, which are fitted with high level alarm, or are to be fitted with alarms for leak detection. In cases where such tank is an integral tank, **-7(1)** and **(2)** above are to be applied to the tank (the term "these tanks" specified in **-7(1)** and **(2)** is to be read as "appropriate tanks, such as residue tanks").

22.4.2 Ventilation Systems

1 If storage tanks for chemical treatment fluids is installed in a closed compartment, the area is to be served by an effective mechanical ventilation system of extraction type providing not less than 6 air changes per hour which is independent from the ventilation system of other spaces. The ventilation system is to be capable of being controlled from outside the compartment. A warning notice requiring the use of such ventilation before entering the compartment is to be provided outside the compartment adjacent to each point of entry.

2 Notwithstanding the requirements specified in **-1** above, where storage tanks for chemical treatment fluids are located within an engine room a separate ventilation system is not required when the general ventilation system for the space is arranged so as to provide an effective movement of air in the vicinity of the storage tank and equipment and is to be maintained in operation continuously except when the storage tank is empty and has been thoroughly air purged.

3 The requirements specified in **-1** also apply to the following closed compartments normally entered by persons:

(1) when they are adjacent to the integral storage tank for chemical treatment fluids and there are possible leak points (e.g. manhole, fittings) from these tanks; or

(2) when the treatment fluid piping systems pass through these compartments, unless the piping system is made of steel or other equivalent material with melting point above 925 °C and with fully welded joints.

22.4.3 Venting Systems of Storage Tanks for Chemical Treatment Fluids

1 The vent pipes of the storage tank are to terminate in a safe location on the weather deck and the tank venting system is to be arranged to prevent entrance of water into the tank for chemical treatment fluids.

2 Storage tanks for chemical treatment fluids are to be arranged so that they can be safely emptied of the fluids and ventilated by means of portable or permanent systems.

22.4.4 Safety Devices and Alarm Devices

1 Exhaust gas cleaning systems are to be fitted with safety devices which are capable of automatically stopping exhaust gas washwater supply pumps and chemical treatment fluids pumps in the event of any of the following failures:

(1) Abnormal increase of the liquid level in the scrubber

(2) Abnormal increase of the pressure at the inlet or the differential pressure across the scrubber chamber (in cases where changeover devices for exhaust gas pipes are not fitted)

2 In cases where changeover devices for exhaust gas pipes are fitted, devices capable of automatically opening bypass sides of changeover devices in the event of any of the following failures are to be fitted.

(1) Abnormal increase of the liquid level in the scrubber

(2) Abnormal increase of the exhaust gas pressure at the inlet or the differential pressure across the scrubber chamber

(3) Abnormal increase of the exhaust gas temperature at the outlet

3 Alarm devices, to be activated in the event of any of the abnormal conditions given in **Table D22.1**, are to be provided at control stations of exhaust gas cleaning systems.

4 Exhaust gas cleaning systems are to be fitted with monitoring devices at control stations for exhaust gas cleaning systems, and these devices are to indicate the information listed in **(1)** to **(5)**:

(1) Liquid levels in scrubber chambers

(2) Liquid levels in tanks for chemical treatment fluids

(3) Temperatures in tanks for chemical treatment fluids **(**where the heating and/or cooling systems specified in **-6** are provided)

(4) Exhaust gas temperatures at outlets

(5) Pressures at inlets or differential pressures across scrubber chambers

5 In addition to the requirements given in **-1** to **-3** above, additional safety, alarm and monitoring systems may be required to be fitted based upon engineering analysis results, such as Failure Mode Effect Analysis (FMEA), for exhaust gas cleaning systems.

6 Each storage tank for chemical treatment fluids is to be provided with level monitoring arrangements and high/low level alarms. In cases where heating and/or cooling systems are provided, high and/or low temperature alarms or temperature monitoring are also to be provided accordingly.

Table D22.1 Alarm points for exhaust gas cleaning system(1)

Monitored Variables	
Liquid level in scrubber chamber	H
Temperature of washwater supply (in cases where the washwater includes sodium hydroxide solutions)	H
Liquid levels in tank for sodium hydroxide solution	H L
Temperature in tank for sodium hydroxide solution	H L
Exhaust gas pressure at the inlet (2)	H
Exhaust gas temperature at the outlet	H
Power loss of control, alarm, monitoring or safety devices	○

Notes:

(1) "H" and "L" mean "high" and "low". "○" means abnormal condition occurred.

(2) Differential pressure across scrubber chamber may be accepted in lieu.

(3) This alarm is not required when heating and/or cooling systems are not provided.

(4) Differential pressure across scrubber chamber may be accepted in lieu.

22.5 Electrical Installations

22.5.1 General

1 Capacities of main sources of electrical power are to cover maximum electric demand during exhaust gas cleaning system operation, including normal seagoing conditions, cargo loading and unloading conditions, and departure and arrival conditions.

2 For items not specified in **-1** above, electrical installations are to comply with relevant requirements specified in **Part H**.

22.6 Safety and Protective Equipment

22.6.1 General

1 For the protection of crew members, the ship is to have on board suitable personnel protective equipment. The number of personnel protective equipment carried on board is to be appropriate for the number of personnel engaged in regular handling operations or that may be exposed in the event of a failure; but in no case is there to be less than two sets available on board.

2 Personnel protective equipment is to consist of the following.

(1) Large apron of chemical-resistant material

(2) Special gloves with long sleeves

(3) Suitable footwear

(4) Suitable protective equipment consisting of coveralls and tight-fitting goggles or face shields or both

3 Eyewash and safety showers are to be provided, the location and number of eyewash stations and safety showers are to be derived from the detailed installation arrangements. As a minimum, the following stations are to be provided:

(1) In the vicinity of transfer or treatment pump locations for chemical treatment fluids. If there are multiple transfer or treatment pump locations on the same deck then one eyewash and safety shower station may be considered for acceptance provided that the station is easily accessible from all such pump locations on the same deck.

(2) An eyewash station and safety shower is to be provided in the vicinity of a chemical bunkering station on-deck. If the bunkering connections are located on both port and starboard sides, then consideration is to be given to providing two eyewash stations and safety showers, one for each side.

(3) An eyewash station and safety shower is to be provided in the vicinity of any part of the system where a spillage/drainage of chemical treatment fluids may occur and in the vicinity of system connections/components of the fluids that require periodic maintenance.

22.7 Tests

22.7.1 Tests at Facilities (Shop tests)

1 Chemical treatment fluids independent storage tanks are to be subjected to hydrostatic tests at pressures corresponding to the maximum heights of fluid columns in overflow pipes, with a minimum of 2.4 *m* above the top plate taking into consideration the specific density of the treatment fluid.

2 After completion of the fabrication process, piping, valves and pipe fittings, for liquids containing chemical treatment fluids, design pressure of which exceeds 0.35 *MPa* are to be subjected to hydrostatic tests together with the welded fittings at a pressure equal to 1.5 times the design pressure.

3 The pressure parts of chemical treatment fluids supply pumps and washwater supply pumps are to be subjected to hydrostatic tests at a pressure equal to 1.5 times the design pressure or 0.2 *MPa*, whichever is greater. Tests carried out in the presence of the Surveyor may be replaced by manufacturer's tests. In such cases, submission or presentation of test records may be required by the Society.

4 For chemical treatment fluids supply pumps and washwater supply pumps, shop trials are to be carried out according to test procedures deemed appropriate by the Society. Tests carried out in the presence of the Surveyor may be replaced by manufacturer's tests. In such cases, submission or presentation of test records may be required by the Society.

5 Electrical motors and their corresponding control gears used for chemical treatment fluids supply pumps and washwater supply pumps are to be tested in accordance with relevant requirements in **Part H**. Shop tests for electrical motors whose continuous rated capacities are less than 100 *kW* and their corresponding control gears may be replaced by manufacturer tests. In such cases, submission or presentation of test records may be required by the Society.

22.7.2 Tests after Installation On Board

1 In cases where chemical treatment fluids are carried in tanks which form part of the ship's hull, the tanks are to be subjected to hydrostatic tests in accordance with item10(1), **Table B2.7, Part B**. Where the specific gravities of the liquids used for the tests are less than those of the chemical treatment fluids, an appropriate additional head is to be considered.

2 After installation on board, exhaust gas cleaning systems are to be tested in accordance with the following:

(1) Piping systems for liquids containing chemical treatment fluids (except overboard discharge pipes) are to be subjected to leak tests at pressures equal to 1.5 times the design pressure or 0.4 *MPa*, whichever is greater.

(2) Operation tests of exhaust gas cleaning systems are to be carried out at maximum quantities of emitted exhaust gas.

(3) Performance tests for control, safety and alarm devices are to be carried out

(4) Operation tests for changeover devices of exhaust gas pipes and the corresponding indicators are to be carried out.

Chapter 23 EXHAUST GAS RECIRCULATION SYSTEMS AND ASSOCIATED EQUIPMENT

23.1.1 Application

1 The requirements in this Chapter apply to exhaust gas recirculation systems and associated equipment installed to reduce nitrogen oxides emitted from reciprocating internal combustion engines.

2 Special consideration is to be given to exhaust gas recirculation systems to which the requirements in this Chapter are not applicable in accordance with their respective designs.

3 In addition to the requirements in this Chapter, the Society may apply special requirements as instructed by the flag administration of the ship or the governments of sovereign nations whose waters ships navigate.

23.1.2 Terminology

The terms used in this Chapter, in addition to those defined in **22.1.2,** are defined as follows:

(1) "Exhaust gas recirculation systems" means systems which clean a part of exhaust gas emitted from an engine in a scrubber chamber and recirculate the cleaned exhaust gas into the engine.

23.1.3 Drawings and Data to be Submitted

Drawings and data to be submitted are generally as follows:

(1) Plans and documents for approval

The plans and documents specified in **22.1.3(1)**. In such case, "exhaust gas cleaning system" is to be read as "exhaust gas recirculation system".

(2) Plans and documents for reference

(a) The plans and documents specified in **22.1.3(2)**. In such case, "exhaust gas cleaning system" is to be read as "exhaust gas recirculation system".

(b) Specifications of blowers fitted onto exhaust gas recirculation systems

(c) Assembly of exhaust gas recirculation systems (except in cases where it is submitted in accordance with **Chapter 2**)

(d) Construction and arrangement of thermal insulation for exhaust gas pipes fitted onto exhaust gas recirculation systems (except in cases where it is submitted in accordance with **Chapter 2**)

23.2 Design

23.2.1 General Requirements

1 The requirements of **22.2.1** are to be applied. In such cases, "exhaust gas cleaning system" is to be read as "exhaust gas recirculation system".

2 Heat exchangers fitted onto exhaust gas recirculation systems are to comply with **Chapter 10**.

23.2.2 Materials

The requirements of **22.2.2** are to be applied. In such cases, "exhaust gas cleaning system" is to be read as "exhaust gas recirculation system".

23.3 Exhaust Gas Cleaning Systems

23.3.1 Construction of Exhaust Gas Cleaning Systems

1 The requirements of **22.3.1** (excluding **-3** and **-6**) are to be applied.

2 Devices to shut down the exhaust gas supply to exhaust gas pipes in which a scrubber is fitted are to be provided.

23.4 Requirements for Construction and Arrangements, etc.

23.4.1 Construction and Arrangement

In addition to **22.4.1**, the following **(1)** and **(2)** requirements are to be applied:

(1) Consideration is to be given to ensure that recirculating exhaust gas does not have any adverse effect on engine performance and safety due to corrosion and fouling, etc.

(2) Consideration is to be given to ensure taken that temperature of the intake air/scavenging air introduced into cylinders does not exceed the allowable temperatures specified by engine manufacturers.

23.4.2 Ventilation Systems

The requirements of **22.4.2** are to be applied.

23.4.3 Venting Systems of Storage Tanks for Chemical Treatment Fluids

The requirements of **22.4.3** are to be applied.

23.4.4 Safety Devices and Alarm Devices

The requirements of **22.4.4** are to be applied.

23.5 Electrical Installations

23.5.1 General

The requirements of **22.5.1** are to be applied. In such cases, “exhaust gas cleaning system” is to be read as “exhaust gas recirculation system”.

23.6 Safety and Protective Equipment

23.6.1 General

The requirements of **22.6.1** are to be applied.

23.7 Tests

23.7.1 Tests at Facilities (Shop tests)

In addition to **22.7.1**, the following **(1)** and **(6)** requirements are to be applied:

(1) Starting and stopping test of exhaust gas recirculation systems

(2) Test for load response

(3) Emergency stop test

(4) Test at normal load with exhaust gas recirculation systems running

(5) Hydrostatic test (at a pressure equal to 1.5 times the maximum working pressure for the pressure receiving parts of the cooling systems for blowers fitted onto exhaust gas recirculation systems and the cooling sides of heat exchangers fitted onto exhaust gas recirculation systems)

(6) Other tests deemed necessary by the Society

23.7.2 Tests after Installation On Board

The requirements of **22.7.2** are to be applies. In such cases, “exhaust gas cleaning system” is to be read as “exhaust gas recirculation system”.

Chapter 24 SPARE PARTS, TOOLS AND INSTRUMENTS

24.1 General

24.1.1 Scope

1 The requirements in this Chapter apply to spare parts, tools and instruments for machinery installations.

2 The term "machinery installations" used in this Chapter is defined as follows:

(1) Reciprocating internal combustion engines used as main propulsion machinery

(2) Reciprocating internal combustion engines used to drive generators or auxiliary machinery essential for main propulsion

(3) Steam turbines used as main propulsion machinery

(4) Steam turbines used to drive generators or auxiliary machinery essential for main propulsion

(5) Main propulsion shafting

(6) Boilers

(7) Pumps and air compressors

(8) Waterjet propulsion systems

(9) Azimuth thrusters

3 Since the requirements for the various spare parts and tools used for ships depend on a variety of different things: the respective regulations of the countries of their flag administration, their purpose/use, the kinds of machinery installations employed, the navigation routes they follow, etc., the requirements in this Chapter may not be applicable in all cases. However, as a rule, the spare parts and tools specified in this Chapter are to be provided for engine rooms, boiler rooms, or any other appropriate places in a ship.

4 Any spare parts, tools and instruments for machinery installations not specified in this Chapter are to be as deemed appropriate by the Society.

5 Spare parts and tools for electrical installations are to comply with the requirements in **3.8 in Part H**.

6 Spare parts for ventilating fans of ships carrying liquefied gases or dangerous chemicals in bulk are to comply with the requirements in **Chapter 12, Part N** or **Chapter 3, Part S** respectively.

24.1.2 Documentation

The ship's owner or shipbuilder is to submit, for approval, a list showing the number of specified spare parts, tools and instruments for machinery installation, that are actually provided on board.

24.2 Spare Parts, Tools and Instruments

24.2.1 Spare Parts

1 Spare parts for reciprocating internal combustion engines used as main propulsion machinery are given in **Table D24.1**.

2 Spare parts for reciprocating internal combustion engines used to drive generators (except emergency generators) or any auxiliary machinery essential for main propulsion are given in **Table D24.2**.

3 Spare parts for steam turbines used as main propulsion machinery and steam turbines used to drive generators (except emergency generators) or any auxiliary machinery essential for main propulsion are given in **Table D24.3**.

4 Spare parts for main propulsion shafting are given in **Table D24.4**.

5 Spare parts for main boilers, essential auxiliary boilers, boilers to supply steam for fuel oil heating necessary for the operation of main propulsion machinery or continuous cargo heating as well as thermal oil installations for essential use are given in **Table D24.5**. However, no spare parts are required, if stand-by means are provided to ensure that normal service conditions and cargo heating are maintained, even in cases where there is a failure of any boilers (other than the main boiler), or of any thermal oil installations.

6 Spare parts for pumps and air compressors (other than those for emergency use) which are classified as auxiliary machinery essential for main propulsion as well as for bilge pumps are given in **Table D24.6**.

7 The spare parts for waterjet propulsion systems are to be in accordance with the following:

(1) Ball bearings: 1 set for each type and each size

(2) Pumps: spare parts specified in **Table D24.6**

8 Spare parts for azimuth thrusters are to be in accordance with the following:

(1) Hydraulic motors: bearings and sealing devices 1 set for each pump type and each pump size

(2) Pumps: bearings and sealing devices 1 set for each pump type and each pump size

9 Spare parts for the machinery installations listed in the **Tables D24.1** to **D24.6** are those required for each single set of such machinery installations. In cases where two or more sets of machinery installations of the same type and same service have been installed on a ship, only one set of spare parts for such machinery installations may be acceptable.

However, the number of water gauge glasses of round type and flat type is required to be the number in **Table D24.5** for each boiler; and, the number of flat type water gauge frames is required to be one for every two boilers.

10 Notwithstanding the requirement specified in **-9** above, no spare parts are required for any machinery installations specified in the following **(1)** and **(2)**:

(1) Machinery installations whose number exceeds that required under the Rules and which have capacity that is adequate with respect to the normal service conditions of the ship.

(2) Pumps classified as auxiliary machinery essential for main propulsion, which have stand-by pumps that have a capacity that is adequate with respect to the normal service conditions of the ship.

24.2.2 Tools and Instruments

The required tools and instruments for a single ship are given in **Table D24.7**.

Table D24.1 Spare Parts for Reciprocating Internal Combustion Engines Used as Main Propulsion Machinery

Item	Spare parts	Number required
Main bearings	Main bearings or shells for one bearing of each size and type fitted, complete with shims, bolts and nuts	1 set
Cylinder liner	Cylinder liner, complete with joint rings and gaskets	1
Cylinder cover	Cylinder cover, complete with all valves, joint rings and gaskets For engine without cylinder cover, the respective valves	1
	Cylinder cover bolts and nuts, for one cylinder	1/2 set
Cylinder valves	Exhaust valves, complete with casings, seats, springs and other fittings for one cylinder	2 sets
	Air inlet valves, complete with casing, seats, springs and other fittings for one cylinder	1 set
	Starting air valve, complete with casing, seat, springs and other fittings	1
	Relief valve, complete with casting, springs and other fittings	1
	Fuel valves, complete with castings, springs and other fittings for one engine *Note: Engines with three or more fuel valves per cylinder: two fuel valves complete per cylinder, and other fuel valves excluding casings.	1 set*
Connecting rod bearings	Bottom end bearings or shells of each size and type fitted, complete with shims, bolts and nuts	1 set
	Top end bearings or shells of each size and type fitted, complete with shims, bolts and nuts	1 set
Pistons	Crosshead type: Piston of each type fitted, complete with piston rod, stuffing box, skirt, rings, studs and nuts	1
	Trunk piston type: Piston of each type fitted, complete with skirt, rings, studs, nuts, gudgeon pin and connecting rod	1
Piston rings	Piston rings for one cylinder	1 set
Pistons cooling devices	Telescopic cooling pipes and fittings or equivalent for one cylinder unit	1 set
Chain for camshaft drives	Chain drive: Separate links with pins and rollers of each size and type fitted	6

Cylinder lubricator	Lubricator, complete, of the largest size, with its driving chain or gear wheel	1
Fuel injection pumps	Fuel injection pump complete, or, when replacement at sea is practicable, a complete set of working parts for one pump (plunger, sleeve, valves, springs, etc.)	1
Fuel injection piping	High pressure fuel pipe of each size shape fitted, complete with couplings	1
Scavenge blowers (including turbo chargers)	Rotors, rotor shafts, bearings, nozzle rings and gear wheels or equivalent working parts if other type (see Note)	1 set
Scavenging system	Suction and delivery valves for one pump of each type fitted, complete	1 set
Reduction and or reversing gear	Complete bearing bush of each size fitted in the gear case assembly	1 set
	Roller or ball bearing, complete, of each size fitted in the gear case assembly	1 set
Gaskets and packings	Special gaskets and packings of each size and type fitted for cylinder cover and cylinder liner for one cylinder	-
Parts for electronically-controlled engines	Control valves	1 of each type
	Accumulator diaphragms	2 of each type
	Sensors provided for each cylinder *Note: Spare parts may be omitted in cases where normal operation of main propulsion machinery is available without these sensors.	1 of each type*

Note:

The spare parts for scavenge blowers (including turbo chargers) may be omitted where it has been demonstrated, at the builder's test bench, for one engine of the type concerned, that the engine can be manoeuvred satisfactorily with one blower out of action. However, in this case the requisite blanking and blocking arrangements for running with one blower out of action are to be available on board.

Table D24.2 Spare Parts for Reciprocating Internal Combustion Engines Driving Generators or Auxiliary Machinery Essential for Main Propulsion

Item	Spare parts	Number required
Main bearings	Main bearings of shells of each size and type fitted, complete with shims, bolts and nuts	1 set
Cylinder valves	Exhaust valves, complete with casings, seats, springs and other fittings for one cylinder	2 sets
	Air inlet valves, complete with casings, seats, springs and other fittings for one cylinder	1 set
	Starting air valve, complete with casting, seat, springs and other fittings	1
	Relief valve, complete with casing, springs and other fittings	1
	Fuel valves of each size and type fitted, complete with casings, springs and other fittings for one engine	1/2 set
Connecting rod bearings	Bottom end bearings or shells of each size and type fitted, complete with shims, bolts and nuts for one cylinder	1 set
	Top end bearings or shells of each size and type fitted, complete with shims, bolts and nuts for one cylinder	1 set
	Trunk piston type: Gudgeon pin with bush for one cylinder	1 set
Piston rings	Piston rings, for one cylinder	1 set
Piston cooling devices	Telescopic cooling pipes and fittings or their equivalent for one cylinder	1 set
Fuel injection pumps	Fuel injection pump (complete). Or, when replacement at sea is practicable, a complete set of working parts for one pump (plunger, sleeve, valves, springs, etc.)	1
Fuel injection piping	High pressure fuel pipe of each size and shape fitted, complete with couplings	1
Gaskets and packings	Special gaskets and packings of each size and type fitted, for cylinder cover and cylinder liner for one cylinder	1 set

Table D24.3 Spare Parts for Steam Turbines

Item	Spare parts	Number required
Main bearings	Bearings of each size for rotor shaft and reduction gear shaft	1 set for each shaft
Rotor thrust bearings	Pads (including adjusting lines and rings) for one face	1 set*
Turbine shaft sealing rings	Carbon sealing rings, with springs, for each size and type	1 set
Oil filters	Strainer baskets or inserts for filters of each size and type Applicable to those of special design	1 set

*Note:

For steam turbines used as main propulsion machinery, in cases where the pads of one face differ from those of the other, a complete set of pads is to be provided.

Table D24.4 Spare Parts for Main Shafting

Spare parts	Number required
Main thrust bearing :	
Pads for one face of Michell type thrust blocks	1 set for each size*
Complete thrust shoe for one face of solid ring types	1 for each size*
Inner and outer race with rollers of roller thrust bearings	1 for each size

*Note:

In cases where the pads of one face differ from those of the other, a complete set of pads is to be provided.

Table D24.5 Spare Parts for Boilers and Thermal Oil Installations

Spare parts	Number required
Safety valve spring of each size including superheater safety valve springs	1
Oil burner nozzles, complete for one boiler	1 set
Round type water gauge glasses including packings	12
Flat type water gauge glasses	2
Flat type water gauge frames	1

Table D24.6 Spare Parts for Pumps and Air Compressors

Item	Spare parts	Number required
Piston pumps	Valves with seats and springs of each size fitted	1 set
	Piston rings of each type and size for one piston	1 set
Centrifugal and gear type pumps	Bearings of each type and size	1
	Rotor sealings of each type and size (Parts liable to deteriorate, such as packings, sleeves)	1
Air Compressors	Piston rings of each type and size	1 set
	Suction and delivery valves and their springs of each size	1/2 set

Notes:

1. Pumps and air compressors including those for remote or automatic control systems.
2. Gear type pumps including vane pumps and screw pumps.

Table D24.7 Tools and Instruments

Description	Spare parts	Number required
Boiler required spare parts in the requirement in 24.2.1-5	Tube stoppers or plugs of each size, including those for superheater tubes and economizer tubes *Note: In the case of cylindrical boilers, half of them are to be those which can be used from the burner side.	For water tube boilers: 12 for each size
		For other type of boilers: 12 in total*
All boilers	Standard pressure gauges *Note: Gauge testers will be acceptable.	1*
	Water testers *Note: Two salinometers will be acceptable.	1 set*
Special tools and instruments for machinery installation maintenance or repair work		1 set

Chapter 25 SPECIAL REQUIREMENTS FOR MACHINERY INSTALLED IN SHIPS WITH RESTRICTED AREA OF SERVICE AND SMALL SHIPS

25.1 General

25.1.1 Scope

The requirements in this Chapter apply to machinery to be installed in ships with a gross tonnage less than 500 *tons* and intended for registry with restricted areas of service in place of any relevant requirements found in the Chapters up to and including **Chapter 24**.

25.2 Modified Requirements

25.2.1 Ships with Class Notation "*Coasting Service*" or Equivalent

1 The provisions of spare units for any of the machinery or devices in the following **(1)** to **(7)** need not to be installed on board provided that two sets of such machinery of nearby the same capacity are installed on board with a total capacity sufficient enough to obtain the maximum continuous output of main propulsion machinery or the maximum evaporative capacity of main and essential auxiliary boilers, and the capacity of one unit of either machinery set is sufficient enough for ships to obtain navigable speed.

(1) The pressure source for driving the clutches of power transmission systems used for main propulsion specified in **5.2.4-3**.

(2) Hydraulic pumps used for the pitch control gears of controllable pitch propellers specified in **7.2.2-8**.

(3) Fuel oil supply pumps specified in **13.9.6-1** and **-2**.

(4) Burning systems for boilers specified in **13.9.7-1** and **-2**.

(5) Lubricating oil pumps specified in **13.10.2-1** and **-2**.

(6) Cooling water (oil) pumps for main propulsion machinery specified in **13.12.1-1** and **-2**.

(7) Feed water systems specified in **13.15.1-1** and **-2**.

2 In the requirements specified in **13.9.6-1(2)**, **13.10.2-1(2)**, and **13.12.1-1(3)** the requirements to provide a complete set of the spare pump do not apply.

3 The requirement specified in **15.3.1-6** and **20.9.1-3** need not apply.

4 For ships with the Class Notation "*Coasting Service*" or equivalent, which are not engaged in international voyages, or whose gross tonnage is less than 500 *tons*, the following requirements may be applied in addition to **-1** to **-3** above.

(1) The requirements specified in **1.3.1-5** need not apply.

(2) The requirements specified in **1.3.8** need not apply. (however, only for those ships not engaged in international voyages)

(3) The requirements specified in **1.3.9** need not apply.

(4) Appropriate devices specified in **5.2.4-3** may be replaced with emergency fixing bolts for clutches to enable the ship to obtain navigable speed.

(5) Appropriate devices specified in **7.2.2-8**, may be replaced with propeller pitch-fixing devices to enable the ship to obtain a navigable speed.

(6) The requirements specified in **13.5.10**, **13.6.1-5**, **13.8.5**, **13.8.7**, **13.9.1-5** and **13.9.1-6** need not apply.

(7) The requirements specified in **15.1.5** need not apply.

(8) The requirements specified in **15.2.4-5** and **-6** need not apply (excluding those cases where any provision of auxiliary steering gear has been omitted according to the requirements in **15.2.1-2**).

(9) The requirements for alternative source of powers specified in **15.2.6** need not apply.

(10) The requirements in **15.2.7-1** and **-7** need not apply.

(11) The requirements for overload for circuits and motors specified in **15.2.7-5** need not apply.

(12) A means of communication between the navigating bridge and the steering gear compartment specified in **15.2.9** may be replaced with an appropriate alternative means.

(13) The requirements in **15.3.1-5** need not apply.

(14) The requirements specified in **19.1.5**, **19.5.4-4**, **19.6.2**, **19.6.3(2)**, **19.6.3(5)** (only those requirements concerned with overload alarms of motors), **19.6.3(7)**, and **19.7.1-7** need not apply.

(15) Notwithstanding the requirements of **19.6.3(1)**, each steering system may be served separately by exclusive circuits fed directly from main switchboards. In cases where three or more propulsion systems are provided, these exclusive circuits may be composed of at least two systems. In addition, one of these circuits may be supplied through the emergency switchboard.

(16) The requirements specified in **20.2.5**, **20.5.3(4)**, **20.6.1-2**, **20.6.2**, **20.6.3(2)** (only those requirements concerned with overload alarms of motors), **20.6.3(4)**, and **20.7.1-5** need not apply.

(17) Notwithstanding the requirements of **20.6.1-1**, each steering system may be served separately by exclusive circuits fed directly from main switchboards. In cases where three or more propulsion systems are provided, these exclusive circuits may be composed of at least two systems. In addition, one of these circuits may be supplied through the emergency switchboard.

25.2.2 Ships with the Class Notation "*Smooth Water Service*" or Equivalent

1 In addition to the requirements specified in **25.2.1-1**, **-2** and **-3** above, the buffer arrangements specified in **15.4.9** may be omitted.

2 For ships with an upper stock diameter of not more than 120 *mm* as calculated by the formula in **Chapter 13, Part 1, Part C** (however, in cases where K_S is less than 1, calculations are to be made with a material factor K_S = 1.), the provisions of the auxiliary steering gear specified in **15.2.1** above may be omitted in cases where spare parts for consumables, such as packing and bearings, are provided for power-driven main steering gear or in cases where spare steering wires are provided for manually-powered main steering gear.

3 For ships with the Class Notation "*Smooth Water Service*" or equivalent which are not engaged in international voyages, or whose gross tonnage is less than 500 *tons*, the following requirements may apply in addition to the requirements specified in **25.2.1-1** to **-4**, **25.2.2-1** and **-2**.

(1) Notwithstanding the requirements in **1.3.1-4**, the provision of one unit or one set each of the machinery specified in **25.2.1-1(1)** to **(7)** may be accepted, provided that each has such a capacity sufficient for the main propulsion machinery to obtain the maximum continuous output and for the main and essential auxiliary boiler to obtain the maximum evaporative capacity.

(2) The requirements for fuel oil transfer pumps specified in **13.9.3** may be modified to one set of pumps driven independent sources of power.

(3) Notwithstanding the requirement in **1.3.1-3**, the requirements for two or more starting air compressors specified in **13.13.3** may be modified to one starting air compressor driven by an independent source of power.

25.2.3 Ships with a Gross Tonnage less than 500 *Tons*, etc.

1 For ships with a gross tonnage less than 500 *tons*, the requirements specified in **25.2.1-3** and **-4(1)**, **(3)** and **(6)** through **(13)** above may be complied with. Moreover, the buffer arrangements specified in **15.4.9** may be omitted.

2 For ships which are not engaged on international voyages or whose gross tonnage is less than 500 *tons*, the requirements specified in **13.4.1-4** and **13.8.6** need not apply.

3 For ships which are not engaged on international voyages and whose gross tonnage is not less than 500 *tons*, where deemed appropriate by the Society taking account of various conditions of such ships related to the navigation, **13.8.5** and **13.8.7** need not apply.

25.3 Spare Parts, Tools and Instruments for Ships with Restricted Areas of Service

25.3.1 Spare Parts, Tools and Instruments and etc. for Ships with Class Notation "*Coasting Service*" or Equivalent

Spare parts for machinery installed in ships with a Class Notation of "*Coasting Service*" or equivalent may comply with the requirements specified in **Table D25.1**. Furthermore, for ships equipped with 2 or more reciprocating internal combustion engines or steam turbines for main propulsion and for ships equipped with 2 or more main generators, spare parts for reciprocating internal combustion engines or steam turbines for main propulsion or to drive main generators are not required.

25.3.2 Spare Parts for Ships with Class Notation "*Smooth Water Service*" or Equivalent

Spare parts for the machinery installed in ships with a Class Notation of "*Smooth Water Service*" may comply with the requirements specified in **Table D25.2**. Furthermore, for ships equipped with 2 or more reciprocating internal combustion engines or steam turbines for main propulsion and for ships equipped with 2 or more main generators, spare parts for reciprocating internal combustion or steam turbines for main propulsion or to drive main generators are not required.

Table D25.1 Spare Parts for Coasting Service Ships

<table>
<tr><th>Area of service</th><th colspan="2">Table No. and Paragraph No. in Chapter 24</th><th colspan="2">Items and types of spares</th><th>Quantity</th></tr>
<tr><td rowspan="14">Coasting Service</td><td rowspan="5">Table D24.1</td><td></td><td colspan="2">Cylinder liner, cylinder cover, piston, camshaft driving gear, cylinder lubricator, scavenging air blower (including turbocharger) scavenging air system, reduction gear, reversing gear</td><td rowspan="3">Omitted</td></tr>
<tr><td rowspan="4">Table D24.2</td><td colspan="2">Main bearing, piston cooling system</td></tr>
<tr><td rowspan="2">Cylinder-mounted valve</td><td>Starting air valve, relief valve</td></tr>
<tr><td>Exhaust gas valve, fuel injector</td><td>For one cylinder</td></tr>
<tr><td colspan="2">Connecting rod bearing</td><td>Lower half of small end bearing metal, upper half of big end bearing metal, one piece each</td></tr>
<tr><td colspan="2">Table D24.3 and Table D24.4</td><td colspan="2">All items and all types</td><td>Omitted</td></tr>
<tr><td colspan="2" rowspan="2">Table D24.5</td><td colspan="2">Cylindrical water gauge glass</td><td>6 pieces</td></tr>
<tr><td colspan="2">Flat water gauge glass</td><td>One piece</td></tr>
<tr><td colspan="2">Table D24.6</td><td colspan="2">Centrifugal pump, gear pump, air compressor</td><td rowspan="4">Omitted</td></tr>
<tr><td colspan="2">24.2.1-7</td><td colspan="2">Ball bearings, Pumps</td></tr>
<tr><td colspan="2">24.2.1-8</td><td colspan="2">Hydraulic motors, Pumps</td></tr>
<tr><td colspan="2" rowspan="3">Table D24.7</td><td colspan="2">Standard pressure gauge</td></tr>
<tr><td rowspan="2">Tube plug</td><td>Water tube boiler</td><td>4 pieces each</td></tr>
<tr><td>Other types of boiler</td><td>4 pieces in total</td></tr>
</table>

Table D25.2 Spare Parts for Smooth Water Service Ships

<table>
<tr><th>Area of service</th><th>Table No. and Paragraph No. in Chapter 24</th><th colspan="2">Items and types of spares</th><th>Quantity</th></tr>
<tr><td rowspan="12">Smooth Water Service</td><td rowspan="2">Table D24.1 and Table D24.2</td><td colspan="2">Connecting rod bearing</td><td>Lower half of small end bearing metal, upper half of big end bearing metal, one piece each</td></tr>
<tr><td colspan="2">All items excluding connecting rod bearing</td><td rowspan="3">Omitted</td></tr>
<tr><td>Table D24.3 and Table D24.4</td><td colspan="2">All items and all types</td></tr>
<tr><td rowspan="3">Table D24.5</td><td colspan="2">Safety valve spring, complete set of oil burner</td></tr>
<tr><td colspan="2">Cylindrical water gauge glass</td><td>3 pieces</td></tr>
<tr><td colspan="2">Flat water gauge glass</td><td>One piece</td></tr>
<tr><td>Table D24.6</td><td colspan="2">Centrifugal pump, gear pump, air compressor</td><td rowspan="4">Omitted</td></tr>
<tr><td>24.2.1-7</td><td colspan="2">Ball bearings, Pumps</td></tr>
<tr><td>24.2.1-8</td><td colspan="2">Hydraulic motors, Pumps</td></tr>
<tr><td rowspan="3">Table D24.7</td><td colspan="2">Standard pressure gauge</td></tr>
<tr><td rowspan="2">Tube plug</td><td>Water tube boiler</td><td>2 pieces for each type</td></tr>
<tr><td>Other types of boiler</td><td>2 pieces in total</td></tr>
</table>

Annex 2.3.1 CALCULATION METHOD OF CRANKSHAFT STRESS

1.1 Scope

This annex applies to solid-forged and semi-built crankshafts of reciprocating internal combustion engines made of forged or cast steel, with one crank throw between main bearings.

1.2 Principles of Calculation

1 The principles of calculation in this Guidance are as follows:

(1) The design of crankshafts is based on an evaluation of safety against fatigue in highly stressed areas.

(2) These calculations are also based on the assumption that areas exposed to highest stresses are those that are listed below. In addition, attention is to be paid to prevent any excessive stress concentrations in outlets of journal oil bores.

(a) Fillet transitions between crankpins and webs

(b) Fillet transitions between journals and webs

(c) Outlets of crankpin oil bores

(3) Calculations of crankshaft strength require that nominal alternating bending (*See* **1.3.1**) and nominal alternating torsional stresses (*See* **1.3.2**) are determined first. Then, these values are multiplied by appropriate stress concentration factors (*See* **1.4**) which results in equivalent alternating stresses (uni-axial stresses) (*See* **1.6**).

(4) Equivalent alternating stresses are evaluated in accordance with the following:

(a) In fillets, bending and torsion lead to two different biaxial stress fields which can be represented by a Von Mises equivalent stress under additional assumptions that bending and torsion stresses are time phased and that corresponding peak values occur at the same locations.

(b) At oil hole outlets, bending and torsion lead to two different stress fields which can be represented by equivalent principal stresses equal to the maximum of principal stresses resulting from combinations of these two stress fields under the assumption that bending and torsion are time phased.

(5) Equivalent alternating stresses are then compared with the fatigue strengths of selected crankshaft materials (*See* **1.7**). These comparisons are to show whether or not those crankshafts concerned are dimensioned adequately (*See* **1.8**).

2 In cases where journal diameter is equal to or larger than crankpin diameter, the outlets of main journal oil bores are to be formed in a similar way to the outlets of crankpin oil bores; otherwise, separate documentation for fatigue safety may be required.

1.3 Calculation of Stresses

1.3.1 Alternating Bending Stress

1 Assumptions

Calculations of alternating bending stresses are based on the following assumptions:

(1) Calculations are based on statically determined systems, composed of a single crank throw supported in the centre of adjacent main journals and subject to gas and inertia forces.

(2) Bending lengths are taken as the length between the two main bearing midpoints (distance L_3, *See* **Fig. 1** and **Fig. 2**).

(3) The bending moments M_{BR} and M_{BT} are calculated based on triangular bending moment diagrams due to the radial component F_R and tangential component F_T of the connecting rod force, respectively (*See* **Fig. 1**).

(4) For those crank throws with two connecting rods acting upon one crankpin, the relevant bending moments are obtained by superposition of the two triangular bending moment diagrams in accordance with phase (*See* **Fig. 2**).

(5) Bending moments and radial forces acting in webs

(a) The bending moment M_{BRF} and the radial force Q_{RF} are taken as acting in the centre of solid webs (distance L_1) and are derived from the radial components of connecting rod forces.

(b) Alternating bending and compressive stresses due to bending moments and radial forces are to be related to cross-sections of crank webs. These reference sections result from the web thickness W and the web width B (*See* **Fig. 3**).

(c) Mean stresses are neglected.

(6) Bending moments acting in outlets of crankpin oil bores

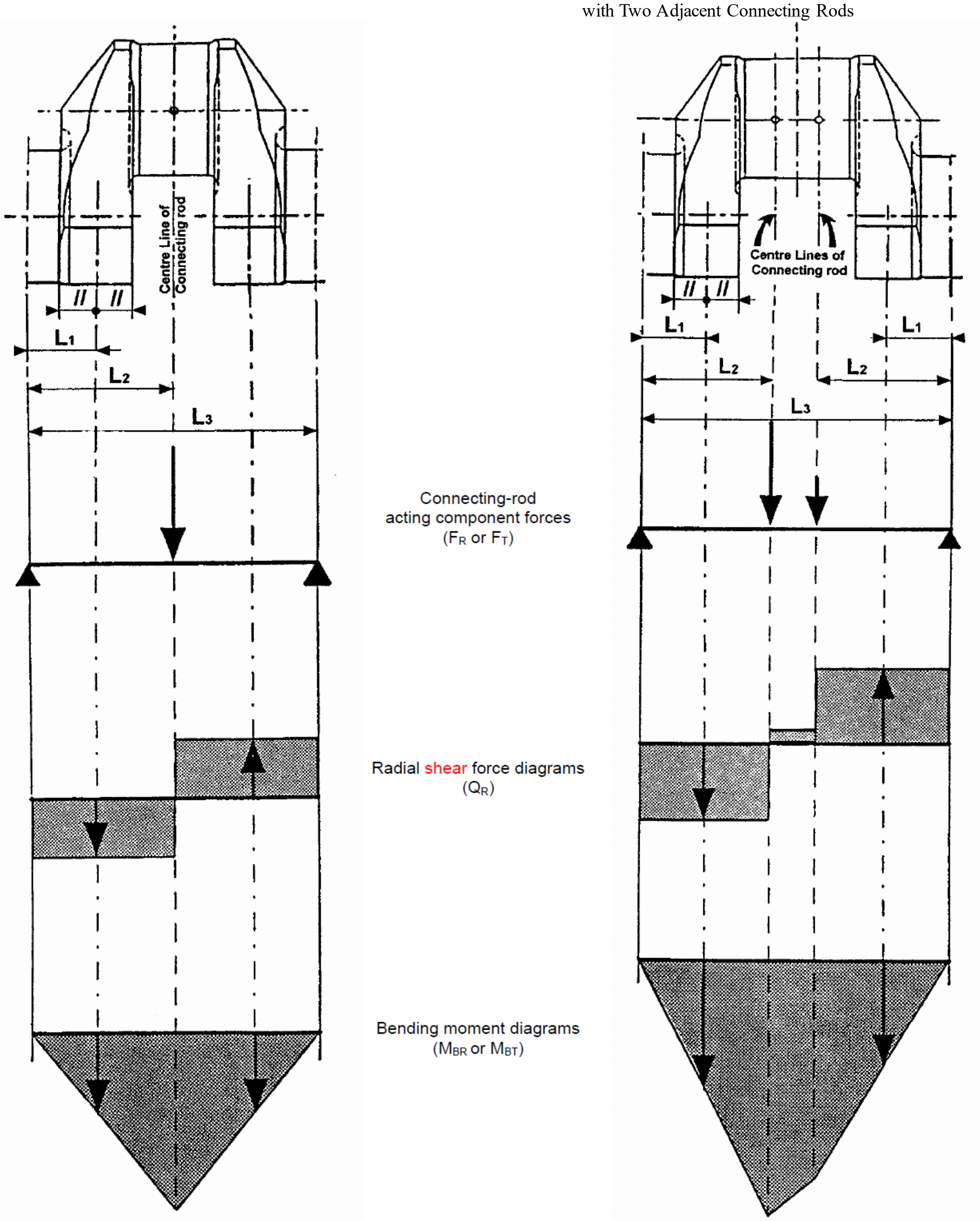
(a) Two relevant bending moments are taken in crankpin cross-sections through oil bores and are derived from the radial and tangential components of connecting rod forces (*See* **Fig. 4**).

(b) Any alternating stresses due to these bending moments are to be related to the cross-sections of axially bored crankpins.

(c) Mean bending stresses are neglected.

Fig. 1 Crank Throw for In-line Engines

Fig. 2 Crank Throw for Vee type Engines with Two Adjacent Connecting Rods

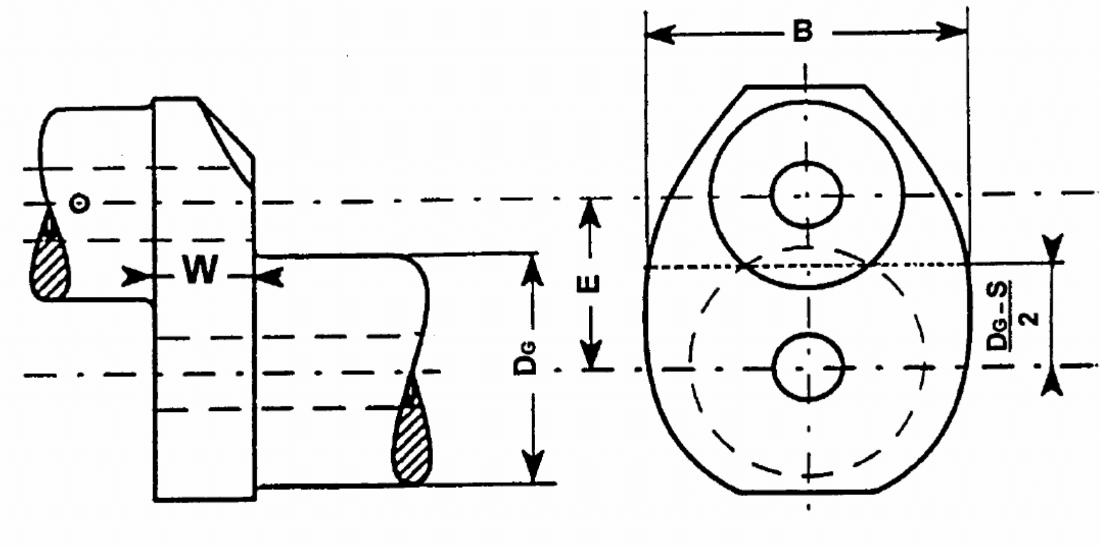


L_1 = Distance between main journal centre line and crank web centre (*See* also **Fig. 3** for crankshafts without overlaps)

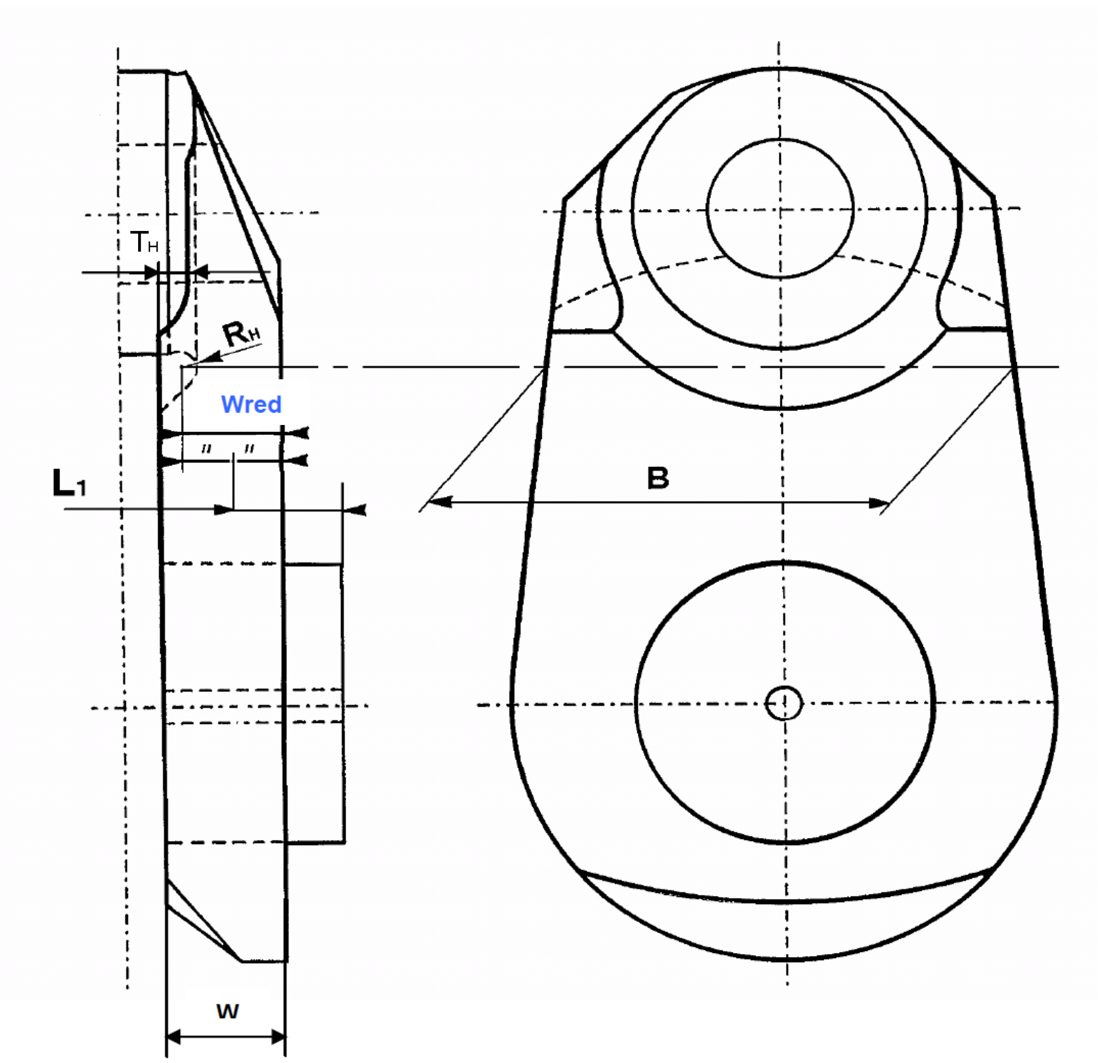
L_2 = Distance between main journal centre line and connecting rod centre

L_3 = Distance between two adjacent main journal centre lines

Fig. 3 Reference Areas of Crank Web Cross Sections

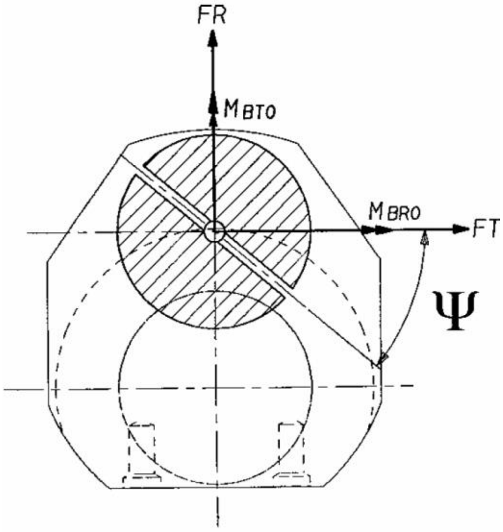


Overlapped crankshaft



Crankshaft without overlap

Fig. 4 Crankpin Sections through Oil Bores



M_{BRO} is the bending moment of the radial component of the connecting rod force.

M_{BTO} is the bending moment of the tangential component of the connecting rod force.

2 Nominal Alternating Bending and Compressive Stresses

(1) Calculation procedures are as follows:

(a) Radial and tangential forces, due to gas and inertia loads, acting upon crankpins at connecting-rod positions will be calculated over one working cycle.

(b) Using the forces calculated over one working cycle and taking into account of the distance from the main bearing midpoint, the time curve of the bending moments M_{BRF}, M_{BRO}, M_{BTO} and radial forces Q_{RF}, as defined in **-1(5)** and **(6)**, will then be calculated.

(c) In case of Vee type engines, bending moments, progressively calculated from gas and inertia forces, of the two cylinders, acting on one crank throw, are superposed in accordance with phase. Different designs (forked connecting rods, articulated-type connecting rods or adjacent connecting rods) are to be taken into account.

(d) In cases where there are cranks of different geometrical configurations in one crankshaft, calculations are to cover all crank variants.

(2) Nominal alternating bending and compressive stresses in web cross-sections

(a) Calculation of nominal alternating bending stresses is as follows:

$$\sigma_{BFN} = \pm \frac{M_{BRFN}}{W_{eqw}} \cdot 10^3 \cdot Ke$$

$$M_{BRFN} = \pm \frac{1}{2}(M_{BRF\max} - M_{BRF\min})$$

$$W_{eqw} = \frac{B \cdot W^2}{6}$$

where

σ_{BFN} : Nominal alternating bending stress related to the web (N/mm^2)

W_{eqw} : Section modulus related to cross-section of web (mm^3)

Ke : Empirical factor considering to some extent the influence of adjacent cranks and bearing restraint with:

Ke = 0.8 for 2-stroke engines

Ke = 1.0 for 4-stroke engines

M_{BRFN} : Alternating bending moment related to the centre of the web ($N \cdot m$) (*See* **Fig. 1** and **Fig. 2**)

$M_{BRF\max}$: Maximum bending moment related to the centre of the web within one working cycle ($N \cdot m$)

$M_{BRF\min}$: Minimum bending moment related to the centre of the web within one working cycle ($N \cdot m$)

(b) Calculation of nominal alternating compressive stresses is as follows:

$$\sigma_{QFN} = \pm \frac{Q_{RFN}}{F} \cdot Ke$$

$$Q_{RFN} = \pm\frac{1}{2}(Q_{RF\max} - Q_{RF\min})$$

$$F = BW$$

where

σ_{QFN} : Nominal alternating compressive stress due to radial force related to the web (N/mm^2)

Q_{RFN} : Alternating radial force related to the web (N) (*See* **Fig. 1** and **Fig. 2**)

$Q_{RF\max}$: Maximum radial force related to the web within one working cycle (N)

$Q_{RF\min}$: Minimum radial force related to the web within one working cycle (N)

F : Area related to cross-section of web (mm^2)

(3) Nominal alternating bending stress in outlets of crankpin oil bores

Calculation of nominal alternating bending stresses is as follows:

$$\sigma_{BON} = \pm\frac{M_{BON}}{We}\cdot 10^3$$

$$M_{BON} = \pm\frac{1}{2}(M_{BO\max} - M_{BO\min})$$

$$We = \frac{\pi}{32}\left(\frac{D^4 - D_{BH}^4}{D}\right)$$

where

σ_{BON} : Nominal alternating bending stress related to the crankpin diameter (N/mm^2)

M_{BON} : Alternating bending moment calculated at the outlet of crankpin oil bore ($N\cdot m$)

$M_{BO\max}$: Maximum value of bending moment M_{BO} within one working cycle ($N\cdot m$)

$M_{BO\min}$: Minimum value of bending moment M_{BO} within one working cycle ($N\cdot m$)

M_{BO} : Bending moment acting in outlet of crankpin oil bore ($N\cdot m$)

$$M_{BO} = (M_{BTO}\cdot\cos\psi + M_{BRO}\sin\psi)$$

ψ: Angular position (*See* **Fig. 4**)

We : Section modulus related to cross-section of axially bored crankpin (mm^3)

D, D_{BH} : *see* **1.4.1**

3 Alternating Bending Stresses in Fillets and Outlets of Crankpin Oil Bores

(1) Calculation of alternating bending stresses in crankpin fillets is as follows:

$$\sigma_{BH} = \pm(\alpha_B\cdot\sigma_{BFN})$$

where

σ_{BH} : Alternating bending stress in crankpin fillet (N/mm^2)

α_B : Stress concentration factor for bending in crankpin fillet (*See* **1.4.2** and **3.1.2-2** of **Appendix 1**)

(2) Calculation of alternating bending stresses in journal fillets (not applicable to semi-built crankshafts) is as the following formulae in **(a)** or **(b)**:

(a) $\sigma_{BG} = \pm(\beta_B\cdot\sigma_{BFN} + \beta_Q\cdot\sigma_{QFN})$

where

σ_{BG} : Alternating bending stress in journal fillet (N/mm^2)

β_B : Stress concentration factor for bending in journal fillet (*See* **1.4.3** and **3.1.2-2** of **Appendix 1**)

β_Q : Stress concentration factor for compression due to radial force in journal fillet (*See* **1.4.3** and **3.1.3-2(1)** of **Appendix 1**)

(b) $\sigma_{BG} = \pm(\beta_{BQ}\cdot\sigma_{BFN})$

β_{BQ} : Stress concentration factor for bending and compression due to radial force in journal fillet (*See* **3.1.3-2(2)** of **Appendix 1**)

(3) The calculation of the alternating bending stress in the outlet of crankpin oil bore (only applicable to radially drilled oil hole) is as follows:

$$\sigma_{BO} = \pm(\gamma_B\cdot\sigma_{BON})$$

where

σ_{BO} : Alternating bending stress in outlet of crankpin oil bore (N/mm^2)

γ_B : Stress concentration factor for bending in crankpin oil bore (*See* **1.4.4** and **3.1.2-2** of **Appendix 4**)

1.3.2 Alternating Torsional Stresses

1 Nominal Alternating Torsional Stresses

Calculations for nominal alternating torsional stresses are to be carried out in accordance with the following in order to specify maximum nominal alternating torsional stresses. In addition, maximum nominal alternating torsional stress is to be specified, and the values obtained from such calculations are to be submitted to the Society.

(1) The maximum and minimum torques are to be ascertained for all of the mass points of complete dynamic systems and for entire speed ranges by means of harmonic synthesis of the forced vibrations from the 1st order up to and including the 15th order for 2-stroke cycle engines and from the 0.5th order up to and including the 12th order for 4-stroke cycle engines.

(2) Whilst doing so, allowances must be made for any damping that exists in such systems and for any unfavourable conditions (misfiring, which is defined as the cylinder condition when only compression cycle without any combustion occurs in one of the cylinders).

(3) Speed step calculations are to be selected in such ways that any resonance found in operational speed ranges of engines is detected.

Nominal alternating torsional stresses in mass points calculated results from the following equations:

$$\tau_N = \pm \frac{M_{TN}}{W_P} \cdot 10^3$$

$$M_{TN} = \pm \frac{1}{2}(M_{T\max} - M_{T\min})$$

$$W_P = \frac{\pi}{16}\left(\frac{D^4 - D_{BH}^4}{D}\right) \text{ or } W_P = \frac{\pi}{16}\left(\frac{D_G^4 - D_{BG}^4}{D_G}\right)$$

where

τ_N : Nominal alternating torsional stress related to crankpin or journal (*N/mm*^{2})

W_P : Polar section modulus related to the cross-section of an axially bored crankpin or a bored journal (*mm*^{3})

M_{TN} : Maximum alternating torque ($N \cdot m$)

$M_{T\max}$: Maximum torque ($N \cdot m$)

$M_{T\min}$: Minimum torque ($N \cdot m$)

D, D_{BH}, D_{BG}, D_G: *see* **1.4.1**

In cases where barred speed ranges are necessary, they are to be so arranged that satisfactory operation is possible despite their existence in accordance with **8.2.5** and **8.3.1, Part D of the Rules**. In addition, there are to be no barred speed ranges above a speed ratio of $\lambda \geq 0.8$ for the normal firing condition.

For crankshaft assessments, the nominal alternating torsional stress considered in **-2** below is the highest calculated value, in accordance with the above method, occurring at the most torsionally loaded mass point of the crankshaft system. Where barred speed ranges exist, the torsional stresses within such ranges are not to be considered in assessment calculations. Crankshaft approval is to instead be based on the installation having the largest nominal alternating torsional stress (but not exceeding the maximum figure specified by engine manufacturer). Thus, for each installation, it is to be ensured through suitable calculation that the approved nominal alternating torsional stress is not exceeded. Such calculations are to be submitted to the Society for assessment.

2 Alternating Torsional Stresses in Fillets and Outlets of Crankpin Oil Bores

(1) Calculation of alternating torsional stresses in crankpin fillets is as follows:

$$\tau_H = \pm(\alpha_T \cdot \tau_N)$$

where

τ_H : Alternating torsional stress in crankpin fillet (*N/mm*^{2})

α_T : Stress concentration factor for torsion in crankpin fillet (*See* **1.4.2** and **3.1.1-3** of **Appendix 1**)

τ_N : Nominal alternating torsional stress related to crankpin diameter (*N/mm*^{2})

(2) Calculation of alternating torsional stresses in journal fillets (not applicable to semi-built crankshafts) is as follows:

$$\tau_G = \pm(\beta_T \cdot \tau_N)$$

where

τ_G : Alternating torsional stress in journal fillet (*N/mm*^{2})

β_T : Stress concentration factor for torsion in journal fillet (*See* **1.4.3** and **3.1.1-3** of **Appendix 1**)

τ_N : Nominal alternating torsional stress related to journal diameter (N/mm^2)

(3) Calculation of alternating stresses in outlets of crankpin oil bores due to torsion (only applicable to radially drilled oil holes) is as follows:

$$\sigma_{TO} = \pm(\gamma_T \cdot \tau_N)$$

where

σ_{TO} : Alternating stress in outlet of crankpin oil bore due to torsion (N/mm^2)

γ_T : Stress concentration factor for torsion in outlet of crankpin oil bore (*See* **1.4.4** and **3.1.1-2** of **Appendix 4**)

τ_N : Nominal alternating torsional stress related to crankpin diameter (N/mm^2)

1.4 Stress Concentration Factors

1.4.1 Explanation of Terms and Symbols

1 The terms used in this **1.4** are defined as follows:

(1) The stress concentration factor for bending (α_B, β_B) is defined as the ratio of the maximum equivalent stress (Von Mises), occurring in fillets under bending loads, to the nominal bending stress related to web cross-sections.

(2) The stress concentration factor for compression (β_Q) in journal fillets is defined as the ratio of the maximum equivalent stress (Von Mises), occurring in fillets due to radial forces, to the nominal compressive stress related to web cross-sections.

(3) The stress concentration factor for torsion (α_T, β_T) is defined as the ratio of the maximum equivalent shear stress, occurring in fillets under torsional loads, to the nominal torsional stress related to axially bored crankpins or journal cross-sections.

(4) The stress concentration factors for bending (γ_B) and torsion (γ_T) are defined as the ratio of the maximum principal stress, occurring at outlets of crankpin oil bores under bending and torsional loads, to the corresponding nominal stress related to axially bored crankpin cross-sections.

2 The symbols used in this **1.4** mean as follows (*See* **Fig. 5**):

D : Crankpin diameter (mm)

D_{BH} : Diameter of axial bore in crankpin (mm)

D_O : Diameter of oil bore in crankpin (mm)

R_H : Fillet radius of crankpin (mm)

T_H : Recess of crankpin fillet (mm)

D_G : Journal diameter (mm)

D_{BG} : Diameter of axial bore in journal (mm)

R_G : Fillet radius of journal (mm)

T_G : Recess of journal fillet (mm)

E : Pin eccentricity (mm)

S : Pin overlap (mm)

$$S = \frac{D + D_G}{2} - E$$

W : Web thickness (mm)

In the case of 2-stroke semi-built crankshafts with $T_H > R_H$, the web thickness is to be considered as equal to: $W_{red} = W - (T_H - R_H)$ (*See* **Fig. 3**)

B : Web width (mm)

In the case of 2-stroke semi-built crankshafts, the web width is to be taken in way of crankpin fillet radius centre in accordance with **Fig. 3**.

$r = R_H/D$ (in crankpin fillets), R_G/D (in journal fillets) $(0.03 \le r \le 0.13)$

$s = S/D$ $(s \le 0.5)$

$w = W/D$ $(0.2 \le w \le 0.8)$

$b = B/D$ $(1.1 \le b \le 2.2)$

$d_O = D_O/D$ $(0 \le d_O \le 0.2)$

$d_G = D_{BG}/D$ $(0 \le d_G \le 0.8)$

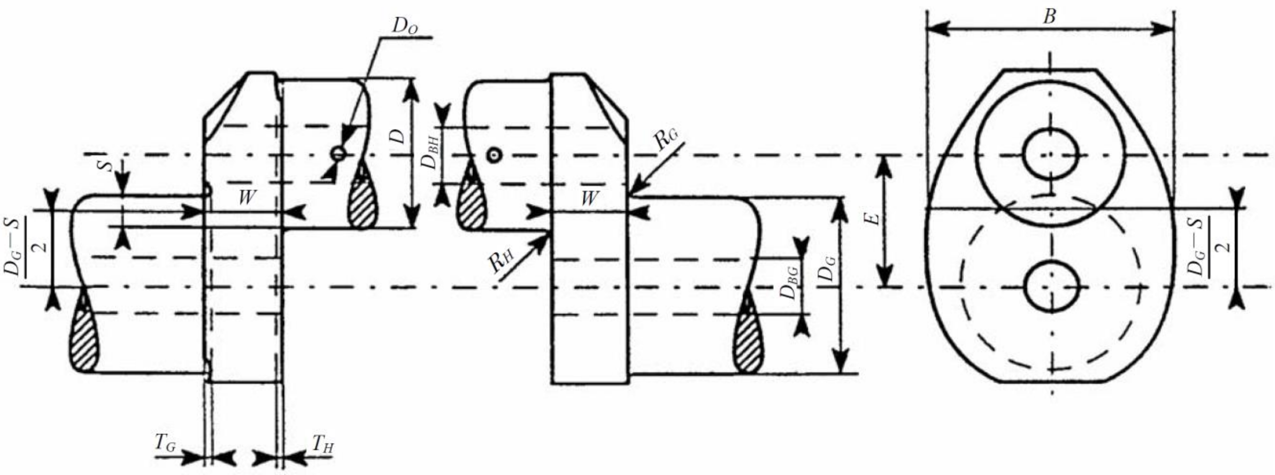
$d_H = D_{BH}/D$ $(0 \le d_H \le 0.8)$

$t_H = T_H/D$

$t_G = T_G/D$

Where the geometry of crankshaft is outside the above ranges, stress concentration factors in crankpin fillets, journal fillets and outlets of crankpin oil bores are to be calculated by utilizing the Finite Element Method (FEM) given in **Appendix 1** and **Appendix 4**. In such cases, care is to be taken to avoid mixing equivalent (von Mises) stresses and principal stresses. In cases where stress concentration factors are evaluated by methods other than theses, relevant documents and the analysis method adopted are to be submitted to the Society in order to demonstrate their equivalence to the methods specified in this paragraph.

Fig. 5 Crank Dimensions



1.4.2 Stress Concentration Factors in Crankpin Fillets

1 The stress concentration factor for bending (α_B) is as follows:

$$\alpha_B = 2.6914 \cdot f(s,w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(recess)$$

where

$$f(s,w) = -4.1883 + 29.2004 \cdot w - 77.5925 \cdot w^2 + 91.9454 \cdot w^3 - 40.0416 \cdot w^4$$
$$+(1-s) \cdot (9.5440 - 58.3480 \cdot w + 159.3415 \cdot w^2 - 192.5846 \cdot w^3 + 85.2916 \cdot w^4)$$
$$+(1-s)^2 \cdot (-3.8399 + 25.0444 \cdot w - 70.5571 \cdot w^2 + 87.0328 \cdot w^3 - 39.1832 \cdot w^4)$$

If $s < -0.5$, then $f(s,w)$ is to be calculated after replacing the actual value of s by -0.5.

$$f(w) = 2.1790 \cdot w^{0.7171}$$
$$f(b) = 0.6840 - 0.0077 \cdot b + 0.1473 \cdot b^2$$
$$f(r) = 0.2081 \cdot r^{(-0.5231)}$$
$$f(d_G) = 0.9993 + 0.27 \cdot d_G - 1.0211 \cdot d_G^2 + 0.5306 \cdot d_G^3$$
$$f(d_H) = 0.9978 + 0.3145 \cdot d_H - 1.5241 \cdot d_H^2 + 2.4147 \cdot d_H^3$$
$$f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$$

If calculated $f(recess) < 1$, then the factor $f(recess)$ is not to be considered ($f(recess) = 1$).

2 The stress concentration factor for torsion (α_T) is as follows:

$$\alpha_T = 0.8 \cdot f(r,s) \cdot f(b) \cdot f(w)$$

where

$$f(r,s) = r^{(-0.322 + 0.1015(1-s))}$$

If $s < -0.5$, then $f(r,s)$ is to be calculated by replacing the actual value of s by -0.5.

$$f(b) = 7.8955 - 10.654 \cdot b + 5.3482 \cdot b^2 + 0.857 \cdot b^3$$
$$f(w) = w^{(-0.145)}$$

1.4.3 Stress Concentration Factors in Journal Fillets

1 The stress concentration factor for bending (β_B) is as follows:

$$\beta_B = 2.7146 \cdot f_B(s,w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(recess)$$

where

$$f_B(s,w) = -1.7625 + 2.9821 \cdot w - 1.5276 \cdot w^2 + (1-s) \cdot (5.1169 - 5.8089 \cdot w + 3.1391 \cdot w^2)$$
$$+(1-s)^2 \cdot (-2.1567 + 2.3297 \cdot w - 1.2952 \cdot w^2)$$
$$f_B(w) = 2.2422 \cdot w^{0.7548}$$
$$f_B(b) = 0.5616 + 0.1197 \cdot b + 0.1176 \cdot b^2$$
$$f_B(r) = 0.1908 \cdot r^{(-0.5568)}$$
$$f_B(d_G) = 1.0012 - 0.6441 \cdot d_G + 1.2265 \cdot d_G^2$$
$$f_B(d_H) = 1.0022 - 0.1903 \cdot d_H + 0.0073 \cdot d_H^2$$
$$f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$$

If calculated $f(recess) < 1$, then the factor $f(recess)$ is not to be considered ($f(recess) = 1$).

2 The stress concentration factor for compression (β_Q) due to the radial force is as follows:

$$\beta_Q = 3.0128 \cdot f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(recess)$$

where

$$f_Q(s) = 0.4368 + 2.1630 \cdot (1-s) - 1.5212 \cdot (1-s)^2$$
$$f_Q(w) = \frac{w}{0.0637 + 0.9369 \cdot w}$$
$$f_Q(b) = -0.5 + b$$
$$f_Q(r) = 0.5331 \cdot r^{(-0.2038)}$$
$$f_Q(d_H) = 0.9937 - 1.1949 \cdot d_H + 1.7373 \cdot d_H^2$$
$$f(recess) = 1 + (t_H + t_G) \cdot (1.8 + 3.2 \cdot s)$$

If calculated $f(recess) < 1$, then the factor $f(recess)$ is not to be considered ($f(recess) = 1$).

3 The stress concentration factor for torsion (β_T) is as follows:

$\beta_T = \alpha_T$ if diameters and fillet radii of crankpins and journals are the same.

$\beta_T = 0.8 \cdot f(r,s) \cdot f(b) \cdot f(w)$ if crankpin and journal diameters and/or radii are of different sizes.

where

$f(r,s)$, $f(b)$ and $f(w)$ are to be determined in accordance with **1.4.2** (*See* calculation of α_T). However, the radius of the journal fillet is to be related to the journal diameter:

$$r = \frac{R_G}{D_G}$$

1.4.4 Stress Concentration Factors in Outlet of Crankpin Oil Bore

1 The stress concentration factor for bending (γ_B) is:

$$\gamma_B = 3 - 5.88 \cdot d_O + 34.6 \cdot {d_O}^2$$

2 The stress concentration factor for torsion (γ_T) is:

$$\gamma_T = 4 - 6 \cdot d_O + 30 \cdot {d_O}^2$$

1.5 Additional Bending Stresses

In addition to the alternating bending stresses in fillets (σ_{BH} and σ_{BG}) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying σ_{add} as follows:

σ_{add} = ±30 N/mm^2 for crosshead engines

= ±10 N/mm^2 for trunk piston engines

(*) The additional stress of ±30 N/mm^2 is composed of the following two components:

(1) an additional stress of ±20 N/mm^2 resulting from axial vibration

(2) an additional stress of ±10 N/mm^2 resulting from misalignment or bedplate deformation

It is recommended that a value of ±20 N/mm^2 be used for the axial vibration component for assessment purposes in cases where axial vibration calculation results of the complete dynamic system (engine, shafting, gears and propellers) are not available. However, in cases where axial vibration calculation results of the complete dynamic system are available, the calculated figures may be used instead.

1.6 Equivalent Alternating Stress

1.6.1 Equivalent Alternating Stress in Crankpin Fillets

Equivalent alternating stress in crankpin fillets is calculated in accordance with the following:

$$\sigma_V = \pm\sqrt{(\sigma_{BH} + \sigma_{add})^2 + 3\tau_H^2}$$

where

σ_V : Equivalent alternating stress (N/mm^2)

for other parameters *see* **1.3.1-3**, **1.3.2-2** and **1.5**.

1.6.2 Equivalent Alternating Stress in Journal Fillets

Equivalent alternating stress in journal fillets is calculated according to the following:

$$\sigma_V = \pm\sqrt{(\sigma_{BG} + \sigma_{add})^2 + 3\tau_G^2}$$

for parameters *see* **1.6.1**.

1.6.3 Equivalent Alternating Stress in Outlets of Crankpin Oil Bores

Equivalent alternating stress in outlets of crankpin oil bores is calculated according to the following:

$$\sigma_V = \pm\frac{1}{3}\sigma_{BO}\cdot\left[1 + 2\sqrt{1 + \frac{9}{4}\left(\frac{\sigma_{TO}}{\sigma_{BO}}\right)^2}\right]$$

for parameters *see* **1.6.1**.

1.7 Fatigue Strength

1.7.1 Fatigue Strength in Crankpin Fillets

1 The fatigue strength in crankpin fillets is evaluated according to the following: (For calculation purposes, R_H is to be taken as not less than 2 *mm*.)

$$\sigma_{DW} = \pm K[0.42\sigma_B + 39.3]\times\left[0.264 + 1.073D^{-0.2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_H}}\right]$$

where

σ_{DW} : Allowable fatigue strength of crankshaft (N/mm^2) in cases where the surfaces of fillets, the outlets of oil bores and the insides of oil bores (down to a minimum depth equal to 1.5 *times* the oil bore diameter) are all smoothly finished

K : Factor for the different types of crankshafts without surface treatment

= 1.05 for continuous grain flow forged or drop-forged crankshafts

= 1.0 for free form forged crankshafts (without continuous grain flow)

Factor for cast steel crankshafts with cold rolling treatment in fillet areas

= 0.93 for cast steel crankshafts manufactured using a cold rolling process approved by the Society

As an alternative, the value of K can be determined by experiments based either on full-size crank throws (or crankshafts) or on specimens taken from full-size crank throws.

σ_B : Minimum tensile strength of crankshaft material (N/mm^2)

for other parameters *see* **1.4**

2 In cases where the fatigue strength of the crankshaft is determined by experiment based either on full-size crank throw (or crankshaft) or on specimens taken from a full-size crank throw, evaluation of test results is to be carried out in accordance with **Appendix 2** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

3 In cases where a surface treatment process is applied to the fillets, every surface-treated area is to be specified on the drawing and the fatigue strength calculations are to be carried out in accordance with **Appendix 3** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

1.7.2 Fatigue Strength in Journal Fillets

The fatigue strength in journal fillets is evaluated according to the following: (For calculation purposes, R_G is to be taken as not less than 2 mm.)

$$\sigma_{DW} = \pm K[0.42\sigma_B + 39.3] \times \left[0.264 + 1.073 {D_G}^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{1}{R_G}}\right]$$

for parameters *see* **1.7.1**

1.7.3 Fatigue Strength in Outlets of Crankpin Oil Bores

1 The fatigue strength in outlets of crankpin oil bores is evaluated according to the following: (For calculation purposes, $D_O/2$ is to be taken as not less than 2 mm.)

$$\sigma_{DW} = \pm K[0.42\sigma_B + 39.3] \times \left[0.264 + 1.073 D^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B}\sqrt{\frac{2}{D_O}}\right]$$

K : Factor for forged crankshafts without surface treatment

= 1.0

Factor for cast steel crankshafts with cold rolling treatment in fillet areas

= 0.93 for cast steel crankshafts manufactured using a cold rolling process approved by the Society

As an alternative, the value of K can be determined by experiments based either on full-size crank throws (or crankshafts) or on specimens taken from full-size crank throws.

for other parameters *see* **1.7.1**

2 In cases where the fatigue strength of the crankshaft is determined by experiment based either on full-size crank throw (or crankshaft), evaluation of test results is to be carried out in accordance with **Appendix 2** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

3 In cases where a surface treatment process is applied to the outlets of oil bores, every surface treated area is to be specified on the drawing and the fatigue strength calculations are to be carried out in accordance with **Appendix 3** or methods considered by the Society to be equivalent. The test results as well as relevant documents are to be submitted to the Society.

1.8 Acceptability Criteria

In order to determine whether the dimensions of crankshafts are sufficient, comparisons between equivalent alternating stresses and fatigue strength are to be made. The acceptability factors of crankpin fillets, journal fillets and the outlets of crankpin oil bores are to comply with the following criteria:

$$Q \geq 1.15$$

where

Q : Acceptability factor

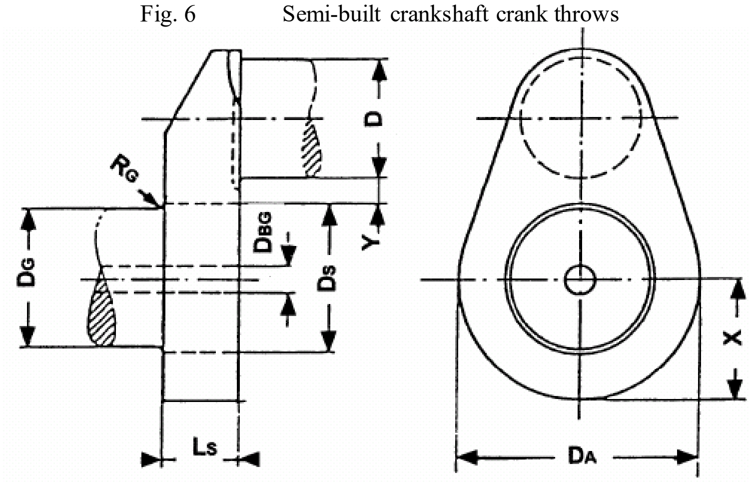
$$= \frac{\sigma_{DW}}{\sigma_V}$$

1.9 Semi-Built Crankshaft Shrink-Fit Calculations

1.9.1 General

1 All crank dimensions necessary for the calculation of the shrink-fit are shown in **Fig. 6**.

Fig. 6 Semi-built crankshaft crank throws



D_A : Outside diameter of web or twice the minimum distance x between centre line of journals and outer contour of web, whichever is less (mm)

D_S : Shrink diameter (mm)

D_G : Journal diameter (mm)

D_{BG} : Journal axial bore diameter (mm)

L_S : Shrink-fit length (mm)

R_G : Journal fillet radius (mm)

y : Distance between the adjacent generating lines of journal and pin (mm)

$y \geq 0.05 \cdot D_S$

Where y is less than $0.1 \cdot D_S$, special consideration is to be given to the effect of the stress due to the shrink-fit on the fatigue strength at the crankpin fillet.

2 Respecting the radius of the transition from the journal to the shrink diameter, the following are to be complied with:

$$R_G \geq 0.015 \cdot D_G$$

$$R_G \geq 0.5 \cdot (D_S - D_G)$$

where the greater value is to be considered.

3 The actual oversize Z of the shrink-fit is to be within the limits $Zmin$ and $Zmax$ calculated in accordance with **1.9.3** and **1.9.4**. In cases where the conditions given in **1.9.2** cannot be fulfilled, the above $Zmin$ and $Zmax$ are not applicable due to multizone-plasticity problems. In such cases, $Zmin$ and $Zmax$ are to be obtained through FEM calculations.

1.9.2 Journal Axial Bore Diameters

Journal axial bore diameters are to comply with the following formula:

$$D_{BG} \leq D_S \cdot \sqrt{1 - \frac{4000 \cdot S_R \cdot M_{\max}}{\mu \cdot \pi \cdot D_S^2 \cdot L_S \cdot \sigma_{SP}}}$$

S_R : Safety factor against slipping (a value not less than 2 is to be taken)

M_{max} : Absolute maximum torque at the shrinkage fit $(N \cdot m)$

μ : Coefficient for static friction (a value not greater than 0.2 is to be taken)

σ_{SP} : Minimum yield strength of material used for journal (N/mm^2)

1.9.3 Necessary Minimum Shrink-Fit Oversize

The necessary minimum oversize is determined by the greater value calculated according to the following formula:

$$Z_{min} \geq \frac{\sigma_{SW} \cdot D_S}{E_m}$$

$$Z_{min} \geq \frac{4000}{\mu \cdot \pi} \cdot \frac{S_R \cdot M_{\max}}{E_m \cdot D_S \cdot L_S} \cdot \frac{1 - Q_A^2 \cdot Q_S^2}{(1 - Q_A^2) \cdot (1 - Q_S^2)}$$

Z_{min} : Minimum oversize (mm)

E_m : Young's modulus (N/mm^2)

σ_{SW} : Minimum yield strength of material for crank web (N/mm^2)

Q_A : Web ratio, $Q_A = {}^{D_S}/_{D_A}$

Q_S : Shaft ratio, $Q_S = {}^{D_{BG}}/_{D_S}$

1.9.4 Maximum Permissible Shrink-Fit Oversize

The maximum permissible oversize is calculated according to the following formula:

$$Z_{max} \leq D_S \cdot \left(\frac{\sigma_{SW}}{E_m} + \frac{0.8}{1000}\right)$$

Annex 5.3.1 CALCULATION OF STRENGTH OF ENCLOSED GEARS

1.1 Application and Basic Principles

1.1.1 Application

This annex applies to enclosed gears used for transmission systems which transmit power from main propulsion machinery and prime movers driving generators and essential auxiliaries (excluding auxiliary machinery for specific use, etc., hereinafter the same in this annex).

1.1.2 Basic Principles

The gear strength calculation methods specified in this annex deal with surface durability (pitting) and tooth root bending strength. All influence factors related to strength are defined regarding their physical interpretation. Some of these factors are to be determined either by gear geometry etc,. Other factors are to be approximated according to methods deemed acceptable by the Society.

1.2 Symbols and Units

The main symbols introduced in this annex are listed below.

a : center distance (mm)

b : common facewidth (mm)

$b_{1,2}$: facewidth of pinion , wheel (mm)

d : reference diameter (mm)

$d_{1,2}$: reference diameter of pinion, wheel (mm)

$d_{a1,2}$: tip diameter of pinion, wheel (mm)

$d_{b1,2}$: base diameter of pinion, wheel (mm)

$d_{f1,2}$: root diameter of pinion, wheel (mm)

$d_{w1,2}$: working diameter of pinion, wheel (mm)

F_t : nominal tangential load (N)

h : tooth depth (mm)

m_n : normal module (mm)

m_t : transverse module (mm)

$n_{1,2}$: rotational speed of pinion, wheel (rpm)

P : maximum continuous power transmitted by the gear set (kW)

$T_{1,2}$: torque in way of pinion, wheel (Nm)

u : gear ratio

v : linear speed at pitch diameter (m/s)

$x_{1,2}$: addendum modification coefficient of pinion, wheel

z : number of teeth

$z_{1,2}$: number of teeth of pinion, wheel

z_n : virtual number of teeth

α_n : normal pressure angle at reference cylinder (°)

α_t : transverse pressure angle at reference cylinder (°)

α_{tw} : transverse pressure angle at working pitch cylinder (°)

β : helix angle at reference cylinder (°)

β_b : helix angle at base cylinder (°)

ε_α : transverse contact ratio

ε_β : overlap contact ratio

ε_γ : total contact ratio

σ_H : contact stress at the operating pitch point or at the inner point of single pair contact (N/mm^2)

σ_{HO} : basic value of contact stress (N/mm^2)

K_A : application factor

K_γ : load sharing factor

K_V : dynamic factor

$K_{H\alpha}$: transverse load distribution factor for contact stress

α_{Pn} : normal pressure angle of basic rack for cylindrical gear (°)

h_{fp} : dedendum of basic rack for cylindrical gear (mm)

$K_{H\beta}$: face load distribution factor for contact stress

σ_{HP} : permissible contact stress (N/mm^2)

σ_{Hlim} : endurance limit for contact stress (N/mm^2)

Z_N : life factor for contact stress

Z_L : lubricant factor

Z_V : speed factor

Z_R : roughness factor

Z_W : hardness ratio factor

Z_X : size factor for contact stress

S_H : safety factor for contact stress

Z_B : single pair mesh factor for pinion

Z_D : single pair mesh factor for wheel

Z_H : zone factor

Z_E : elasticity factor ($\sqrt{N/mm^2}$)

Z_ε : contact ratio factor

Z_β : helix angle factor for contact stress

σ_F : tooth root bending stress (N/mm^2)

Y_F : tooth form factor

h_F : bending moment arm for tooth root bending stress for application of load at the outer point of single tooth pair contact (mm)

S_{FN} : tooth root chord in the critical section (mm)

α_{SFN} : pressure angle at the outer point of single tooth pair contact in the normal section (°)

Y_S : stress correction factor

ρ_F : root fillet radius in the critical section (mm)

ρ_{fp} : root fillet radius of the basic rack for cylindrical gears (mm)

S_{pr} : residual fillet undercut (mm)

Y_β : helix angle factor for tooth root bending stress

Y_B : rim thickness factor

s_R : rim thickness of gears (mm)

h : tooth height (mm)

Y_{DT} : deep tooth factor

$K_{F\alpha}$: transverse load distribution factor for tooth root bending stress

$K_{F\beta}$: face load distribution factor for tooth root bending stress

σ_{FP}: permissible tooth root bending stress (N/mm^2)

σ_{FE}: bending endurance limit (N/mm^2)

Y_N : life factor for tooth root bending stress

Y_d : design factor

$Y_{\delta\, relT}$: relative notch sensitivity factor

q_s : notch parameter

ρ' : slip-layer thickness (mm)

$Y_{R\, relT}$: relative surface factor

Y_X : size factor for tooth root bending stress

S_F : safety factor for tooth root bending stress

1.3 Geometrical Definitions

In the case of internal gearing z_2, a, d_2, d_{a2}, d_{b2} and d_{w2} are negative. The pinion is defined as the gear with the smaller number of teeth; therefore, the absolute value of the gear ratio, defined as follows, is always greater or equal to the unity.

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

In the case of external gears, u is positive. In the case of internal gears, u is negative. In the equation of surface durability, b is the common facewidth on the pitch diameter. In the equation of the tooth root, bending stress b_1 or b_2 are the facewidths at their respective tooth roots. In any case, b_1 and b_2 are not to be taken as greater than b by more than one module (m_n) on either side. The common facewidth b may be used also in the equation of teeth root bending stress if either significant crowning or end relief has been adopted.

$$\tan\alpha_t = \frac{\tan\alpha_n}{\cos\beta}$$

$$\tan\beta_b = \tan\beta\cos\alpha_t$$

$$d_{1,2} = \frac{z_{1,2}m_n}{\cos\beta}$$

$$d_{b1,2} = d_{1,2}\cos\alpha_t$$

$$\left.\begin{aligned} d_{w1} &= \frac{2a}{u+1} \\ d_{w2} &= \frac{2au}{u+1} \end{aligned}\right\} \text{where } a = 0.5(d_{w1} + d_{w2})$$

$$z_{n1,2} = \frac{z_{1,2}}{\cos^2\beta_b \cdot \cos\beta}$$

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

$$\mathrm{inv}\alpha = \tan\alpha - \frac{\pi\alpha}{180};\ \alpha(°)$$

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

or

$$\cos\alpha_{tw} = \frac{m_t(z_1 + z_2)}{2a}\cos\alpha_t$$

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

A positive sign is used for external gears, a negative sign for internal gears.

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

In the case of double helix gears, b is to be taken as the width of one helix.

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

$$v = \frac{\pi d_{1,2} n_{1,2}}{60 \cdot 10^3}$$

1.4 Nominal Tangential Load, F_t

Nominal tangential loads, F_t, which are tangential to cylinders and perpendicular to planes are to be calculated directly from the maximum continuous power transmitted by gear sets using the following equations:

$$T_{1,2} = \frac{30 \cdot 10^3 P}{\pi n_{1,2}}$$

$$F_t = 2000 \frac{T_{1,2}}{d_{1,2}}$$

1.5 Loading Factors

1.5.1 Application Factor, K_A

1 The application factor, K_A, accounts for dynamic overloads from source external to the gearing. K_A for gears designed for infinite lifespans is defined as the ratio between maximum repetitive cyclic torques applied to gear sets and nominal rated torques. Nominal rated torque is defined by rated power and speed and is the torque used in rating calculations. This factor mainly depends on the following:

(1) The characteristics of driving and driven machines;

(2) The ratio of masses;

(3) The type of couplings;

(4) Operating conditions (over speed, changes in propeller load conditions, etc.)

2 In cases where drive systems are operating at level near their critical speed, a careful analysis of conditions is to be made. K_A is to be determined either by direct measurements or by a system analysis that is acceptable to the Society. In cases where values determined in such ways cannot be provided, the following values may be used:

(1) Main propulsion

K_A = 1.00 (reciprocating internal combustion engines with hydraulic or electromagnetic slip couplings)
= 1.30 (reciprocating internal combustion engines with high elasticity couplings)
= 1.50 (reciprocating internal combustion engines with other couplings)

However, in cases where vessels using reduction gears are affixed with **Ice Class** Notation, as required in **8.6, Part I of the Rules**.

(2) Auxiliary gears

K_A = 1.00 (electric motors, reciprocating internal combustion engines with hydraulic or electromagnetic slip couplings)
= 1.20 (reciprocating internal combustion engines with high elasticity couplings)
= 1.40 (reciprocating internal combustion engines with other couplings)

1.5.2 Load Sharing Factor, K_γ

The load sharing factor, K_γ, accounts for the maldistribution of loads in multiple path transmissions (dual tandems, epicyclics, double helixes, etc.). K_γ is defined as the ratio between those maximum loads through actual paths and those evenly distributed loads. This factor mainly depends on the accuracy and the flexibility of the branches. K_γ is to be determined by measurements or by system analysis. In cases where values determined in such ways cannot be provided, the following values can be used with respect to epicyclic gears:

K_γ = 1.00 (up to 3 planetary gears)
= 1.20 (4 planetary gears)
= 1.30 (5 planetary gears)
= 1.40 (6 planetary gears and over)

1.5.3 Internal Dynamic Factor, K_V

1 The internal dynamic factor, K_V, accounts for those internally generated dynamic loads due to vibrations of pinions and wheels against each other. K_V is defined as the ratio between those maximum loads which dynamically act on tooth flanks and maximum externally applied loads ($F_t K_A K_V$). This factor mainly depends on the following:

(1) Transmission errors depending on pitch and profile errors;

(2) Masses of pinions and wheels;

(3) Gear mesh stiffness variations as gear teeth pass through meshing cycles;

(4) Transmitted loads including application factors;

(5) Pitch line velocities;

(6) Dynamic unbalance of gears and shafts;

(7) Shaft and bearing stiffness;

(8) Damping characteristics of gear systems.

2 The internal dynamic factor, K_V, is to be calculated as follows; however, this method is to be applied only to cases where all of the following conditions **(1)** to **(4)** are satisfied:

(1) Running speeds in the following subcritical ranges:

$$\frac{v z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 10\ (m/s)$$

(2) $\beta = 0°$ (In the case of spur gears)

$\beta \leq 30°$ (In the case of helical gears)

(3) pinion with relatively low number of teeth:

$z_1 < 50$

(4) solid disc wheels or heavy steel gear rim

This method may be applied to all types of gears, if $\frac{v z_1}{100}\sqrt{\frac{u^2}{1+u^2}} < 3\ (m/s)$, as well as to helical gears where $\beta > 30°$. For gears other than the above, reference is to be made to Method B outlined in the reference standard *ISO* 6336-1:2019.

(a) For those helical gears with an overlap ratio $\geq$ unity and spur gears, the value of K_V is to be determined as follows:

$$K_V = 1 + \left(\frac{K_1}{K_A \frac{F_t}{b}} + K_2\right) \cdot \frac{v \cdot z_1}{100} K_3 \sqrt{\frac{u^2}{1+u^2}}$$

K_1 : Factor specified in **Table 5.3-1.**

K_2 : Factors for all *ISO* accuracy grades. Values are as follows:

= 0.0193 (In the case of spur gears)

= 0.0087 (In the case of helical gears)

K_3 : Values are to be calculated as follows:

$$= 2.0 \left(\frac{v \cdot z_1}{100}\sqrt{\frac{u^2}{1+u^2}} \leq 0.2\right)$$

$$= 2.071 - 0.357 \cdot \frac{v \cdot z_1}{100}\sqrt{\frac{u^2}{1+u^2}} \left(\frac{v \cdot z_1}{100}\sqrt{\frac{u^2}{1+u^2}} > 0.2\right)$$

If $K_A F_t / b$ is less than 100 *N/mm*, this value is assumed to be equal to 100 *N/mm*.

(b) In the case of helical gears with an overlap ratio < unity, the value of K_V is to be obtained by means of linear interpolation as follows:

$$K_V = K_{V2} - \varepsilon_\beta (K_{V2} - K_{V1})$$

K_{V1} : Values for helical gears specified in accordance with **(a)**

K_{V2} : Values for spur gears specified in accordance with **(a)**

In the case of mating gears with different grades of accuracy, the grade corresponding to the lower accuracy is to be used.

Table 5.3-1 Values of K_1

Type of gears	*ISO* grades of accuracy					
	3	4	5	6	7	8
Spur gears	2.1	3.9	7.5	14.9	26.8	39.1
Helical gears	1.9	3.5	6.7	13.3	23.9	34.8

Notes:

ISO accuracy grades according to *ISO* 1328-1:2013.

1.5.4 Face Load Distribution Factors, $K_{H\beta}$ and $K_{F\beta}$

The face load distribution factors, $K_{H\beta}$ for contact stress, and $K_{F\beta}$ for tooth root bending stress account for the effects of the non-uniform distribution of loads across facewidths. $K_{H\beta}$ is defined as the ratio between the maximum load per unit facewidth and the mean load per unit facewidth, and $K_{F\beta}$ is defined as the ratio between the maximum bending stress at tooth root per unit facewidth and the mean bending stress at tooth root per unit facewidth.

The mean bending stress at tooth root relates to the considered facewidth b_1 and b_2 respectively. $K_{F\beta}$ can be expressed as a function of the factor, $K_{H\beta}$. $K_{H\beta}$ and $K_{F\beta}$ mainly depend on the following:

(1) Gear tooth manufacturing accuracy;
(2) Errors in mounting due to bore errors;
(3) Bearing clearances;
(4) Wheel and pinion shaft alignment errors;
(5) Elastic deflections of gear elements, shafts, bearings, housing, and foundations which support the gear elements;
(6) Thermal expansion and distortion due to operating temperature;
(7) Compensating design elements (tooth crowning, end relief, etc.).

The value for $K_{H\beta}$ is to be determined as follows:

if $\dfrac{F_{\beta y}C_\gamma b}{2F_tK_AK_\gamma K_V} \geq 1$

then $K_{H\beta} = \sqrt{\dfrac{2F_{\beta y}C_\gamma b}{F_tK_AK_\gamma K_V}}$

if $\dfrac{F_{\beta y}C_\gamma b}{2F_tK_AK_\gamma K_V} < 1$

then $K_{H\beta} = 1 + \dfrac{F_{\beta y}C_\gamma b}{2F_tK_AK_\gamma K_V}$

C_γ : tooth stiffness parameter ($N/(mm \cdot \mu m)$)

The value for C_γ is to be calculated as follows:

$$C_\gamma = 0.8C_{th}C_RC_B\cos\beta(0.75\varepsilon_\alpha + 0.25)$$

$$C_{th} = \frac{1}{q}$$

$$q = 0.04723 + \frac{0.15551}{z_{n1}} - \frac{0.25791}{z_{n2}} - 0.00635x_1 - \frac{0.11654}{z_{n1}}x_1$$
$$-0.00193x_2 - \frac{0.24188}{z_{n2}}x_2 + 0.00529x_1^2 + 0.00182x_2^2$$

$C_R = 1$ (In the case of solid disc gears)

$C_R = 1 + \dfrac{\ln\left(\frac{b_s}{b}\right)}{5e^{\left(\frac{S_R}{5m_n}\right)}}$ (In the case of non-solid disc gears)

b_s : thickness of central web (mm)

S_R : average thickness of rim (mm)

However, in cases where for non-solid disc gears of $\dfrac{b_s}{b} < 0.2$ or $\dfrac{S_R}{m_n} < 1.0$, the value for C_R is to be determined by the Society on a case by case basis.

C_B is a basic rack coefficient that accounts for the deviations between the actual basic rack profile of gears and their standard basic rack profile. The value for C_B is to be calculated as follows:

$$C_B = \left[1 + 0.5\left(1.2 - \frac{h_{fp}}{m_n}\right)\right][1\text{-}0.02(20^\circ - \alpha_{Pn})]$$

$F_{\beta y}$ =effective equivalent misalignment (μm). The value for $F_{\beta y}$ is to be calculated as follows:

$$F_{\beta y} = F_{\beta x} - y_\beta$$

In the case of gears that are not surface hardened

$$y_\beta = \frac{320}{\sigma_{Hlim}} F_{\beta x}$$

However, the following conditions are to be satisfied.

$y_\beta \leq F_{\beta x}$

$y_\beta \leq 25600/\sigma_{Hlim}$ (5<v<10 m/sec)

$y_\beta \leq 12800/\sigma_{Hlim}$ (10 m/sec<v)

In the case of surface hardened gears

$$y_\beta = 0.15 F_{\beta x}$$

However, $y_\beta \leq 6.0$ (μm) is to be satisfied.

$F_{\beta x}$: original effective equivalent misalignment (μm), $F_{\beta x}$ is to be calculated as follows:

$$F_{\beta x} = 1.33 f_{sh} + f_{ma}$$

f_{sh}: takes into account the components of equivalent misalignment resulting from bending and twisting of pinion and pinion shaft, f_{sh} is to be calculated as follows (μm):

In the case of gears without crowning or end relief

$$f_{sh} = 0.023 \frac{F_t K_A K_\gamma K_V \gamma}{b}$$

In the case of gears with end relief

$$f_{sh} = 0.016 \frac{F_t K_A K_\gamma K_V \gamma}{b}$$

In the case of gears with crowning

$$f_{sh} = 0.012 \frac{F_t K_A K_\gamma K_V \gamma}{b}$$

In the case of gears with helix angle modification

$$f_{sh} = 0$$

However, in all cases f_{sh} is not to be taken as value less than that calculated by the following expressions:

$$0.005 \frac{F_t K_A K_\gamma K_V}{b} \quad \text{(In the case of spur gears)}$$

or

$$0.010 \frac{F_t K_A K_\gamma K_V}{b} \quad \text{(In the case of helical gears)}$$

γ = pinion ratio factor. The value for γ is to be calculated as follows:

$$\gamma = \left[\left|0.7 + K' \frac{\ell S}{d_1^2}\left(\frac{d_1}{d_{sh}}\right)^4\right| + 0.3\right]\left(\frac{b}{d_1}\right)^2 \quad \text{(In the case of spur and helical gears)}$$

$$\gamma = 2\left[\left|1.2 + K' \frac{\ell S}{d_1^2}\left(\frac{d_1}{d_{sh}}\right)^4\right| + 0.3\right]\left(\frac{b}{d_1}\right)^2 \quad \text{(In the case of double helical gears)}$$

K', ℓ and S are constant factors used for the calculation of the pinion ratio factor, γ, the bearing span and the distance between mid-plane of pinion and middle of such bearing spans, respectively. Values for K' are given in **Table 5.5-1**.

f_{ma} = the misalignment resulting from manufacturing errors (μm). The value for f_{ma} is to be calculated as follows:

$f_{ma} = 1.0 F_\beta$ (In the case of the assembly of gears without any modification or adjustment)

$= 0.7F_\beta$ (In the case of gear pairs with well-designed end relief)

$= 0.5F_\beta$ (In the case of gear pairs with means for adjustment or with helix modifications or suitably crowned)

F_β : tolerance on total helix deviation (μm)

The value for $K_{F\beta}$ is to be determined as follows:

(1) In cases where the hardest contact is at the end of the facewidth, $K_{F\beta}$ is to be calculated as follows:

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

$$N = \frac{\left(\frac{b}{h}\right)^2}{1 + \frac{b}{h} + \left(\frac{b}{h}\right)^2}$$

b/h= facewidth/tooth height ratio, the smaller of b_1/h_1 or b_2/h_2. In the case of double helical gears, the facewidth of only one is to be used. However, in cases where $b/h<3.0$, b/h is to be taken as 3.0.

(2) In cases of gears where the ends of the facewidth are lightly loaded or unloaded (end relief or crowning), the value for $K_{F\beta}$ is to be calculated as follows:

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

Table 5.5-1 Values of K'

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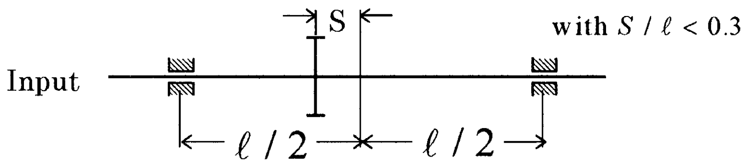
<tr><td>
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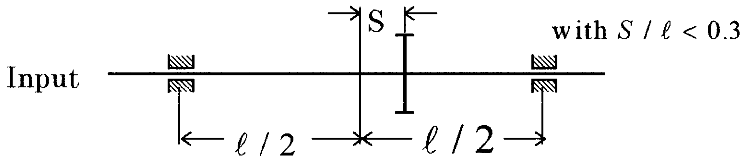
<tr><td>
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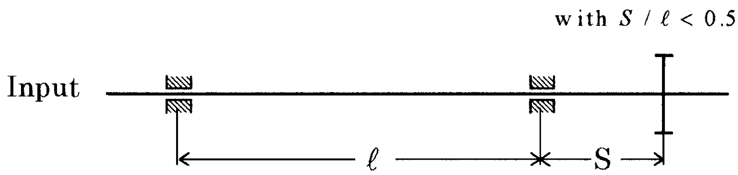
<tr><td>
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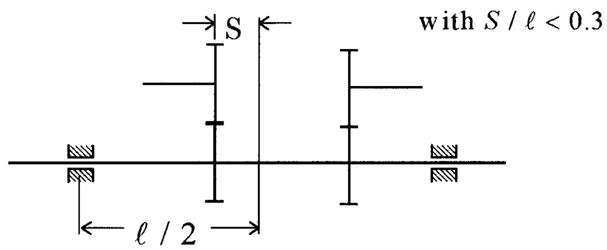
<tr><td>
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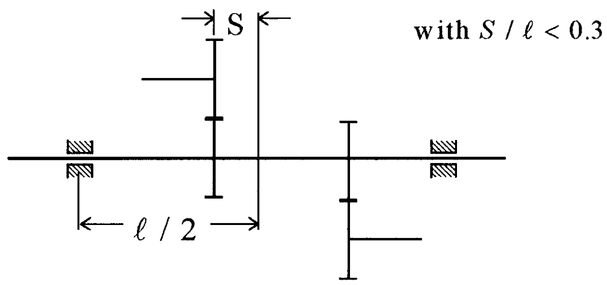
<tr><td>
</td><td>-0.6</td><td>-1.0</td></tr>
</table>

Notes:

1) In cases where $d_1/d_{sh} \geq 1.15$, stiffening is assumed.

2) In cases where $d_1/d_{sh} < 1.15$, or where the pinion slides on a shaft or is shrink fitted, no stiffening is assumed.

However, d_{sh} is the external diameter of a solid shaft equivalent to the actual one in bending deflection (mm)

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

The transverse load distribution factors, $K_{H\alpha}$ for contact stress and $K_{F\alpha}$ for tooth root bending stress, account for the effects of pitch and profile errors on the transversal load distribution between two or more pairs of teeth in mesh. $K_{H\alpha}$ and $K_{F\alpha}$ mainly depend on the following:

(1) Total mesh stiffness;

(2) Total tangential loads, $F_t\ K_A\ K_\gamma\ K_V\ K_{H\beta}$

(3) Base pitch errors

(4) Tip relief

(5) Running-in allowances

$K_{H\alpha}$ and $K_{F\alpha}$ are to be determined as follows:

$$K_{H\alpha} = K_{F\alpha} = \frac{\varepsilon_\gamma}{2}\left(0.9 + 0.4\frac{C_\gamma(f_{pb}-y_\alpha)^b}{F_tK_AK_\gamma K_V K_{H\beta}}\right) \quad (\varepsilon_\gamma \leq 2)$$

$$K_{H\alpha} = K_{F\alpha} = 0.9 + 0.4\sqrt{\frac{2(\varepsilon_\gamma-1)}{\varepsilon_\gamma}\frac{C_\gamma(f_{pb}-y_\alpha)^b}{F_tK_AK_\gamma K_V K_{H\beta}}} \quad (\varepsilon_\gamma > 2)$$

however

$$1.0 \leq K_{H\alpha} \leq \frac{\varepsilon_\gamma}{\varepsilon_\alpha Z_\varepsilon^2}$$

$$1.0 \leq K_{F\alpha} \leq \frac{\varepsilon_\gamma}{\varepsilon_\alpha Y_\varepsilon}$$

where

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_{\alpha n}}$$

The value for $\varepsilon_{\alpha n}$ is the same as that specified in **1.7.2**.

All symbols used in the equation to determine $K_{H\alpha}$ and $K_{F\alpha}$, except for y_α and f_{pb}, are the same as those used in the equation for determining $K_{H\beta}$.

y_α and f_{pb} are to be calculated as follows:

$$y_\alpha = \frac{160}{\sigma_{Hlim}}f_{pb} \quad \text{(In the case of through hardened gears)}$$

$$= 0.075f_{pb} \quad \text{(In the case of surface hardened gears)}$$

f_{pb} is to be taken as the larger value of base pitch deviation of pinions or wheels (μm)

1.6 Surface Strength

1.6.1 Equation

The criterion for surface strength is based on the Hertz pressure on operating pitch points or at inner points of single pair contacts. This criterion, as given by the following equation, is that contact stress σ_H is to be equal to or less than permissible contact stress σ_{HP}.

$$\sigma_H = \sigma_{HO}\sqrt{K_AK_\gamma K_V K_{H\alpha}K_{H\beta}} \leq \sigma_{HP}$$

where

σ_{HO} is basic value of contact stress for pinions and wheels (N/mm^2).

1.6.2 Equations for Basic Contact Stress

1 Basic contact stresses for pinions and wheels are to be calculated as follows:

$$\sigma_{HO} = Z_BZ_HZ_EZ_\varepsilon Z_\beta\sqrt{\frac{F_t}{d_1b}\frac{u+1}{u}} \quad \text{(In the case of pinions)}$$

$$= Z_DZ_HZ_EZ_\varepsilon Z_\beta\sqrt{\frac{F_t}{d_1b}\frac{u+1}{u}} \quad \text{(In the case of wheels)}$$

2 Single Pair Mesh Factors Z_B and Z_D

The single-pair mesh factors, Z_B for pinions and Z_D for wheels, account for the influence on contact stress of tooth flank curvatures at inner points of single pair contacts. These factors transform those contact stresses determined at pitch points to contact stresses considering flank curvatures at inner points of single pair contacts. Z_B and Z_D are to be determined as follows:

(1) In the case of spur gears, the value for Z_B is to be taken as 1.0 or as follows, whichever is greater.

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

(2) In the case of helical gears with $\varepsilon_\beta \geq 1$, the value for Z_B is to be taken as 1.0;

(3) In the case of helical gears with $\varepsilon_\beta > 1$, the value for Z_B is to be taken as 1.0 or as follows, whichever is greater:

$$Z_B = M_1 - \varepsilon_\beta (M_1 - 1)$$

(4) In the case of spur gears, the value for Z_D is to be taken as 1.0 or as follows, whichever is greater.

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

(5) In the case of helical gears with $\varepsilon_\beta \geq 1$, Z_D is to be taken as 1.0.

(6) In the case of helical gears with $\varepsilon_\beta < 1$, Z_D is to be taken as 1.0 or as follows, whichever is greater.

$$Z_D = M_2 - \varepsilon_\beta (M_2 - 1)$$

(7) In the case of internal gears, Z_D is to be taken as 1.0.

3 Zone Factor, Z_H

The zone factor, Z_H, accounts for the influence on Hertzian pressure of tooth flank curvatures at pitch points and relates those tangential forces at reference cylinders to those normal forces at pitch cylinders. Z_H is to be calculated as follows:

$$Z_H = \sqrt{\frac{2\cos\beta_b}{\cos^2 \alpha_t \tan\alpha_{tw}}}$$

4 Elasticity Factor, Z_E

The elasticity factor, Z_E, accounts for the influence of the material properties E (modulus of elasticity) and υ (Poisson's ratio) on the Hertzian pressure. In the case of steel gears, Z_E is to be calculated as follows:

$$Z_E = 189.8(\sqrt{N/mm^2}$$

In other cases, reference is to be made to the reference standard *ISO* 6336-2:2019.

5 Contact Ratio Factor, Z_ε

The contact ratio factor, Z_ε, accounts for the influence of transverse contact ratios and overlap ratios on the specific surface loads of gears.

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}} \quad \text{(In the case of spur gears)}$$

$$= \sqrt{\frac{4-\varepsilon_\alpha}{3}(1-\varepsilon_\beta) + \frac{\varepsilon_\beta}{\varepsilon_\alpha}} \quad \text{(In the case of helical gears with } \varepsilon_\beta < 1\text{)}$$

$$= \sqrt{\frac{1}{\varepsilon_\alpha}} \quad \text{(In the case of helical gears with } \varepsilon_\beta \geq 1\text{)}$$

6 Helix Angle Factor, Z_β

The helix angle factor, Z_β, accounts for the influence of helix angles on surface durability, allowing for such variables as distribution of loads along lines of contact. Z_β is dependent only on helix angles and its value can be obtained by the following formula:

$$Z_\beta = \sqrt{\frac{1}{\cos\beta}}$$

where β is the reference helix angle.

1.6.3 Permissible Contact Stress

1 Permissible contact stress, σ_{HP} is to be calculated as follows:

$$\sigma_{HP} = \sigma_{Hlim} \frac{Z_N Z_L Z_V Z_R Z_W Z_X}{S_H}$$

2 Endurance Limit for Contact Stress, σ_{Hlim}

For a given material, σ_{Hlim} is the limit of repeated contact stress which can be permanently endured. σ_{Hlim} can be regarded as the level of contact stress which the material will endure without pitting for at least 5×10^7 load cycles. "Pitting" is defined in the case of non-surface hardened gears, the pitted area > 2 % of total active flank area; in the case of surface hardened gears, the pitted area > 0.5 % of total active flank area, or > 4 % of one particular tooth flank area. The σ_{Hlim} values are to correspond to a failure probability of 1 % or less.

The endurance limit mainly depends on the following:

(1) Material composition, cleanliness and defects;

(2) Mechanical properties;

(3) Residual stresses;

(4) Hardening process, depth of hardened zone, hardness gradient;

(5) Material structure (forged, rolled bar, cast).

Endurance limit for contact stress σ_{Hlim} is as given in **Table 6.3-1**. However, for materials having enough data showing their higher endurance limit, values larger than those given in the table may be allowed by the Society in consideration of factors **(1)** through **(5)** mentioned above.

Table 6.3-1 Value of σ_{Hlim}(N/mm^2)

Steel type	σ_{Hlim}
Normalized structural steels	HB+190
Through hardening carbon steels	HB+350
Through hardening alloy steels	1.33HB+367
Induction hardened alloys	0.6HV+850
Nitrided alloys	1000
Soft nitrided alloys	1.14HV+437; however, 950 for HV>450
Nitrided steels	1250
Carburized hardened alloys	1500

Note:

HB: Brinell Hardness; HV: Vickers Hardness

3 Life Factor for Contact Stress, Z_N

The life factor for contact stress, Z_N, accounts for the higher permissible contact stress in cases where a limited life (number of cycles) is required. Values larger than 1.0 are to be considered by the Society on a case-by-case basis.

This factor mainly depends on the following:

(1) Material and heat treatment;

(2) Number of cycles;

(3) Influence factors (Z_R, Z_V, Z_L, Z_W, Z_x).

The life factor, Z_N, is to be determined according to Method B outlined in the reference standard *ISO* 6336-2:2019.

4 Lubricant Factor, Z_L

The lubricant factor, Z_L, like the speed factor, Z_V, and roughness factor, Z_R accounts for the influence of the type of lubricant and its viscosity on surface endurance. These factors are to be determined for softer materials in cases where gear pairs are of different hardness. These factors mainly depend on the following:

(1) Viscosity of lubricant in contact zones;

(2) The sum of the instantaneous velocity of tooth surfaces;

(3) Loads;

(4) Relative radius of curvature at pitch points;

(5) Surface roughness of teeth flanks;

(6) Hardness of pinions and wheels.

The value for Z_L is to be calculated as follows:

$$Z_L = C_{ZL} + \frac{4(1.0 - C_{ZL})}{(1.2 + 134/v_{40})^2}$$

where

$$C_{ZL} = \frac{\sigma_{Hlim} - 850}{350} 0.08 + 0.83$$

(In cases where 850 $N/mm^2 \leq \sigma_{Hlim} \leq$ 1200 N/mm^2)

= 0.83 (In cases where $\sigma_{Hlim} <$ 850 N/mm^2)

= 0.91 (In cases where $\sigma_{Hlim} >$1200 N/mm^2)

v_{40} : Nominal kinematic viscosity of the oil at 40℃ (mm^2/s)

5 Speed Factor, Z_V

The speed factor, Z_V, accounts for the influence of pitch line velocities on surface endurance. The value for Z_V is to be calculated as follows:

$$Z_V = C_{Z_V} + \frac{2(1.0 - C_{Z_V})}{\sqrt{0.8 + 32/v}}$$

where

$$C_{Z_V} = \frac{\sigma_{Hlim} - 850}{350} 0.08 + 0.85$$

(In cases where 850 $N/mm^2 \leq \sigma_{Hlim} \leq$ 1200 N/mm^2)

= 0.85 (In cases where $\sigma_{Hlim} <$ 850 N/mm^2)

= 0.93 (In cases where $\sigma_{Hlim} >$1200 N/mm^2)

6 Roughness Factor, Z_R

The roughness factor, Z_R, accounts for the influence of surface roughness on surface endurance. The value for Z_R is to be calculated as follows:

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

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

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

where

R_{Z1}, R_{Z2} : Respective mean peak to valley roughness for pinions and wheels.

R_Z : Refer to the reference standard *ISO* 6336-2:2019.

ρ_{red} : Relative radius of curvature. The value for ρ_{red} is to be calculated as follows:

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

$\rho_{1.2} = 0.5 \cdot d_{b1 \cdot 2} \cdot \tan\alpha_{tw}$ (In the case of internal gears the value d_b is negative)

In cases where the roughness stated is an arithmetic mean roughness, i.e. R_a value, and the conversion $R_Z = 6R_a$ can be applied.

$C_{ZR} = 0.32 - 0.0002\sigma_{Hlim}$ (In cases where 850 $N/mm^2 \leq \sigma_{Hlim} \leq$ 1200 N/mm^2)

= 0.15 (In cases where $\sigma_{Hlim} <$ 850 N/mm^2)

= 0.08 (In cases where $\sigma_{Hlim} >$1200 N/mm^2)

7 Hardness Ratio Factor, Z_W

The hardness ratio factor, Z_W, accounts for the increase in surface durability of soft steel gears meshing with significantly harder gears with smooth surfaces in the following cases:

(1) Surface-hardened pinion with through-hardened wheel

$$Z_W = 1.2\left(\frac{3}{R_{zH}}\right)^{0.15} \quad (HB < 130)$$

$$= \left(1.2 - \frac{HB-130}{1700}\right)\cdot\left(\frac{3}{R_{zH}}\right)^{0.15} \quad (130 \le HB \le 470)$$

$$= \left(\frac{3}{R_{zH}}\right)^{0.15} \quad (HB > 470)$$

where

HB : Brinell hardness of the tooth flanks of the softer gear of the pair

R_{ZH} : equivalent roughness (μm)

$$R_{zH} = \frac{R_{Z1}(10/\rho_{red})^{0.33}(R_{Z1}/R_{Z2})^{0.66}}{(v\cdot v_{40}/1500)^{0.33}}$$

(2) Through-hardened pinion and wheel

When the pinion is substantially harder than the wheel, the work hardening effect increases the load capacity of the wheel flanks. Z_W applies to the wheel only, not to the pinion.

$$Z_W = 1 \quad (HB_1/HB_2 < 1.2)$$

$$= 1 + \left(0.00898\frac{HB_1}{HB_2} - 0.00829\right)\cdot(u-1) \quad (1.2 \le HB_1/HB_2 \le 1.7)$$

$$= 1 + 0.00698\cdot(u-1) \quad (HB_1/HB_2 > 1.7)$$

$HB_{1,2}$: Brinell hardness of the pinion and the wheel respectively.

If gear ratio u>20, then the value u=20 is to be used. In any case, if the calculated Z_W <1, then the value Z_W = 1 is to be used.

(3) In cases other than **(1)** and **(2)** above;

8 Size Factor for Contact Stress, Z_X

The size factor for contact stress, Z_X, accounts for the influence of tooth dimensions on permissible contact stress and reflects the inhomogeneity of material properties. This factor mainly depends on the following:

(1) Materials and heat treatments;

(2) Tooth and gear dimensions;

(3) Ratio of case depth to tooth size;

(4) Ratio of case depth to equivalent radius of curvature.

For through hardened gears and for surface hardened gears with adequate case depth relative to tooth size and radius of relative curvature Z_X= 1.0, in cases where the case depth is relatively shallow then a smaller value of Z_X is to be taken.

9 Safety Factor for Contact Stress, S_H

The safety factor for contact stress, S_H, is to be taken as follows:

(1) In the case of main propulsion gears: 1.20

(2) In the case of auxiliary gears: 1.15

In cases where the gears of duplicated independent propulsion or auxiliary machinery, has been duplicated beyond that what is required for its respective class, a reduced value may be taken at the discretion of the Society.

1.7 Bending Strength

1.7.1 Equation

The tooth root bending stress σ_F and the permissible tooth root bending stress σ_{FP} are to be calculated separately for the pinion and the wheel. The criterion for tooth root bending strength, as given by the following equation, is that the tooth root bending stress in the tooth root fillet σ_F is to be equal to or less than the permissible tooth root bending stress σ_{FP}.

$$\sigma_F = \frac{F_t}{bm_n}Y_F Y_S Y_\beta Y_B Y_{DT} K_A K_\gamma K_V K_{F\alpha} K_{F\beta} \le \sigma_{FP}$$

However, the following definitions and equations apply only to those gears having a rim thickness greater than 3.5 m_n. The results of calculations using the following method are acceptable for normal pressure angles up to 25 degrees and reference helix angles up to 30 degrees. In the case of larger pressure angles and larger helix angles, the calculated results are to be confirmed by experience as by

Method A of the reference standard *ISO* 6336-3:2019.

1.7.2 Tooth Root Bending Stress for Pinion and Wheel

1 Tooth Form Factor, Y_F

The tooth form factor, Y_F, accounts for the influence on nominal bending stress of the tooth form with load applied at the outer point of single pair tooth contact. Y_F is to be determined separately for the pinion and the wheel. In the case of helical gears, the form factors for gearing are to be determined in normal sections (i.e. for virtual spur gears with virtual numbers of teeth, Z_n).The value for Y_F is to be calculated as follows:

$$Y_F = \frac{6\frac{h_F}{m_n}\cos\alpha_{Fen}}{\left(\frac{S_{Fn}}{m_n}\right)^2\cos\alpha_n}$$

S_{Fn}, h_F and α_{Fen} are to be calculated as follows:

$$S_{Fn} = m_n z_n \sin\left(\frac{\pi}{3}-\theta\right) + \sqrt{3}m_n\left(\frac{G}{\cos\theta}-\frac{\rho_{fp}}{m_n}\right)$$

$$G = \frac{\rho_{fp}}{m_n} - \frac{h_{fp}}{m_n} + x$$

$$\theta = \frac{2G}{z_n}\tan\theta - \frac{2}{z_n}\left(\frac{\pi}{2}-\frac{E}{m_n}\right)+\frac{\pi}{3}$$

$$E = \frac{\pi}{4}m_n - h_{fp}\tan\alpha_n + \frac{S_{Pr}}{\cos\alpha_n} - (1-\sin\alpha_n)\frac{\rho_{fp}}{\cos\alpha_n}$$

S_{pr} is illustrated in **Fig. 7.2-1.** However, in cases where racks are without undercuts, S_{pr} is to be taken as zero.

$$h_F = \frac{m_n}{2}\left[(\cos\gamma_e - \sin\gamma_e\tan\alpha_{Fen})\frac{d_{en}}{m_n} - z_n\cos\left(\frac{\pi}{3}-\theta\right) - \frac{G}{\cos\theta} + \frac{\rho_{fp}}{m_n}\right]$$

$$\alpha_{Fen} = \alpha_{en} - \gamma_e$$

$$\gamma_e = \frac{0.5\pi + 2x\tan\alpha_n}{z_n} + \mathrm{inv}\alpha_n - \mathrm{inv}\alpha_{en}$$

$$\alpha_{en} = \arccos\left(\frac{d_{bn}}{d_{en}}\right)$$

$$d_{en} = 2\frac{z}{|z|}\left\{\left[\sqrt{\left(\frac{d_{an}}{2}\right)^2-\left(\frac{d_{bn}}{2}\right)^2} - \frac{\pi d\cos\beta\cos\alpha_n}{|z|}(\varepsilon_{\alpha n}-1)\right]^2 + \left(\frac{d_{bn}}{2}\right)^2\right\}^{\frac{1}{2}}$$

$$\varepsilon_{\alpha n} = \frac{\varepsilon_\alpha}{\cos^2\beta_b}$$

$$\beta_b = \arccos\sqrt{1-\sin^2\beta\cos^2\alpha_n}$$

$$d_{bn} = d_n\cos\alpha_n$$

$$d_n = m_n z_n$$

$$d_{an} = d_n + d_a - d$$

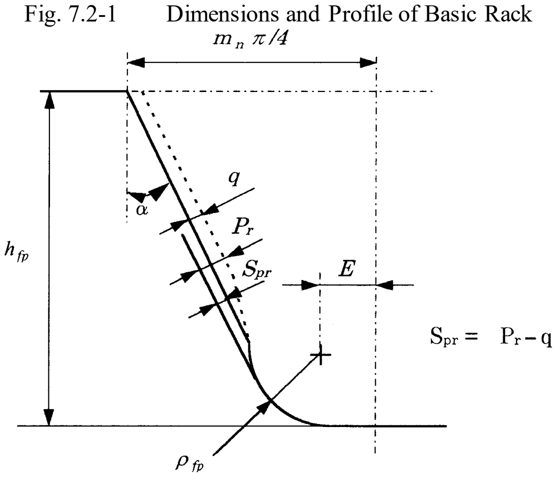
In the case of internal gears, the following coefficients used for determining the form factor are to be calculated as follows:

$$S_{Fn2} = m_n z_n \sin\left(\frac{\pi}{6}-\theta\right) + m_n\left(\frac{G}{\cos\theta}-\frac{\rho_{fp}}{m_n}\right)$$

$$h_{Fn2} = \frac{m_n}{2}\left[(\cos\gamma_e - \sin\gamma_e\tan\alpha_{Fen})\frac{d_{en}}{m_n} - z_n\cos\left(\frac{\pi}{6}-\theta\right) - \sqrt{3}\left(\frac{G}{\cos\theta}-\frac{\rho_{fp2}}{m_n}\right)\right]$$

$$d_{en2} = 2\frac{z}{|z|}\left\{\left[\sqrt{\left(\frac{d_{an}}{2}\right)^2-\left(\frac{d_{bn}}{2}\right)^2} - \frac{\pi d_2\cos\beta\cos\alpha_n}{|z|}(\varepsilon_{\alpha n}-1)\right]^2 + \left(\frac{d_{bn2}}{2}\right)^2\right\}^{\frac{1}{2}}$$

Fig. 7.2-1 Dimensions and Profile of Basic Rack



2 Stress Correction Factor, Y_S

The stress correction factor, Y_S, is used to convert nominal bending stress into local tooth root stress, taking into account that not only bending stresses arise at roots. Y_S applies to those load applications at the outer points of single tooth pair contacts. Y_S is to be determined separately for pinions and for wheels. The value for Y_S is to be determined as follows (in the effective range: $1 \leq q_S < 8$).

$$Y_s = \left(1.2 + 0.13\frac{S_{Fn}}{h_F}\right)q_s^L$$

$$L = \frac{1}{1.21 + 2.3\frac{h_F}{S_{Fn}}}$$

where

However, since q_S is a notch parameter, it is to be calculated as follows:

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

However, since ρ_F is the root fillet radius in the critical section (mm), it is to be calculated as follows:

$$\rho_F = \rho_{fp} + \frac{2m_nG^2}{\cos\theta\,(z_n\cos^2\theta - 2G)}$$

For the calculation of ρ_F, the procedure outlined in the reference standard *ISO* 6336-3:2019 is to be used.

3 Helix Angle Factor for Bending Stress, Y_β

The helix angle factor for bending stress, Y_β, is used to convert the stress calculated for point loaded cantilever beams representing gear teeth to the stress induced by loads along oblique load lines into cantilever plates which represent helical gear teeth. The value for Y_β can be calculated as follows:

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

where β is the reference helix angle in degree.

However, in cases where $\beta > 30°$ then $\beta = 30°$; and,

Also, in cases where $\varepsilon_\beta > 1$ then ε_β is to be taken as 1.0.

4 Rim thickness factor, Y_B

The rim thickness factor, Y_B, is a simplified factor used to de-rate thin rimmed gears. For critically loaded applications, this method is to be replaced by a more comprehensive analysis. Factor Y_B is to be determined as follows:

(1) For external gears:

$$Y_B = 1.6 \cdot \ln\left(2.242\frac{h}{s_R}\right) \quad (0.5 < s_R/h < 1.2)$$

$$= 1 \quad (s_R/h \geq 1.2)$$

s_R : rim thickness of external gears (mm)

h : tooth height (mm)

The case $s_R/h \leq 0.5$ is to be avoided.

(2) For internal gears:

$$Y_B = 1.15 \cdot \ln\left(8.324\frac{m_n}{s_R}\right) \qquad (1.75 < s_R/m_n < 3.5)$$

$$= 1 \qquad (s_R/m_n \geq 3.5)$$

s_R : rim thickness of internal gears (mm)

The case $s_R/m_n \leq 1.75$ is to be avoided.

5 Deep tooth factor, Y_{DT}

The deep tooth factor, Y_{DT}, adjusts the tooth root stress to take into account high precision gears and contact ratios within the range of virtual contact ratio $2.05 \leq \varepsilon_{\alpha n} \leq 2.5$.

$$\varepsilon_{\alpha n} = \frac{\varepsilon_\alpha}{\cos^2\beta_b}$$

Factor Y_{DT} is to be determined as follows:

$Y_{DT} = 0.7$(ISO accuracy grade ≤ 4 and $\varepsilon_{\alpha n} > 2.5$)

$Y_{DT} = 2.366 - 0.666\,\varepsilon_{\alpha n}$($ISO$ accuracy grade ≤ 4 and $2.05 < \varepsilon_{\alpha n} \leq 2.5$)

$Y_{DT} = 1.0$(In all other cases)

1.7.3 Permissible Tooth Root Bending Stress, σ_{FP}

1 Permissible tooth root bending stress σ_{FP} is to be calculated as follows:

$$\sigma_{FP} = \frac{\sigma_{FE} Y_d Y_N Y_{\delta relT} Y_{RrelT} Y_X}{S_F}$$

2 Bending Endurance Limit, σ_{FE}

For a given material, σ_{FE} is the local tooth root stress which can be permanently endured. According to the reference standard ISO 6336-5:2016, the number of 3×10^6 cycles may be regarded as the beginning of the endurance limit. σ_{FE} is defined as unidirectional pulsating stress with a minimum stress zero (neglecting any residual stresses due to heat treatment). Other conditions such as alternating stress or prestressing etc. are covered by the design factor Y_d. σ_{FE} values are to correspond to a failure probability of 1 % or less. These endurance limits mainly depends on the following:

(1) Material composition, cleanliness and defects;

(2) Mechanical properties;

(3) Residual stresses;

(4) Hardening process, depth of hardened zone, hardness gradient

(5) Material structure (forged, rolled bar, cast)

The value for σ_{FE} is to be calculated as follows:

$$\sigma_{FE} = 2\sigma_{Flim}$$

Values for $\sigma_{F\mathrm{lim}}$ are given in **Table 7.3-1.** However, for materials having enough data showing their higher endurance limit, values larger than those given in the table may be allowed by the Society in consideration of the factors **(1)** through **(5)** mentioned above.

3 Design Factor, Y_d

The design factor, Y_d, takes into account the influence of load reversing and shrink fit prestressing on tooth root strength, relative to tooth root strength with unidirectional loads as defined for σ_{FE}. Y_d for load reversing is to be determined as follows:

$Y_d = 1.00$ (In general cases)

$= 0.90$ (In the case of gears with occasional part loads in reversed directions, such as the main gears in reversing gearboxes)

$= 0.70$ (In the case of idler gears)

4 Life Factor for Bending Stress, Y_N

The life factor for bending stress, Y_N, accounts for the higher tooth root bending stress permissible in cases where limited life is required. Values greater than 1.0 will be considered by the Society on a case-by-case basis.

This factor mainly depends on the following:

(1) Material and heat treatment;

(2) Number of load cycles (service life);

(3) Influence factors($Y_{\delta relT}$, Y_{RrelT}, Y_X).

Y_N is to be determined according to Method B outlined in the reference standard *ISO* 6336-3:2019.

5 Relative Notch Sensitivity Factor, $Y_{\delta relT}$

The relative notch sensitivity factor, $Y_{\delta relT}$, indicates the extent of the influence of concentrated stress on fatigue endurance limits. This factor mainly depends on materials and relative stress gradients. This factor is to be calculated as follows:

$$Y_{\delta relT} = \frac{1+\sqrt{0.2\rho'(1+2q_s)}}{1+\sqrt{1.2\rho'}}$$

q_s : notch parameter

ρ' : slip-layer thickness (mm)

However, the values to be used for ρ' are those given in **Table 7.3-2**.

6 Relative Surface Factor, Y_{RrelT}

The relative surface factor, Y_{RrelT}, takes into account the influence of surface conditions in tooth root fillets on root strength and mainly depends on peak to valley surface roughness. The value for Y_{RrelT} is to be determined as shown in **Table 7.3-3**.

7 Size Factor for Bending Stress, Y_X

The size factor for bending stress Y_X takes into account decreases of strength with increasing size. This factor mainly depends on:

(1) Material and heat treatment;

(2) Tooth and gear dimensions;

(3) Ratio of case depth to tooth size.

The value for Y_X is to be determined as shown in **Table 7.3-4**.

8 Safety Factor for Tooth Root Bending Stress, S_F

The safety factor for tooth root bending stress, S_F, is to be taken as follows:

(1) 1.55 for main propulsion gears;

(2) 1.40 for auxiliary gears;

In addition, in cases where the gears of duplicated independent propulsion or auxiliary machinery which are duplicated beyond that required for their respective classes, a reduced value may be used at the discretion of the Society.

Table 7.3-1 Values of σ_{Flim} (N/mm^2)

Steel type	σ_{Flim}
Normalized structural steels	0.45*HB*+70
Through hardening carbon steels	0.25*HB*+160
Through hardening alloy steels	0.45*HB*+180
Induction hardened alloys	0.14*HV*+285; however, 365 for *HV*>570
Nitrided alloys	365
Soft nitrided alloys	0.66*HV*+88; however, 385 for *HV*>450
Nitrided steels	420
Carburized hardened alloys	465; however, 500 for *HRC*>30 in core

HB: Brinell Hardness; *HV*: Vickers Hardness; *HRC*: C Scale Rockwell Hardness

Table 7.3-2 Values of ρ' (mm)

Materials	ρ'
Nitriding steels, surface or through hardened	0.1005
Steels having yield strength about 300 N/mm^2	0.0833
Steels having yield strength about 400 N/mm^2	0.0445
Through hardened steels having yield strength about 500 N/mm^2	0.0281
Through hardened steels having yield strength 600 N/mm^2	0.0194
Through hardened steels having $\sigma_{0.2}$ about 800 N/mm^2	0.0064
Through hardened steels having $\sigma_{0.2}$ about 1000 N/mm^2	0.0014
Surface hardening steels, surface hardened	0.0030

Table 7.3-3 Values of Relative Surface Factor, Y_{RrelT}

Materials	Y_{RrelT}
Case hardened steels, Through hardened steels ($\sigma_B \geq 800$ N/mm^2)	1.120 (for R_Z <1) $1.674 - 0.529(R_Z + 1)^{0.1}$ (for 1≤ R_Z ≤ 40)
Normalized steels ($\sigma_B < 800$ N/mm^2)	1.070 (for R_Z <1) $5.306 - 4.203(R_Z + 1)^{0.01}$ (for 1≤ R_Z ≤40)
Nitriding steels	1.025 (for R_Z <1) $4.299 - 3.259(R_Z + 1)^{0.0058}$(for1≤ R_Z ≤40)

Notes

1: R_Z mean peak to valley roughness of tooth root fillets (μm).

2: This method is only valid when scratches or similar defects deeper than $2R_Z$ do not exist.

3: If the roughness stated is an arithmetic mean roughness, i.e. R_a value, the approximation $R_Z = 6R_a$ may be applied.

Table 7.3-4 Size Factor for Bending Stress, Y_X

m_n	Materials	Y_X
$m_n \leq 5$	General	1.00
$5 < m_n < 30$	Normalized and through hardened steels	$1.03 - 0.006m_n$
$m_n \geq 30$		0.85
$5 < m_n < 25$	Surface hardened steels	$1.05 - 0.010m_n$
$m_n \geq 25$		0.80

Annex 6.2.2 USE OF HIGH-STRENGTH MATERIALS FOR INTERMEDIATE SHAFTS

1.1 Application

This annex applies to low alloy steel forgings (excluding stainless steel forgings, etc.) which have specified tensile strengths greater than 800 N/mm^2, but less than 950 N/mm^2 and which are intended for use as intermediate shaft material.

1.2 Torsional Fatigue Test

1.2.1 General Requirements

A torsional fatigue test is to be performed to verify that the material exhibits similar fatigue life as conventional steels. The torsional fatigue strength of said material is to be equal to or greater than the allowable limit of the torsional vibration stresses τ_1 given by the formulae in **8.2.2-1(1), Part D of the Rules**. The test is to be carried out with notched and unnotched specimens respectively. For calculation of the stress concentration factor of the notched specimen, the notch factor is to be evaluated in consideration of the severest torsional stress concentration in the design criteria.

1.2.2 Test Conditions

Test conditions are to be in accordance with **Table 1.1**. Mean surface roughness is to be less than 0.2 μm for R_a and the absence of localised machining marks is to be verified by visual examination at low magnification (x20) as required by *Section* 8.4 of *ISO* 1352:2011. Test procedures are to be in accordance with *Section* 10 of *ISO* 1352:2011.

Table 1.1 Test conditions

Loading type	Torsion
Stress ratio	R = −1
Load waveform	Constant-amplitude sinusoidal
Evaluation	S-N curve
Number of cycles for test termination	1 x 10^7 cycles

1.2.3 Acceptance criteria

Measured high-cycle torsional fatigue strength τ_{C1} and low-cycle torsional fatigue strength τ_{C2} are to be equal to or greater than the values given by the following formulae:

$$\tau_{C1} \geq \tau_{1,\lambda=0} = \frac{\sigma_B + 160}{6} \cdot C_K \cdot C_D$$

$$\tau_{C2} \geq 1.7\tau_{C1}/\sqrt{C_K}$$

where

C_K: Coefficient related to the type and shape of the shaft. To be determined using the formulae (modified as needed) specified in Note (1) of **Table D8.1, Part D of the Rules**. However, the stress concentration factor for computing C_K can be determined in consideration of actual design conditions. For unnotched specimens, the stress concentration factor is 1.0.

C_D: Coefficient related to shaft size. To be determined using the formula (modified as needed) specified **8.2.2-1(1), Part D of the Rules**.

σ_B: Specified tensile strength of the shaft material (N/mm^2)

1.3 Cleanliness Requirements

Low alloy steel forgings are to have a degree of cleanliness shown in **Table 1.2** when tested according to *ISO* 4967:2013 *method A*. Representative samples are to be obtained from each heat of forged or rolled products. In addition, the forgings are also to comply with the minimum requirements of **Table K6.2, Part K of the Rules**, with particular attention given to minimising the concentrations of sulphur, phosphorus and oxygen in order to achieve the cleanliness requirements. The specific steel composition is required to be approved by the Society.

Table 1.2 Cleanliness requirements

Inclusion group	Series	Limiting chart diagram index I
Type A	Fine	1
	Thick	1
Type B	Fine	1.5
	Thick	1
Type C	Fine	1
	Thick	1
Type D	Fine	1
	Thick	1
Type DS	-	1

1.4 Inspection

Low alloy steel forging are to be subjected to the ultrasonic testing specified in **6.1.10-1(1), Part K of the Rules**.

Annex 6.2.13 CALCULATION OF SHAFT ALIGNMENT

1.1 General

1.1.1 Application

1 This annex applies to the shaft alignment calculations required by **6.2.10**, **6.2.11** and **6.2.13, Part D of the Rules**. With regard to the paragraphs in **1.3** of this annex, application is to be in accordance with **Table 1.1.1-1**.

Table 1.1.1-1 Application of conditions for calculation etc.

Type of main propulsion machinery	Paragraphs1)2)		
	1.3.1	**1.3.2**	**1.3.3** 3)
Two-stroke cycle engines	●	●	●
Four-stroke cycle engines	●	●	-
Steam turbines	●	●	-

Notes:

1) ●: Applicable -: Not applicable

2) **1.3.1**: Light draught condition (cold condition)
1.3.2: Light draught condition (hot condition)
1.3.3: Full draught condition (hot condition)

3) Only applicable to oil tankers, ships carrying dangerous chemicals in bulk, bulk carriers and general dry cargo ships, in cases where:

- Oil tankers are those ships defined in **1.3.1(11)**, **Part B of the Rules**;
- Ships carrying dangerous chemicals in bulk are those ships defined in **2.1.43**, **Part A of the Rules**;
- Bulk carriers are those ships defined in **1.3.1(13)**, **Part B of the Rules**; and
- General dry cargo ships are those ships defined in **1.3.1(15)**, **Part B of the Rules**.

2 Notwithstanding **-1** above, **1.1.2**, **1.2.1** and **1.3.1** (excluding **1.3.1-4**) below are to apply to those shaft alignment calculations required by **6.2.10** and **6.2.11, Part D of the Rules** in cases where main propulsion shafting is comprised of oil-lubricated propeller shafts with diameters less than 400 *mm*.

3 Alternative methods of calculation different from those described in this annex may be employed subject to the prior approval of the Society.

1.1.2 Calculation Sheets for Shaft Alignment

Calculation sheets for shaft alignment that include the following data are to be submitted for approval:

(1) Diameters (outer and inner) and lengths of shafts
(2) Length of bearings
(3) Concentrated loads and loading points
(4) Support points
(5) Bearing offsets from reference lines
(6) Reaction influence numbers
(7) Bending moments and bending stresses
(8) Bearing loads and nominal bearing pressures
(9) Relative inclination of propeller shafts and aftmost stern tube bearings or the maximum bearing pressure in aftmost stern tube bearings
(10) Deflection curves for any shafting
(11) Sags and gaps between shaft coupling flanges

(12) Procedures for measuring bearing loads (in cases where such measurements are required)

1.2 Models of Shafting

1.2.1 Loads

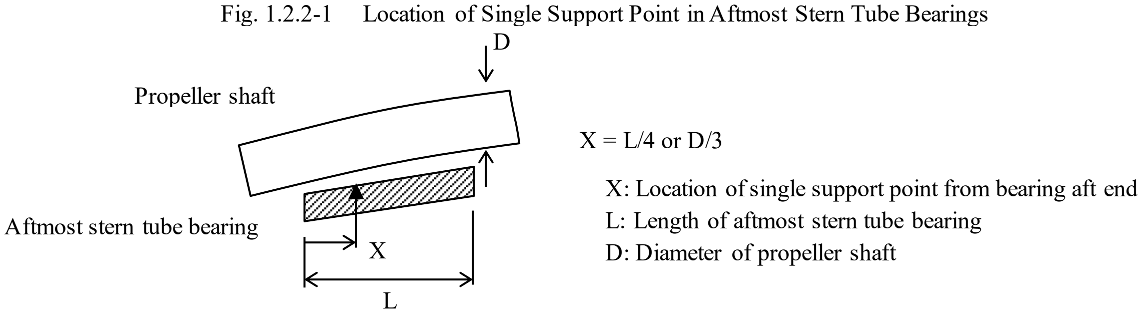
1 Static loads are to be used in shaft alignment calculations.

2 Any buoyancy forces working on shafting are to be considered as loads. Tensile forces due to cam shaft drive chains specified by engine manufacturers are also to be considered as loads for engines.

1.2.2 Bearings

1 In cases where only one support point is assumed in aftmost stern tube bearings, its location is to be at L/4 or D/3 from the aft end of such bearings. In cases where two support points are assumed, their locations are to be at each end of those aftmost stern tube bearings. In cases where three or more support points are assumed, their locations may be decided by the designer. The location of support points in each bearing, other than those aftmost stern tube bearings, is to be at the centre of such bearings.

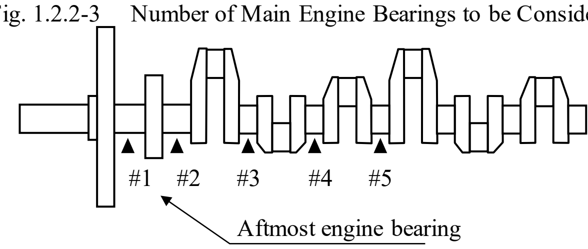
Fig. 1.2.2-1 Location of Single Support Point in Aftmost Stern Tube Bearings



2 Either rigid supports or elastic supports may be acceptable as the type of supports used.

3 In cases where thrust shafts are integrated with crankshafts, not less than five main bearings of such engines are to be considered in shaft alignment calculations.

Fig. 1.2.2-3 Number of Main Engine Bearings to be Considered



1.2.3 Equivalent Diameter of Crankshafts

When evaluating the shafting of two-stroke cycle engines used as main propulsion machinery, the equivalent diameters of crankshafts specified by engine manufacturers are to be used in shaft alignment calculations in order to give due consideration to any lesser bending stiffness that exists in actual crankshafts compared with simply using those diameters of crank journals in models.

1.2.4 Shafting with Reduction Gears

In the case of shafting with reduction gears such as those found in main steam turbines or geared reciprocating internal combustion engines, shafting from propellers to wheel gears is to be considered in shaft alignment calculations.

1.3 Load Condition and Evaluation of Calculation Results

1.3.1 Light Draught Condition (Cold Condition)

1 Shaft alignment calculations are to be performed under the assumption that ships are in light draught conditions and main propulsion machinery are in cold conditions. In cases where shafts are coupled before launching, shaft alignment calculations are to be performed for such coupled conditions instead of for light draught conditions without taking any buoyancy forces on propellers into account.

2 In cases where aftmost stern tube bearings consist of oil-lubricated white metal, evaluations are to be made of nominal bearing pressure together with either the relative inclination between propeller shafts and aftmost stern tube bearings or the maximum bearing pressures in such aftmost stern tube bearings, either of which is to be determined in order to prevent any edge loading on bearings. Calculated values are to be within those allowable limits shown in **Table 1.3.1-2**.

Table 1.3.1-2 Allowable Limits for Aftmost Stern Tube Bearings (Oil-Lubricated White Metal)

	Allowable Limit	Notes
Nominal bearing pressure	0.8 *MPa*	
Relative inclination between the propeller shaft and the aftmost stern tube bearing	3×10^{-4} *rad*	Applicable in cases where the number of support points is one or two. In the case of two support points, the relative inclination is to be calculated at each end of bearings. (see **Fig. 1.3.1-2(a)**)
Maximum bearing pressure	40 *MPa*	Applicable in cases where the maximum bearing pressure is calculated. (see **Fig. 1.3.1-2(b)**)

Fig. 1.3.1-2(a) Relative Inclination

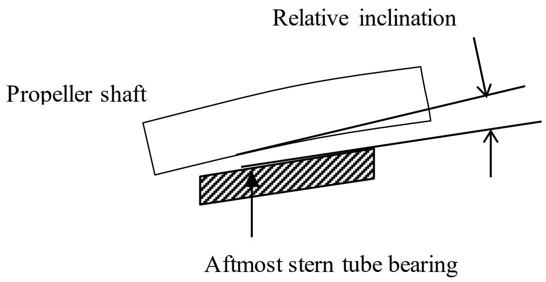
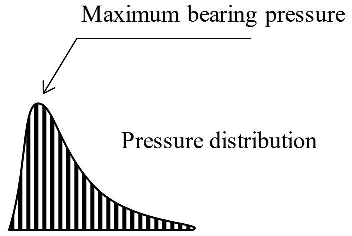


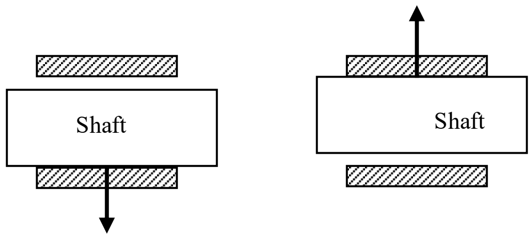
Fig. 1.3.1-2(b) Maximum Bearing Pressure



3 Bending moments (absolute values) calculated for any bearing are not to be more than the value determined for the aftmost stern tube bearings.

4 In principle, bearing loads calculated at each bearing are to be positive values. However, in the case of aftmost bearings of two-stroke cycle engines used as main propulsion machinery, bearing loads of zero may be accepted as zero (negative values are not acceptable.) subject to the agreement of the engine manufacturer. Directions of bearing loads are shown in **Fig. 1.3.1-4**.

Fig. 1.3.1-4 Direction of Bearing Loads



1.3.2 Light Draught Condition (Hot Condition)

1 Shaft alignment calculations are to be performed under the assumption that ships are in light draught conditions and reciprocating internal combustion engines used as main propulsion machinery is in hot conditions. In such cases, any increases in offset specified by manufacturers for engine bearings and those bearings in reduction gears are to be considered under hot conditions.

2 In the calculations **-1** above, full immersion condition of propellers may also be taken into account in such calculations.

3 In cases where shafts are coupled before launching, the calculations in **-1** above are to be performed under the assumption that there is no change in bearing offsets from reference lines between those conditions before and after launching.

4 Bearing loads calculated at each bearing are to be positive values.

5 In the case of shafting with reduction gears, any differences in bearing loads between the fore and aft bearings of wheel gears in hot conditions are to be within those allowable limits specified by manufacturers.

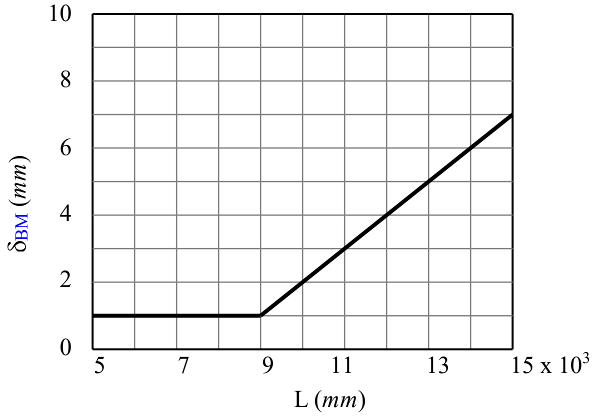
6 Bending moments due to propeller eccentric thrusts may be taken into account in such calculations.

1.3.3 Full Draught Condition (Hot Condition)

1 Shaft alignment for oil tankers, ships carrying dangerous chemicals in bulk, bulk carriers, and general dry cargo ships is to be designed so as to satisfy the following criteria in order that all engine bearings are fairly evenly loaded, even under any hull deflection that occurs in cases where ships are in full draught conditions. The extent of any relative displacement due to differences between hull deflection that occurs in light draught conditions and hull deflection that occurs in full draught conditions which result in second or third aftmost engine bearings becoming unloaded, as measured at aftmost bulkheads of engine rooms (calculated as δ_{B2} and δ_{B3}, respectively), is to be greater than those allowable lower limits δ_{BM} shown in **Fig. 1.3.3-1 (a)**.

The relative displacements δ_{B2} and δ_{B3} given above are to be calculated using the formulae in **(1)** or **(2)** below, which are used to calculate the reaction influence numbers in alignment calculations, depending on the type of bearing supports adopted (elastic or rigid supports).

Fig. 1.3.3-1(a) Allowable Lower Limit δ_{BM} for δ_{B2} and δ_{B3}.



Distance from support points of aftmost engine bearings to aftmost bulkheads of engine rooms (see Fig. 1.3.3-1(b))

(1) In the case of elastic supports, δ_{B2} and δ_{B3} can be obtained with i = 2 or 3, respectively as follows:

$$\delta_{Bi} = -R_i/S_i$$

where

i : Engine bearing numbers as counted from the aft of engines

R_i : Reaction forces at the i-th number engine bearing as determined by those calculations in 1.3.2 (*kN*)

S_i : Influence numbers for the i-th number engine bearing in cases where hull deflection at aftmost bulkheads of engine rooms becomes -1 *mm*; obtained from the following equation (*kN/mm*):

$$S_i = \sum_{n=1}^{a-1} C_{b+i-1,n}(1.5x_n - 0.5) + \sum_{n=a}^{b-1} C_{b+i-1,n}{x_n}^{1.5}$$

where

$$x_n = X_n/L$$

n : Support point numbers of shafting (counted from the aft of such shafting)

a : Number of nearest support points forward of aftmost bulkheads of engine rooms (counted from the aft of such shafting)

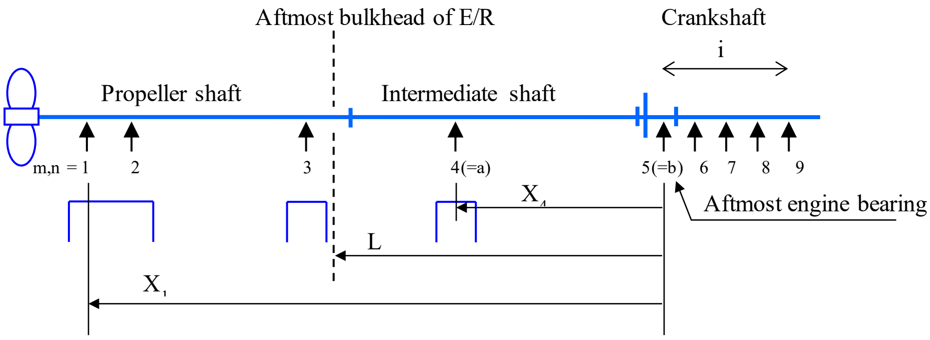
b : Support point numbers of aftmost engine bearings (counted from the aft of such shafting)

X_n : Distance from support point b to support point n (*mm*)

L : Distance from the support point b to aftmost bulkheads of engine rooms (*mm*)

$C_{m,n}$: Influence number at support point m in cases where the relative displacement at support point n becomes -1 *mm* (*kN/mm*) (see Fig. 1.3.3-1(b))

Fig. 1.3.3-1(b) Engine Bearing Numbers and Support Point Numbers



(2) In the case of rigid supports, δ_{B2} or δ_{B3} can be obtained by solving the following simultaneous equations **(1)** and **(2)**, respectively as follows:

$$\left.\begin{aligned}
S_1\delta_{B2} + (C_{1,1} - K)\delta_1 + C_{1,3}\delta_3 + C_{1,4}\delta_4 + C_{1,5}\delta_5 &= C_{1,2}R_2/K \\
S_2\delta_{B2} + C_{2,1}\delta_1 + C_{2,3}\delta_3 + C_{2,4}\delta_4 + C_{2,5}\delta_5 &= (C_{2,2} - K)R_2/K \\
S_3\delta_{B2} + C_{3,1}\delta_1 + (C_{3,3} - K)\delta_3 + C_{3,4}\delta_4 + C_{3,5}\delta_5 &= C_{3,2}R_2/K \\
S_4\delta_{B2} + C_{4,1}\delta_1 + C_{4,3}\delta_3 + (C_{4,4} - K)\delta_4 + C_{4,5}\delta_5 &= C_{4,2}R_2/K \\
S_5\delta_{B2} + C_{5,1}\delta_1 + C_{5,3}\delta_3 + C_{5,4}\delta_4 + (C_{5,5} - K)\delta_5 &= C_{5,2}R_2/K
\end{aligned}\right\} \quad (1)$$

$$\left.\begin{aligned}
S_1\delta_{B3} + (C_{1,1} - K)\delta_1 + C_{1,2}\delta_2 + C_{1,4}\delta_4 + C_{1,5}\delta_5 &= C_{1,3}R_3/K \\
S_2\delta_{B3} + C_{2,1}\delta_1 + (C_{2,2} - K)\delta_2 + C_{2,4}\delta_4 + C_{2,5}\delta_5 &= C_{2,3}R_3/K \\
S_3\delta_{B3} + C_{3,1}\delta_1 + C_{3,2}\delta_2 + C_{3,4}\delta_4 + C_{3,5}\delta_5 &= (C_{3,3} - K)R_3/K \\
S_4\delta_{B3} + C_{4,1}\delta_1 + C_{4,2}\delta_2 + (C_{4,4} - K)\delta_4 + C_{4,5}\delta_5 &= C_{4,3}R_3/K \\
S_5\delta_{B3} + C_{5,1}\delta_1 + C_{5,2}\delta_2 + C_{5,4}\delta_4 + (C_{5,5} - K)\delta_5 &= C_{5,3}R_3/K
\end{aligned}\right\} \quad (2)$$

where

K : Stiffness of bearing supports, given as $K = 5000$ (*kN/mm*)

S_i : Influence number for i-th number engine bearing (see **(1)** above)

$C_{i,j}$: Influence number for the i-th number engine bearing in cases where the relative displacement at the j-th number engine bearing becomes -1 *mm* (*kN/mm*) (However, the i-th and j-th numbers are counted from the aft of engines.)

$\delta_i (i = 1,2,3,4,5)$: Elastic relative displacement at each engine bearing resulting from the relative displacement δ_{B2} or δ_{B3}. (δ_i is unknown.)

2 Notwithstanding **-1 above**, the Society may examine and accept alternative criteria, provided that documentation is submitted that makes it possible to evaluate the condition of engine bearings in cases where ships are in full draught conditions.

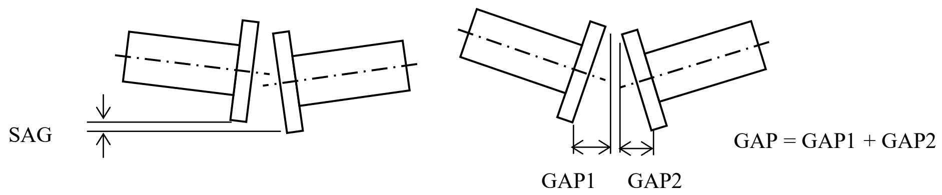
3 Other documents such as those showing results structural analysis evaluating the extent of hull deflection may be required by the Society in cases where stern hull construction is considered to be unconventional.

1.4 Matters Relating to Shaft Alignment Procedures

1.4.1 Sags and Gaps between Shaft Coupling Flanges

Sags and gaps between shaft coupling flanges in an uncoupled condition are to be calculated under the condition that bearing offsets from reference lines are those used in those calculation described in **1.3.1** above.

Fig. 1.4.1 Sag and Gap between Shaft Coupling Flanges



1.4.2 Procedure for Measuring Bearing Loads

In cases where bearing loads are measured using the jack-up technique, documentation describing the measurement procedures followed (including jack-up positions, load correction factors and expected jack-up loads) is to be prepared. The immersion of propellers at the time of such measurements is also to be considered in the bearing loads measured.

Annex 12.1.6 PLASTIC PIPES

1.1 Scope

1 This annex is to apply to the materials, construction, strength, application, assembly and tests of piping systems on ships, including pipe joints and fittings, made predominately of materials other than metal.

2 The use of mechanical joints only approved for the use in metallic piping systems is not permitted.

3 The specification of the pipes is to be in accordance with a recognised national or international standard acceptable to the Society and the following requirements. However, the requirements in **1.4** (except **1.4.1-2(2)**) and **1.5** (except **1.5.2**) need not apply to the pipes specified in **1.3-2**.

1.2 Terminology

Terms used in this annex are defined as follows:

(1) "Plastic" means both thermoplastic and thermosetting plastic materials with or without reinforcement, such as *PVC* and fibre reinforced plastics - FRP. Plastic includes synthetic rubber and materials of similar thermo /mechanical properties.

(2) "Pipe/piping Systems" means those made of plastic(s) and include pipes, fittings, system joints, methods of joining and any internal or external liners, coverings and coatings required to comply with this annex.

(3) "Joint" means the location at which two pieces of pipe or a pipe and a fitting are connected together. The joint may be made by adhesive bonding, laminating, welding, flanges and mechanical joints according to **Fig. D12.1** in **12.3.3, Part D of the Rules**.

(4) "Fittings" means bends, elbows, fabricated branch pieces, etc. of plastic materials.

(5) "Nominal pressure" means the maximum permissible working pressure which is to be determined in accordance with **1.4.1-2**.

(6) "Design pressure" means the maximum working pressure which is expected under operation conditions or the highest set pressure of any safety valve or pressure relief device on the system, if fitted.

(7) "Fire endurance" means the capability of piping to maintain its strength and integrity (i.e. capable of performing its intended function) for some predetermined period of time while exposed to fire.

(8) "Pipes and their associated fittings whose integrity is essential to the safety of ship" means piping systems specified in **Table 1** and includes piping systems deemed by the Society that in event of failure will pose a threat to personnel and the ship.

(9) "Essential services" are those services essential for propulsion and steering and safety of the ship as specified in **3.2.1-2, Part H of the Rules**.

(10) "*FTP* Code" means as defined in **3.2.23, Part R of the Rules**.

1.3 Materials

1 Plastic pipes are to be those approved by the Society in accordance with **12.1.6, Part D of the Rules** and adequate for their service conditions.

2 Notwithstanding the requirement in **-1 above**, pipes which comply with recognised standards such as *JIS* or *JWWA*, comply with **1.4.1-2(2)** and **1.5.2** and are adequate for their service conditions may be used for the following **(1)** and **(2)**:

(1) Drinking water pipes, domestic water pipes (including hot water pipes) and sanitary pipes located within accommodation spaces and engine rooms as well as deck scuppers located within spaces.

(2) Other pipes not used for essential services.

1.4 Design Requirements

1.4.1 Strength

1 The strength of fittings and joints is to be not less than that of the pipes.

2 The nominal pressure is to be determined from the following **(1)** to **(3)**:

(1) Internal Pressure

In the case of internal pressure, the smaller of the following is to be taken:

$$P_{nint} \le \frac{P_{sth}}{4} \text{ or } P_{nint} \le \frac{P_{lth}}{2.5}$$

where

P_{sth} : Short-term hydrostatic test failure pressure

P_{lth} : Long-term hydrostatic test failure pessure (>100,000h)

(2) External Pressure (for any installation which may be subject to vacuum conditions inside the pipe or a head of liquid acting on the outside of the pipe or for any pipes that would allow progressive flooding to other compartments through damaged piping or through open ended pipes in the compartments)

External pressure is to comply with the following formula. Maximum working external pressure is the sum of the vacuum inside pipes and heads of liquid acting on the outside of pipes.

$$P_{next} \le \frac{P_{col}}{3}$$

where

P_{col} : Pipe collapse pressure (However, in no instance is the pipe collapse pressure to be less than 0.3 *MPa*)

(3) Wall Thickness

Notwithstanding the requirements of **(1)** or **(2)** above as applicable, the pipe or pipe layer minimum wall thickness is to follow recognised standards. In the absence of standards for pipes not subject to external pressure, the requirements of **(2)** above are to be met.

(4) Temperature

Nominal pressure is to be specified in accordance with the manufacturer's recommendations with due regard being given to the maximum possible working temperature.

3 Design temperature

(1) In this annex, design temperatures are to be the highest and lowest working temperatures of any liquid inside such pipes and atmospheric temperatures of the area where such pipes are arranged at the designed conditions. The design temperatures of ballast pipes are not to be less than 50 ºC for high temperature sides and are not to be more than 0 ºC for low temperature sides.

(2) The permissible working temperature depending on the working pressure is to be in accordance with manufacturer's recommendations, but in each case it is to be at least 20 ºC lower than the minimum heat distortion/deflection temperature of the pipe material, determined according to *ISO* 75-2:2013 method A, or equivalent e.g. *ASTM D648-18*. The minimum heat distortion/deflection temperature is to be not less than 80 ºC.

4 The sum of the longitudinal stresses due to pressure, weight and other loads is not to exceed the allowable stress in the longitudinal direction.

5 In the case of fibre reinforced plastic pipes, the sum of the longitudinal stresses is not to exceed half of the nominal circumferential stress derived from the nominal internal pressure condition according to **-2(1)** above.

6 Plastic pipes and joints are to have a minimum resistance to impact in accordance with recognised national or international standards, e.g. *ISO* 9854, *ISO* 9653, *ISO* 15493, *ASTM D2444* or their equivalent.

1.5 Requirements for Pipe/Piping Systems Depending On Service and/or Locations

1.5.1 Fire Endurance

1 Pipes and their associated joints and fittings whose integrity is essential to the safety of ships are required to meet the minimum fire endurance requirements of *Appendix* 1 or 2, as applicable, of *IMO Res. A.*753(18) (including any amendments due to *IMO Res. MSC.*313(88) and *IMO Res. MSC.*399(95)).

2 Unless instructed otherwise by the Administration, fire endurance tests are to be carried out with representative specimens for pipes, joints and fittings in accordance with the following **(1)** and **(2)**. A test specimen incorporating several components of a piping system may be tested in a single test:

(1) Pipes

(a) For sizes with outer diameter less than 200 *mm*, the pipe with the minimum outer diameter and wall thickness is used.

(b) For sizes with outer diameter of 200 *mm* or more, the pipe with the minimum outer diameter is used for each category of t/D ratio (where "D" is the outer diameter and "t" is the structural wall thickness). A scattering of ±10 % for t/D is regarded as the same group.

If fire protective coatings, etc. are included in the pipe used in the fire test, the pipe with the minimum fire protective coatings, etc. are used regardless of the (t/D) ratio.

(2) Joints

Each type of joint applicable for applied fire endurance level on pipe to pipe specimen is to be tested.

3 Means are to be provided to ensure a constant media pressure inside the test specimen during the fire test as specified in *Appendix* 1 or 2 of *IMO Res.A.*753(18), as amended by *IMO Res. MSC.*313(88) and *MSC.*399(95). During the test, it is not permitted to replace media drained by fresh water or nitrogen.

4 Permitted use of piping depending on fire endurance, location and piping system is given in **Table 1**.

1.5.2 Flame Spread

1 All pipes, except those fitted on open decks and within tanks, cofferdams, pipe tunnels and ducts, if separated from accommodation, permanent manned areas and escape ways by means of an "*A*" class bulkhead, are to have low surface flame spread characteristics as determined by the test procedures given in *Appendix* 3 of *IMO Res. A.*753(18) (including any amendments due to *IMO Res. MSC.*313(88) and *IMO Res. MSC.*399(95)). Piping with both the total heat release (Q_t) of not more than 0.2 *MJ* and the peak heat release rate (Q_p) of not more than 1.0*kW* (both values determined in accordance with the requirements of "Test for Surface Flammability" specified in the *FTP* Code, *ANNEX* 1, *Part* 5) are considered to comply with the above requirements and may be exempted from testing in accordance to standard *ISO* 1716:2010 related to calorific value.

2 Surface flame spread characteristics are to be determined using the procedure given in the *FTP* Code, *ANNEX* 1, *Part* 5 with regard to the modifications due to the curvilinear pipe surfaces as also listed in *Appendix* 3 of *IMO Res. A.*753(18), as amended by *IMO Res. MSC.*313(88) and *IMO Res. MSC.* 399(95).

3 Surface flame spread characteristics may also be determined using the test procedures given in *ASTM D635-18*, or in other national equivalent standards. Under the procedure of *ASTM D635-18*, a maximum burning rate of 60 *mm/min* applies. In case of adoption of other national equivalent standards, the relevant acceptance criteria are to be defined.

1.5.3 Fire Protection Coatings

In cases where the fire protective coating of pipes and fittings is necessary for achieving required fire endurance levels, such coating is to meet the requirements in the following **(1)** to **(4)**:

(1) Pipes are generally to be delivered from the manufacturer with the protective coating already applied.

(2) The fire protection properties of such coatings are not to be diminished when exposed to salt water, oil or bilge slops. It is to be demonstrated that such coatings are resistant to those products that are likely to come into contact with the piping.

(3) When considering fire protection coatings, characteristics such as thermal expansion, resistance against vibrations, and elasticity are to be taken into account.

(4) Fire protection coatings are to have sufficient resistance to impact and be able to retain their integrity.

1.5.4 Electrical Conductivity

1 In cases where the piping systems for fluids with conductivity of less than 1,000 *pS/m* (*pico siemens per metre*), such as refined products and distillates, conductive pipes are to be used.

2 Regardless of the fluid being conveyed, plastic piping is to be electrically conductive if such piping passes through the hazardous areas specified in **4.3, Part H of the Rules**.

3 Pipes and fittings having conductive layers are to be protected against any possibility of spark damage to pipe walls.

4 In cases where electrical conductivity is to be ensured, the resistance of pipes and fittings is not to exceed 0.1 $M\Omega/m$.

1.5.5 Durability against Chemicals

Pipes are to be resistant to any chemical substances they might possibly come it to contact with during service.

1.5.6 Smoke Generation and Toxicity

Piping materials within the accommodation, service, and control spaces are to fulfill the requirements of *Appendix* 3 of *IMO Res. A.*753(18) (including any amendments due to *IMO Res. MSC.*313(88) and *IMO Res. MSC.*399(95)), on smoke and toxicity tests. Procedure modifications are necessary due to the curvilinear pipe surfaces listed in **Chapter 6, Part R of the Rules**.

1.6 Installation

1.6.1 Supports

1 Selection and spacing of pipe supports in shipboard systems are to be determined as a function of allowable stresses and maximum deflection criteria. Support spacing is not to be greater than that recommended the pipe manufacturer. The selection and spacing of pipe supports are to take into account pipe dimensions, length of the piping, mechanical and physical properties of pipe materials, mass of pipes and contained fluids, external pressures, operating temperatures, thermal expansion effects, loads due to external forces, thrust forces, water hammers, vibrations, fatigue and maximum accelerations to which such systems may be subjected. Combination of loads is to be considered.

2 Each support is to evenly distribute the load of the pipe and its contents over the full width of the support. Measures are to be taken to minimise any wearing down of such pipes in the places where they come in contact with their supports.

3 Heavy components in piping systems, such as valves and expansion joints, are to be independently supported.

1.6.2 Expansion

1 Suitable provisions are to be made in pipelines to allow for relative movement between pipes made of plastic and steel structures, paying due regard to:

(1) The difference in the coefficients of thermal expansion.

(2) Deformations of the ship's hull and its structure.

2 When calculating the thermal expansions, system working temperatures as well as those temperatures at which assembly is performed are to be taken into account.

1.6.3 External Loads

1 When installing piping, allowances are to be made for temporary point loads in cases where applicable. Such allowances are to include at least the force exerted by a load (person) of 100 *kg* at mid-span on any pipe of more than 100 *mm* nominal outside diameter.

2 Besides for providing adequate robustness for all piping including open-ended piping a minimum wall thickness, complying with **1.4.1-2**, may be increased taking into account the conditions encountered during service on board ships.

3 Pipes are to be protected from mechanical damage in cases where necessary.

1.6.4 Strength of Connections

1 The strength of connections is to be not less than that of the piping system in which they are installed.

2 Pipes may be assembled using adhesive-bonded, welded, flanged or other joints.

3 Adhesives, when used for joint assembly, are to be suitable for providing permanent seals between pipes and fittings throughout the temperature and pressure ranges of their intended application.

4 All tightening of joints is to be performed in accordance with manufacturer instructions.

1.6.5 Installation of Conductive Pipes

1 In cases where pipes are required to be electrically conductive (as specified in **1.5.4)**, sufficient consideration is to be given to electrical continuity.

2 Any resistance to earth from any points in such piping systems is not to exceed 1 $M\Omega$.

3 Earthing wires are to be accessible for inspection.

1.6.6 Application of Fire Protection Coatings

1 Fire protection coatings are to be applied on joints, in cases where such coatings are necessary for meeting the required fire endurance in accordance with **1.5.3**, after performing hydrostatic pressure tests of such piping systems.

2 Such fire protection coatings are to be applied in accordance with manufacturer recommendations, using procedures approved for each particular case.

3 Pipes are to be electrically conductive, even after being coated with fire protective coatings, in cases where it is necessary to coat conductive pipes.

1.6.7 Penetration of Divisions

1 Where plastic pipes pass through "*A*" or "*B*" class divisions, arrangements are to be made to ensure that their fire endurance is not impaired in accordance with **9.3, Part R of the Rules**.

2 When plastic pipes pass through oiltight and watertight bulkheads or decks, the watertight or oiltight integrity of the bulkhead or deck is to be maintained, and such penetrations are to be of steel. Steel penetration may also be required for other steel divisions in cases where deemed necessary. For pipes not able to satisfy the requirements in **1.4.1-2(2)**, a metallic shut-off valve operable from above the freeboard deck is to be fitted at the bulkhead or deck.

3 If bulkheads or decks are also fire divisions and destruction by fire of any plastic pipes may cause the inflow of liquid from tanks, metallic shut-off valves operable from above freeboard decks should be fitted at such bulkheads or decks.

1.6.8 Control During Installation

1 Pipes are to be properly protected from any damage caused by sparks from things such as welding and cutting as well as from any mechanical impact with heavy objects during assembling.

2 Installation is to be in accordance with manufacturer guidelines.

3 Sufficient consideration is to be given to fire protection and safety of life in cases where adhesives are being used as well as in cases of cutting or grinding pipes.

4 Methods for connecting pipes are to be approved by the Society before such work is started.

5 The tests and explanations specified in this annex are to be completed before shipboard piping installation commences.

6 All personnel involved in either connecting or bonding plastic pipes by welding, lamination or similar methods are to be properly qualified. Records for each person, including the bonding procedure with dates as well as the results of any qualification testing are to be shown to the Surveyor if necessary.

1.6.9 Bonding Procedure Quality Testing

1 Procedures for making bonds are to include:

(1) materials used,

(2) tools and fixtures,

(3) joint preparation requirements,

(4) cure temperatures,

(5) dimensional requirements and tolerances, and

(6) test acceptance criteria upon completion of assembly.

2 Test assemblies are to be fabricated in accordance with procedures in order to be qualified and such assemblies are to consist of at least one pipe-to-pipe joint and one pipe-to-fitting joint.

3 In cases where such test assemblies have been cured, they are to be subjected to hydrostatic test pressures at safety factors 2.5 *times* the design pressures of such test assemblies for not less than one *hour*. No leakages or separation of joints are allowed. Such tests are to be conducted so that joints are loaded in both longitudinal and circumferential directions.

4 Selection of pipes used for test assemblies is to be in accordance with the following:

(1) In cases where the largest size to be joined has a nominal outside diameter that is 200 *mm* or smaller, test assemblies are to be the largest piping size to be joined.

(2) In cases where the largest size to be joined has a nominal outside diameter that is greater than 200 *mm*, the size of the test assembly is to be either 200 *mm* or 25 % of the largest piping size to be joined, whichever is greater.

5 When conducting performance qualifications, each bonder and each bonding operator are to make up test assemblies, the size and number of which are to be as required in **-4**.

6 Any change in the bonding procedure which will affect the physical and mechanical properties of the joint is to be approved by the Society.

1.6.10 Miscellaneous

1 Sufficient consideration is to be given to any wearing down caused by materials such as sand and sludge.

2 In cases where *GRP* pipes are used as drain pipes from scrubbers and blower casings of inert gas systems, the requirements specified in **35.2.2-1(3), Part R of the Rules** are also to be applied.

3 In cases where plastic pipes are to be installed in external areas, such pipes are to either be specifically approved for external use or be protected against ultraviolet radiation.

4 After installation on board, plastic pipes are to be easily distinguishable from pipes made of other materials.

1.7 Tests

1.7.1 Shop Tests

1 Plastic pipes, except for those piping systems specified in **1.3-2,** are to be subjected to the following tests and measurements of dimension after they have been manufactured. The number of test specimens, testing procedures, results, procedures of measurement of dimension and tolerance are to comply with the internal standards of manufacturers that have been approved by the Society.

(1) Tensile tests

(2) Hydrostatic tests (Hydrostatic pressures are not to be less than 1.5 *times* nominal pressure. Alternatively, for pipes and fittings not employing hand lay-up techniques, the hydrostatic pressure test may be carried out in accordance with the hydrostatic testing requirements stipulated in the recognised national or international standard to which the pipe or fittings are manufactured, provided that there is an effective quality system in place.)

(3) Outside diameter and wall thickness measurements

(4) Ascertainment of uniform quality and the presence of no harmful defects

(5) Electric conductivity test (only for those pipes which require electric conductivity in accordance with **1.5.4 above**)

2 For tests and measurements specified in **-1** above, in cases where the manufacture has been assessed in accordance with the **Rules for Approval of Manufacturers and Service Suppliers**, the requirements that items be tested in the presence of the Surveyor may be reduced. In such cases, the Society's Surveyor may require submission of all relevant test results instead.

3 The plastic pipes specified in **1.3-2** are to be subjected to the tests specified in **-1(2)** and **1.5.2** above for every batch of pipes. Those tests are to be conducted in the presence of the Surveyor. In cases where the manufacturer has been assessed in accordance with the **Rules for Approval of Manufacturers and Service Suppliers** or the manufacturer has a quality system that meets *ISO* 9001:2015 standards or their equivalent, the tests are to be conducted by the manufacturer at the frequency specified in the quality system. In such cases, the Society may require submission of all relevant test results instead. The quality system is to consist of elements necessary to ensure that pipes and fittings are produced with consistent and uniform mechanical and physical properties.

4 Plastic pipes which have been connected by adhesive bonding, laminating, welding, etc. are to be subjected to hydrostatic tests after completion of all fabrication processes at pressures of 1.5 *times* design pressure. (See **1.1.4, Part D of the Rules**) These tests may be carried out after installation on board.

5 Notwithstanding the requirements specified in **-1** above, the Society may request hydrostatic tests for all plastic pipes at a hydrostatic pressure not less than 1.5 *times* the nominal pressure taking into consideration the pipe service conditions.

1.7.2 On board Tests and Inspection

After installation on board, in addition to those tests and inspections specified in **2.1.7, Part B**, the following tests and inspections are to be carried out.

(1) Hydrostatic tests at pressures 1.5 *times* design pressure or 0.4 *MPa*, whichever is greater, used for essential services. Notwithstanding the requirement above, the following **(2)** may be applied to open ended pipes (drains, effluents, etc.).

(2) Leakage tests at service conditions, used for other than auxiliary machinery specified in **(1)** above.

(3) Sufficient earthing to hulls for those pipes required to be electrically conductive in accordance with **1.5.4** above.

(4) Safe support of pipes and no harmful defects on their external surface.

Table 1 Fire Endurance Requirements Matrix

N	Piping Systems	Location										
		A	B	C	D	E	F	G	H	I	J	K
CARGO (FLAMMABLE CARGO f.p.[11] ≤ 60 ℃)												
1	Cargo lines	–	–	L1	–	–	○	–	○[10]	○	–	L1[2]
2	Crude oil washing lines	–	–	L1	–	–	○	–	○[10]	○	–	L1[2]
3	Vent lines	–	–	–	–	–	○	–	○[10]	○	–	×
INERT GAS												
4	Water seal effluent lines	–	–	○[1]	–	–	○[1]	○[1]	○[1]	○[1]	–	○
5	Scrubber effluent lines	○[1]	○[1]	–	–	–	–	–	○[1]	○[1]	–	○
6	Main lines	○	○	L1	–	–	–	–	–	○	–	L1[6]
7	Distribution lines	–	–	L1	–	–	○	–	–	○	–	L1[2]
FLAMMABLE LIQUIDS (f.p.[11] > 60 ℃)												
8	Cargo lines	×	×	L1	×	×	–[3]	○	○[10]	○	–	L1
9	Fuel oil	×	×	L1	×	×	–[3]	○	○	○	L1	L1
10	Lubricating	×	×	L1	×	×	–	–	–	○	L1	L1
11	Hydraulic oil	×	×	L1	×	×	○	○	○	○	L1	L1
SEAWATER[1]												
12	Bilge mains & branches	L1[7]	L1[7]	L1	×	×	–	○	○	○	–	L1
13	Fire mains & water sprays	L1	L1	L1	×	–	–	–	○	○	×	L1
14	Foam systems	L1W	L1W	L1W	–	–	–	–	–	○	L1W	L1W
15	Sprinkler systems	L1W	L1W	L3	×	–	–	–	○	○	L3	L3
16	Ballast	L3	L3	L3	L3	×	○[10]	○	○	○	L2W	L2W
17	Cooling water, essential services[12]	L3	L3	–	–	–	–	–	○	○	–	L2W
18	Tank cleaning services fixed machines	–	–	L3	–	–	○	–	○	○	–	L3[2]
19	Non-essential systems[13]	○	○	○	○	○	–	○	○	○	○	○
FRESHWATER												
20	Cooling water essential services[12]	L3	L3	–	–	–	–	○	○	○	L3	L3
21	Condensate returns	L3	L3	L3	○	○	–	–	–	○	○	○
22	Non-essential systems[13]	○	○	○	○	○	–	○	○	○	○	○
SANITARY/DRAINS/SCUPPERS												
23	Deck drains (internal)	L1W[4]	L1W[4]	–	L1W[4]	○	–	○	○	○	○	○
24	Sanitary drains (internal)	○	○	–	○	○	–	○	○	○	○	○
25	Scuppers and discharges (overboard)	○[1,8]	○[1,8]	○[1,8]	○[1,8]	○[1,8]	○	○	○	○	○[1,8]	○
SOUNDING/AIR												
26	Water tanks/dry spaces	○	○	○	○	○	○[10]	○	○	○	○	○
27	Oil tanks (f.p.[11] > 60 ℃)	×	×	×	×	×	×[3]	○	○[10]	○	×	×
MISCELLANEOUS												
28	Control air	L1[5]	L1[5]	L1[5]	L1[5]	L1[5]	–	○	○	○	L1[5]	L1[5]
29	Service air (non-essential)[13]	○	○	○	○	○	–	○	○	○	○	○
30	Brine	○	○	–	○	○	–	–	–	○	○	○
31	Auxiliary low pressure steam	L2W	L2W	○[9]	○[9]	○[9]	○	○	○	○	○[9]	○[9]

N	Piping Systems	Location										
		A	B	C	D	E	F	G	H	I	J	K
	(≦ 0.7*MPa*)											
32	Central vacuum cleaners	—	—	—	○	—	—	—	—	○	○	○
33	Exhaust gas cleaning system / Exhaust gas recirculation system effluent line	L3[1]	L3[1]	—	—	—	—	—	—	○	L3[1, 14] —	○
34	Reductant agent transfer / supply system (SCR installations)	L1[15]	L1[15]	—	—	—	—	—	—	○	L3[14] —	○

Notes:

(1) LOCATION

A : "Machinery spaces of category *A*": Machinery spaces of category *A* as defined in **2.1.32, Part A of the Rules**

B : "Other machinery spaces and pump rooms": Spaces, other than category *A* machinery spaces and cargo pump rooms, containing: propulsion machinery; boilers; fuel oil units; steam and internal combustion engines; generators and major electrical machinery; oil filling stations; refrigerating, stabilising, ventilation and air-conditioning machinery as well as similar spaces and trunks to such spaces.

C : "Cargo pump rooms": Spaces containing cargo pumps and entrances and trunks to such spaces.

D : "Ro-ro cargo holds": Ro-ro cargo holds are ro-ro cargo spaces and special category spaces as defined in **3.2.41, Part R of the Rules** and **2.1.38, Rules for High Speed Craft**

E : "Other dry cargo holds": All spaces other than ro-ro cargo holds used for non-liquid cargo and trunks to such spaces.

F : "Cargo tanks": All spaces used for liquid cargo and trunks to such spaces.

G : "Fuel oil tanks": All spaces used for fuel oil (excluding cargo tanks) and trunks to such spaces.

H : "Ballast water tanks": All spaces used for ballast water and trunks to such spaces.

I : "Cofferdams, voids, etc.": Cofferdams and voids are those empty spaces between two bulkheads separating two adjacent compartments.

J : "Accommodation, service": Accommodation spaces, service spaces and control stations as defined in **2.1.36, 2.1.37, Part A of the Rules** and **9.2.3-2(1), Part R of the Rules**

K : "Open decks": Open deck spaces as defined in **9.2.4-2(10), Part R of the Rules** (excluding lifeboat and liferaft embarkation and lowering stations)

(2) ABBREVIATIONS

L1 : Pipes without leakage during pressure tests as a result of fire endurance tests (for more than one hour) and pressure tests (for more than 15 *minutes*) in dry conditions in accordance with *IMO Res*. *A*.753(18) *Appendix* 1 (including any amendments due to *IMO Res*. *MSC*.313(88) and *IMO Res*. *MSC*.399(95))

L1W : For piping systems which do not carry flammable fluid or any gas, pipes with negligible leakage (i.e. not exceeding 5 % flow loss) during pressure tests as a result of fire endurance tests (for more than one hour) and pressure tests (for more than 15 *minutes*) in dry conditions in accordance with *IMO Res*. *A*.753(18) *Appendix* 1 (including any amendments due to *IMO Res*. *MSC*.313(88) and *IMO Res*. *MSC*.399(95))

L2 : Pipes without leakage during pressure tests as a result of fire endurance tests (for more than 30 *minutes*) and pressure tests (for more than 15 *minutes*) in dry conditions in accordance with *IMO Res*. *A*.753(18) *Appendix* 1 (including any amendments due to *IMO Res*. *MSC*.313(88) and *IMO Res*. *MSC*.399(95))

L2W : Pipes with negligible leakage (i.e. not exceeding 5 % flow loss) during pressure tests as a result of fire endurance tests (for more than 30 *minutes*) and pressure tests (for more than 15 *minutes*) in dry conditions in accordance with *IMO Res*. *A*.753(18) *Appendix* 1 (including any amendments due to *IMO Res*. *MSC*.313(88) and *IMO Res*. *MSC*.399(95))

L3 : Pipes without significant leakage (i.e. not exceeding 0.2 *l*/*min*) during pressure tests as a result of fire endurance tests (for more than 30 *minutes*) and pressure tests (for more than 15 *minutes*) in wet conditions in accordance with *IMO Res*. *A*.753(18) *Appendix* 1 (including any amendments due to *IMO Res*. *MSC*.313(88)

N	Piping Systems	Location										
		A	B	C	D	E	F	G	H	I	J	K

and *IMO Res. MSC*.399(95))

○ : No fire endurance test required

– : Not applicable

× : Metallic materials having a melting point greater than 925 ºC

(3) FOOTNOTES

1 : In cases where non-metallic piping is used, remotely controlled valves are to be provided at ship's side (such valves are to be controlled from outside spaces).

2 : Remote closing valves are to be provided at cargo tanks.

3 : When cargo tanks contain flammable liquids with a f.p. (to be determined by an approved closed cup method) > 60 ºC, "○" may replace "–" or "×".

4 : In the case of drains serving only the space concerned, "○" may replace "L1W".

5 : When controlling functions are not required by statutory requirements or guidelines, "○" may replace "L1".

6 : In the case of pipes between machinery spaces and deck water seals, "○" may replace "L1".

7 : In the case of passenger vessels, "×" is to replace "L1".

8 : Scuppers serving open decks in positions I and II, as defined in **1.4.3.2, Part 1, Part C**, are to be "×" throughout unless fitted at the upper end with the means of closing capable of being operated from a position above the freeboard deck in order to prevent downflooding.

9 : In the case of essential services, such as fuel oil tank heating and the ship's whistle, "×" is to replace "○".

10 : In the case of tankers where compliance with **3.2.4(1)(a)vi), Part 3 of the Rules for Marine Pollution Prevention Systems** is required, "–" is to replace "○".

11 : To be determined by an approved closed cup method.

12 : Pipelines used for essential services.

13 : Pipes specified in **1.3-2(1)** and **(2)**

14 : L3 in service spaces, NA in accommodation and control spaces.

15 :Type approved plastic piping without fire endurance test (○) is acceptable downstream of the tank valve, provided this valve is metal seated and arranged as fail-to-closed or with quick closing from a safe position outside the space in the event of fire.

Appendix 1 GUIDANCE FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE WEB FILLET RADII OF CRANKSHAFTS BY UTILIZING FINITE ELEMENT METHOD

1.1 General

The objective of the analysis is to develop Finite Element Method (FEM) calculated figures as an alternative to the analytically calculated Stress Concentration Factors (SCF) at the crankshaft fillets. The analytical method is based on empirical formulae developed from strain gauge measurements of various crank geometries and accordingly the application of these formulae is limited to those geometries.

The SCF's calculated in accordance with the rules of this appendix are defined as the ratio of stresses calculated by FEM to nominal stresses in both journal and pin fillets. When used in connection with the present method in **Annex 2.3.1**, von Mises stresses is to be calculated for bending and principal stresses for torsion.

The procedure as well as evaluation guidelines are valid for both solid cranks and semi-built cranks (except journal fillets).

The analysis is to be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude are to be applied for all load cases.

The calculation of SCF at the oil bores is covered by **Appendix 4**.

It is advised to check the element accuracy of the FE solver in use, e.g. by modeling a simple geometry and comparing the stresses obtained by FEM with the analytical solution for pure bending and torsion.

Boundary Element Method (BEM) may be used instead of FEM.

2.1 Model Requirements

The basic recommendations and perceptions for building the FE-model are presented in **2.1.1**. It is obligatory for the final FE-model to fulfill the requirement in **2.2**.

2.1.1 Element Mesh Recommendations

1 In order to fulfil the mesh quality criteria, it is advised to construct the FE model for the evaluation of Stress Concentration Factors in accordance with the following recommendations:

(1) The model consists of one complete crank, from the main bearing centre line to the opposite side main bearing centre line.

(2) Element type used in the vicinity of the fillets:

  (a) 10-node tetrahedral elements

  (b) 8-node hexahedral elements

  (c) 20-node hexahedral elements

(3) Mesh properties in fillet radii applied to ±90 degrees in a circumferential direction from the crank plane are as follows:

  (a) Maximum element size a through the entire fillet as well as in the circumferential direction is to be $a=R_H/4$ in crankpin fillets and $a=R_G/4$ in journal fillets. When using 20-node hexahedral elements, the element size in the circumferential direction may be extended up to $5a$. In the case of multi-radii fillet, the local fillet radius is to be applied.

  (b) Element size in fillet depth direction (*See* **Fig. 1**):

    i) First layer thickness equal to element size of a

    ii) Second layer thickness equal to element to size of $2a$

    iii) Third layer thickness equal to element to size of $3a$

(4) A minimum of 6 elements are to be set across the web thickness.

(5) The rest of the crank is to be suitable for numeric stability of the solver.

(6) Counterweights have to be modelled only when influencing the global stiffness of the crank significantly.

(7) Modelling of oil drillings is not necessary as long as the influence on global stiffness is negligible and the proximity to the fillet is more than $2R_H$ or $2R_G$ (*See* **Fig. 2**)

(8) Drillings and holes for weight reduction have to be modelled.

(9) Sub-modelling may be used as far as the software requirements are fulfilled.

Fig. 1 Element Size in Fillet Depth Direction

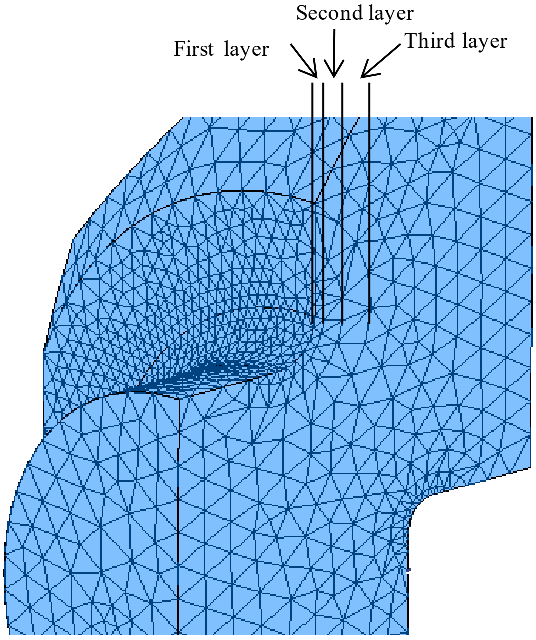
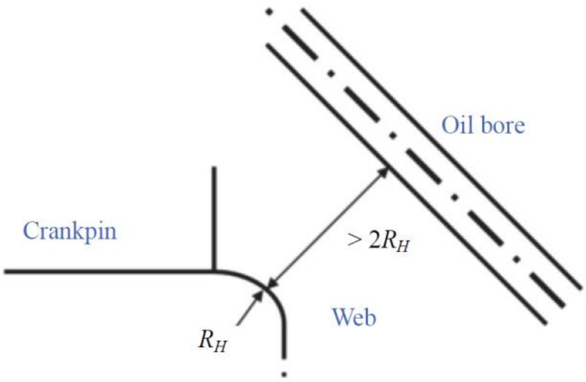


Fig. 2 Oil Bore Proximity to Fillet



2.1.2 Material

1 Material properties applied to steels are as follows:

Young's Modulus : $E = 2.05 \cdot 10^5 MPa$

Poisson's ratio : $\nu = 0.3$

2 For materials other than steels, reliable values for material parameters are to be used, either as quoted in literature or as measured on representative material samples.

2.2 Element Mesh Quality Criteria

If the actual element mesh does not fulfill any of the following criteria at the area examined for SCF evaluation, then a second calculation with a refined mesh is to be performed.

2.2.1 Principal Stresses Criterion

The quality of the mesh is to be assured by checking the stress component normal to the surface of the fillet radius. Ideally, this stress is to be zero. With principal stresses σ_1, σ_2 and σ_3, the following criterion is required:

$$\min(|\sigma_1|, |\sigma_2|, |\sigma_3|) < 0.03 \cdot \max(|\sigma_1|, |\sigma_2|, |\sigma_3|)$$

2.2.2 Averaged/Unaveraged Stresses Criterion

Unaveraged nodal stress results calculated from each element connected to a node is to differ less than by 5 % from the 100 % averaged nodal stress results at this node at the examined location.

3.1 Load Cases

The following load cases have to be calculated.

3.1.1 Torsion

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 3** where the torque is applied to the central node located at the crankshaft axis.

2 For all nodes in both the journal and crankpin fillet, principal stresses are extracted and the equivalent torsional stress is calculated as follows:

$$\tau_{equiv} = \max\left(\frac{|\sigma_1-\sigma_2|}{2}, \frac{|\sigma_2-\sigma_3|}{2}, \frac{|\sigma_1-\sigma_3|}{2}\right)$$

3 The maximum value taken for the subsequent calculation of the stress concentration factors for torsion in crankpin and journal fillet.

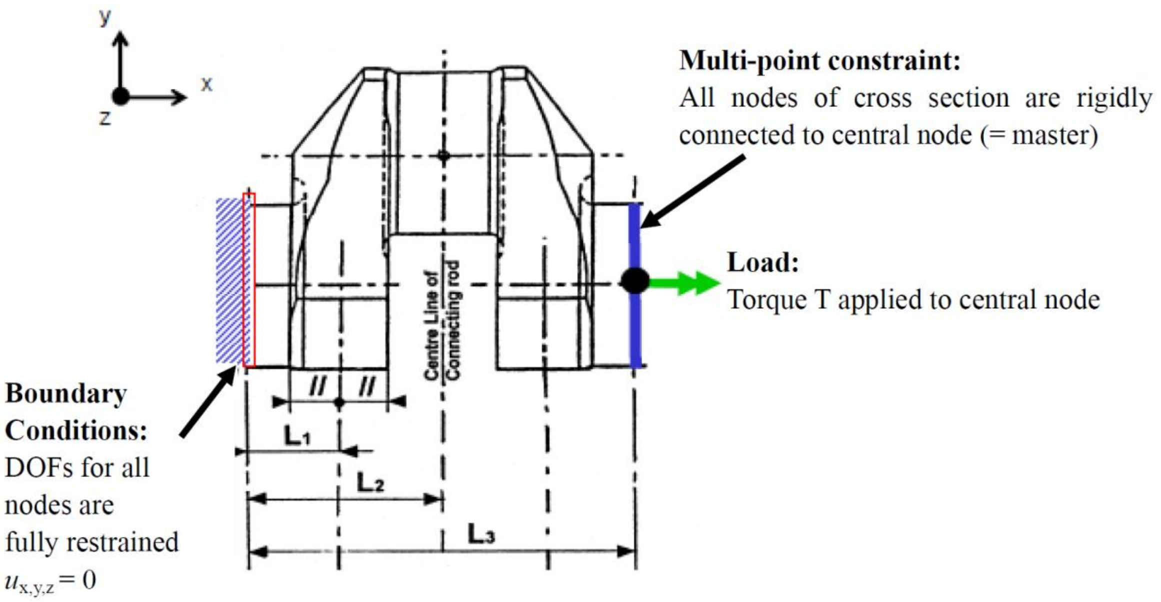
$$\alpha_T = \frac{\tau_{equiv,\alpha}}{\tau_N}$$

$$\beta_T = \frac{\tau_{equiv,\beta}}{\tau_N}$$

where τ_N is nominal torsional stress for the crankpin and journal respectively and is calculated as follows (for W_P *see* **1.3.2** of **Annex 2.3.1**):

$$\tau_N = \frac{T}{W_P}$$

Fig. 3 Boundary and Load Conditions for the Torsion Load Case



3.1.2 Pure Bending (4-point Bending)

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 4** where the bending moment is applied to the central node located at the crankshaft axis.

2 For all nodes in both the journal and crankpin fillet, von Mises equivalent stresses σ_{equiv} are extracted. The maximum value is used to calculate the stress concentration factors for bending in crankpin and journal fillet according to the following formulae:

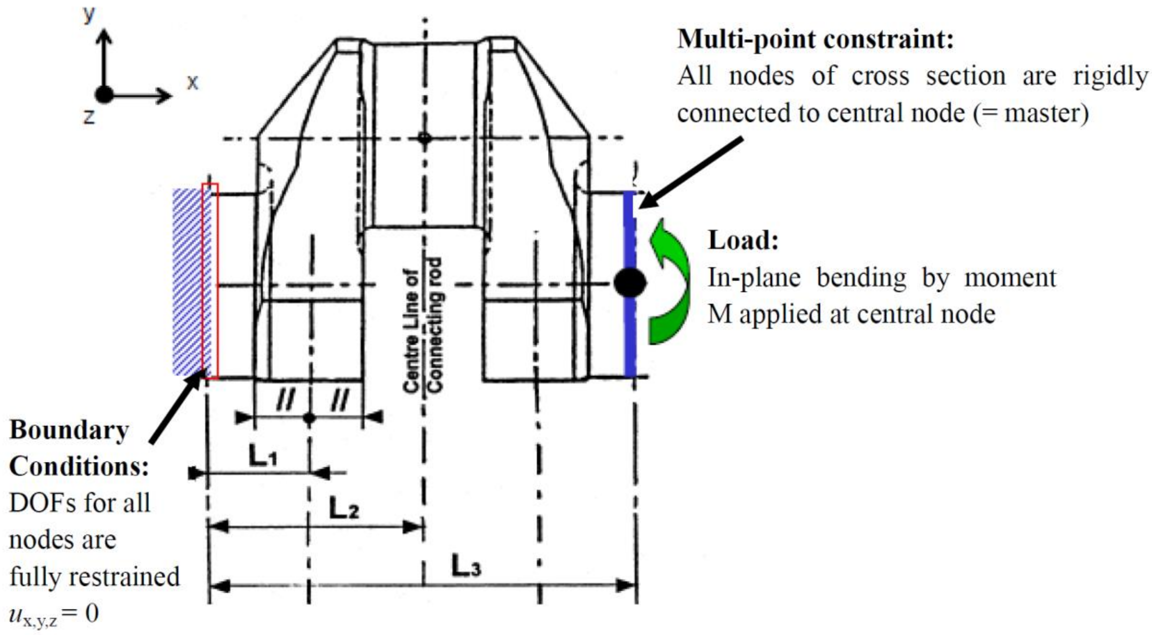
$$\alpha_B = \frac{\sigma_{equiv,\alpha}}{\sigma_N}$$

$$\beta_B = \frac{\sigma_{equiv,\beta}}{\sigma_N}$$

where σ_N is nominal bending stress for the crankpin and journal respectively and is calculated as follows (for W_{eqw} *see* **1.3.1-2(2)** of **Annex 2.3.1**):

$$\sigma_N = \frac{M}{W_{eqw}}$$

Fig. 4 Boundary and Load Conditions for the Pure Bending Load Case



3.1.3 Bending with Shear Force (3-point Bending)

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 5** where the force is applied to the central node located at the pin centre line of the connecting rod.

Fig. 5 Boundary and Load Conditions for the 3-point Bending Load Case of an Inline Engine.

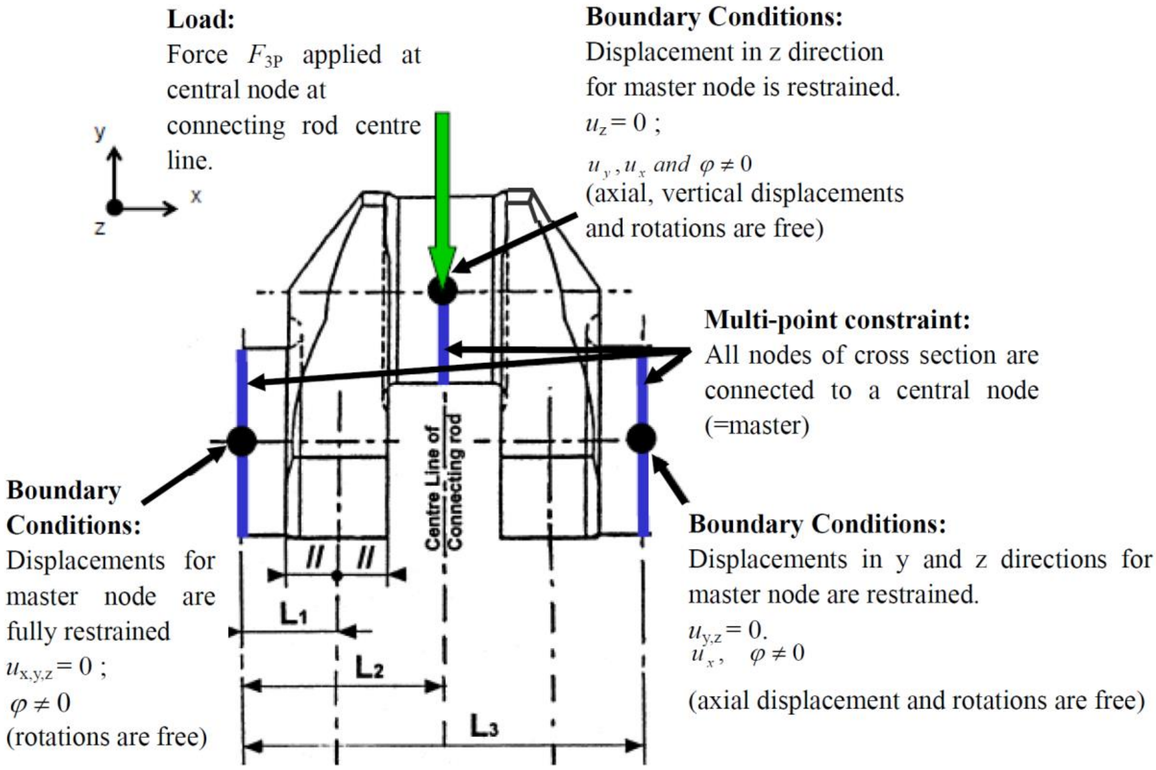
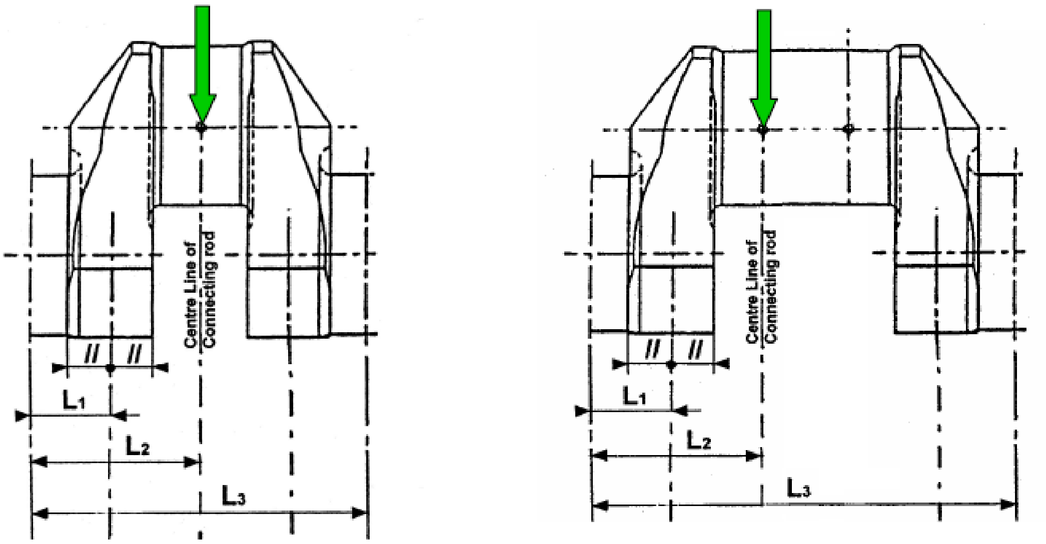


Fig. 6 Load Applications for In-line and Vee Type Engines



2 The maximum equivalent von Mises stress σ_{3P} in the journal fillet is evaluated. The stress concentration factors in the journal fillet can be determined as shown **i)** or **ii)**.

(1) Stress concentration factor for compression due to radial force in journal fillet β_Q is calculated according to the following:

$$\sigma_{3P} = \sigma_{N3P} \cdot \beta_B + \sigma_{Q3P} \cdot \beta_Q$$

where

σ_{3P} : as found by the Finite Element Calculation

σ_{N3P}: Nominal bending stress in the web centre due to force F_{3P} applied to the centre line of the actual connecting rod (*See* **Fig. 6**)

β_B : as determined in **3.1.2-2**

$$\sigma_{Q3P} = Q_{3P}/(B.W)$$

Q_{3P} : the radial (shear) force in the web due to the force F_{3P} applied to the centre line of the actual connecting rod (*See* **Fig. 1** and **Fig. 2** of **Annex 2.3.1**)

(2) The stress concentration factor for bending and compression due to radial force in journal fillet β_{BQ} is calculated according to the following:

$$\beta_{BQ} = \frac{\sigma_{3P}}{\sigma_{N3P}}$$

for the relevant parameters *See* **(1)**.

Appendix 2 GUIDANCE FOR EVALUATION OF FATIGUE TESTS

1.1 Introduction

Fatigue testing can be divided into two main groups; testing of small specimens and full-size crank throws. Testing can be made using the staircase method or a modified version thereof which is presented in this appendix. Other statistical evaluation methods may also be applied.

1.2 Small Specimen Testing

1 For crankshafts without any fillet surface treatment, the fatigue strength can be determined by testing small specimens taken from a full-size crank throw.

2 When other areas in the vicinity of the fillets are surface treated introducing residual stresses in the fillets, this approach cannot be applied.

3 One advantage of this approach is the rather high number of specimens which can be then manufactured. Another advantage is that the tests can be made with different stress ratios (R-ratios) and/or different modes e.g. axial, bending and torsion, with or without a notch. This is required for evaluation of the material data to be used with critical plane criteria.

1.3 Full-size Crank Throw Testing

1 For crankshafts with surface treatment the fatigue strength can only be determined through testing of full-size crank throws.

2 The load can be applied by hydraulic actuators in a 3- or 4-point bending arrangement, or by an exciter in a resonance test rig. The latter is frequently used, although it usually limits the stress ratio to $R = -1$.

2.1 Evaluation of Test Results

2.1.1 Principles

1 Prior to fatigue testing the crankshaft is to be tested as required by quality control procedures, e.g. for chemical composition, mechanical properties, surface hardness, hardness depth and extension, fillet surface finish, etc.

2 The test samples are to be prepared so as to represent the "lower end" of the acceptance range e.g. for induction hardened crankshafts this means the lower range of acceptable hardness depth, the shortest extension through a fillet, etc. Otherwise, the mean value test results is to be corrected with a confidence interval: a 90 % confidence interval may be used both for the sample mean and the standard deviation.

3 The test results are to be evaluated to represent the mean fatigue strength, with or without taking into consideration the 90 % confidence interval as mentioned above. The standard deviation is to be considered by taking the 90 % confidence into account. Subsequently the result to be used as the fatigue strength is then the mean fatigue strength minus one standard deviation.

4 If the evaluation aims to find a relationship between (static) mechanical properties and the fatigue strength, the relation is to be based on the real (measured) mechanical properties, not on the specified minimum properties.

5 The calculation technique in **2.1.4** was developed for the original staircase method. However, since there is no similar method dedicated to the modified staircase method the same is applied for both.

2.1.2 Staircase Method

1 In the original staircase method, fatigue testing is carried out as follows:

(1) The first specimen is subjected to a stress corresponding to the expected average fatigue strength.

(2) If the specimen specified in **(1)** survives 10^7 cycles, it is discarded and the next specimen is subjected to a stress that is one increment above the previous.

(3) A survivor is always followed by the next using a stress one increment above the previous, as specified in **(2)**. The increment is

to be selected to correspond to the expected level of the standard deviation.

(4) When a specimen fails prior to reaching 10^7 cycles, the obtained number of cycles is noted and the next specimen is subjected to a stress that is one increment below the previous.

2 This original staircase method is only suitable when a high number of specimens are available.

3 The minimum number of test specimens is to be 25.

2.1.3 Modified Staircase Method

1 When a limited number of specimens are available, it is advisable to apply the modified staircase method.

2 In the modified staircase method, fatigue testing is carried out as follows:

(1) The first specimen is subjected to a stress level that is most likely well below the average fatigue strength.

(2) When this specimen specified in **(1)** has survived 10^7 cycles, this same specimen is subjected to a stress level one increment above the previous. The increment is to be selected to correspond to the expected level of the standard deviation. This is continued with the same specimen until failure.

(3) Then the number of cycles is recorded and the next specimen is subjected to a stress that is at least 2 increments below the level where the previous specimen failed.

3 The acquired result of a modified staircase method is to be used with care, since some results available indicate that testing a runout on a higher test level, especially at high mean stresses, tends to increase the fatigue limit. However, this "training effect" is less pronounced for high strength steels (e.g. *UTS* > 800 *MPa*).

4 The minimum number of test specimens is to be 3.

2.1.4 Calculation of Sample Mean and Standard Deviation

A hypothetical example of tests for 5 crank throws is presented further in the subsequent text.

(1) When using the modified staircase method and the evaluation method of Dixon and Mood, the number of samples will be 10, meaning 5 run-outs and 5 failures, i.e.:

Number of samples, $n = 10$

(2) Furthermore, the method distinguishes between:

(a) Less frequent event is failures: $C = 1$

(b) Less frequent event is run-outs: $C = 2$

The method uses only the less frequent occurrence in the test results, i.e. if there are more failures than run-outs, then the number of run-outs is used.

(3) In the modified staircase method, the number of run-outs and failures are usually equal. However, the testing can be unsuccessful, e.g. the number of run-outs can be less than the number of failures if a specimen with 2 increments below the previous failure level goes directly to failure. On the other hand, if this unexpected premature failure occurs after a rather high number of cycles, it is possible to define the level below this as a run-out.

(4) Dixon and Mood's approach, derived from the maximum likelihood theory, which also may be applied here, especially on tests with few samples, presented some simple approximate equations for calculating the sample mean and the standard deviation from the outcome of the staircase test.

(a) The sample mean can be calculated as follows:

$$\overline{S_a} = S_{a0} + d \cdot \left(\frac{A}{F} - \frac{1}{2}\right) \qquad \text{when } C = 1$$

$$\overline{S_a} = S_{a0} + d \cdot \left(\frac{A}{F} + \frac{1}{2}\right) \qquad \text{when } C = 2$$

(b) The standard deviation can be found by

$$s = 1.62 \cdot d \cdot \left(\frac{F \cdot B - A^2}{F^2} + 0.029\right)$$

where:

S_{a0} is the lowest stress level for the less frequent occurrence

d is the stress increment

$$F = \sum f_i$$

$$A = \sum i \cdot f_i$$

$$B = \sum i^2 \cdot f_i$$

i is the stress level numbering

f_i is the number of samples at stress level i

The formula for the standard deviation is an approximation and can be used when

$$\frac{B \cdot F - A^2}{F^2} > 0.3$$

and

$$0.5 \cdot s < d < 1.5 \cdot s$$

If any of these two conditions are not fulfilled, a new staircase test is to be considered or the standard deviation is to be taken quite large in order to be on the safe side.

(5) If increment d is greatly higher than the standard deviation s, the procedure leads to a lower standard deviation and a slightly higher sample mean, both compared to values calculated when the difference between the increment and the standard deviation is relatively small. Respectively, if increment d is much less than the standard deviation s, the procedure leads to a higher standard deviation and a slightly lower sample mean.

****

Example

Hypothetical test results are shown in **Fig. 1**. The processing of the results and the evaluation of the sample mean and the standard deviation are shown in **Fig. 2**.

Fig. 1 Log Sheet of a Modified Staircase Test.

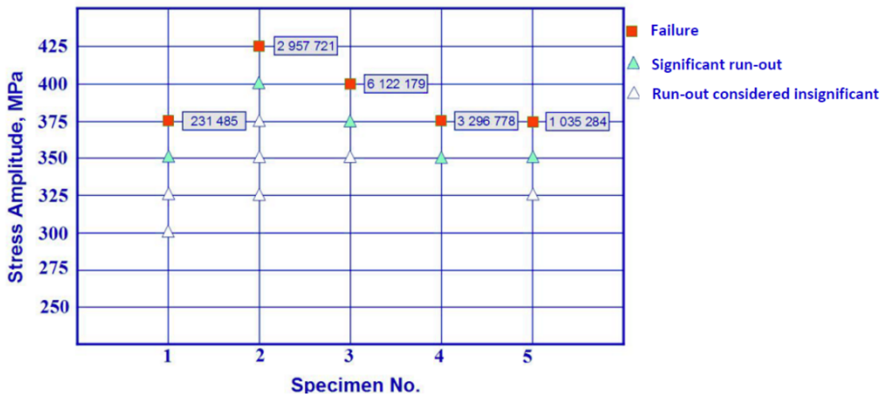
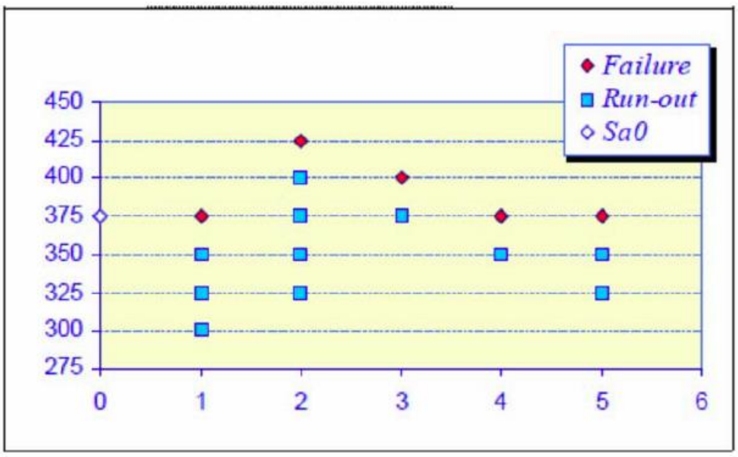


Fig. 2 Processing of a Modified Staircase Test Results.



i	fi	i*fi	i^2*fi
2	1	2	4
1	1	1	1
0	3	0	0
Σ	5	3	5
	F	A	B

$$F = \sum f_i$$

$$A = \sum i \cdot f_i$$

$$B = \sum i^2 \cdot f_i$$

Notes:

$i = 0, 1, 2\ldots$ is the stress level numbering, the numbering usually starts from zero

f_i is number of test specimen at stress level, i

Sample mean and standard deviation are evaluated as follows based upon **Fig. 2**.

(1) Stress level 0, $S_{a0} := 375\,MPa$

Level 0 is the lowest value of the less frequent occurrence in the test results.

(2) Stress increment, $d := 25\,MPa$

(3) $F := 5$, $A := 3$, $B := 5$

(4) Calculation of sample mean is as follows:

$$S_a := S_{a0} + d \cdot \left(\frac{A}{F} - \frac{1}{2}\right) \quad C = 1 \quad S_a = 375.5MPa$$

(5) Calculation of sample standard deviation is as follows:

$$s := 1.62 \cdot d \cdot \left(\frac{B \cdot F - A^2}{F^2} + 0.029\right) \quad S = 27.09MPa$$

(6) Calculation of standard deviation ratio is as follows:

$$S_r := \frac{s}{S_a} \quad S_r = 0.072$$

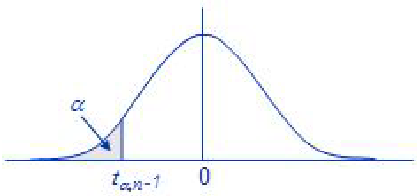
****

2.1.5 Confidence Interval for Mean Fatigue Limit

1 If the staircase fatigue test is repeated, the sample mean and the standard deviation will most likely be different from the previous test. Therefore, it is necessary to assure with a given confidence that the repeated test values will be above the chosen fatigue limit by using a confidence interval for the sample mean.

2 The confidence interval for the sample mean value with unknown variance is known to be distributed in accordance with the *t*-distribution (also called student's *t*-distribution) which is a distribution symmetric around the average. (*See* **Fig. 3**)

Fig. 3 Student's *t*-distribution



Note:

The confidence level normally used for the sample mean is 90 %, meaning that 90 % of sample means from repeated tests will be above the value calculated with the chosen confidence level. The **Fig. 3** shows the t-value for (1-α) · 100 % confidence interval for the sample mean.

3 If S_a is the empirical mean and *s* is the empirical standard deviation over a series of *n* samples, in which the variable values are normally distributed with an unknown sample mean and unknown variance, the $(1-\alpha) \cdot 100\ \%$ confidence interval for the mean is:

$$P\left(S_a - t_{\alpha,n-1} \cdot \frac{s}{\sqrt{n}} < S_{aX\%}\right) = 1 - \alpha$$

4 The resulting confidence interval is symmetric around the empirical mean of the sample values, and the lower endpoint can be found as:

$$S_{aX\%} = S_a - t_{\alpha,n-1} \cdot \frac{s}{\sqrt{n}}$$

which is the mean fatigue limit (population value) to be used to obtain the reduced fatigue limit where the limits for the probability of failure are taken into consideration.

****

Example

Applying a 90 % confidence interval ($\alpha = 0.1$) and n = 10 (5 failures and 5 run-outs) leads to $t_{\alpha,n-1} = 1.383$, taken from a table for statistical evaluations (E. Dougherty: Probability and Statistics for the Engineering, Computing and Physical Sciences, 1990. Note that $v = n - 1$ in the tables.). Hence:

$$S_{a90\%} = S_a - 1.383 \cdot d \cdot \frac{s}{\sqrt{n}} = S_a - 0.4373 \cdot s$$

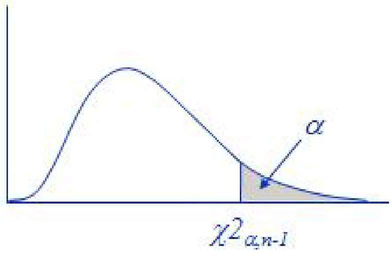
To be conservative, some authors would consider n to be 5, as the physical number of used specimens, then $t_{\alpha,n-1} = 1.533$.

****

2.1.6 Confidence Interval for Standard Deviation

1 The confidence interval for the variance of a normal random variable is known to possess a chi-square distribution with $n - 1$ degrees of freedom (*See* **Fig. 4**).

Fig. 4 Chi-square Distribution



Note:

The confidence level on the standard deviation is used to ensure that the standard deviations for repeated tests are below an upper limit obtained from the fatigue test standard deviation with a confidence level. **Figure 4** shows the chi-square for (1-α)· 100 % confidence interval for the variance.

2 An assumed fatigue test value from n samples is a normal random variable with a variance of σ^2 and has an empirical variance s^2. Then a $(1 - \alpha) \cdot 100$ % confidence interval for the variance is obtained according to the following formulae:

$$P\left(\frac{(n-1)s^2}{\sigma^2} < \chi^2_{\alpha,n-1}\right) = 1 - \alpha$$

3 A $(1 - \alpha) \cdot 100$ % confidence interval for the standard deviation is obtained by the square root of the upper limit of the confidence interval for the variance and can be obtained according to the following formula:

$$S_{X\%} = \sqrt{\frac{n-1}{\chi^2_{\alpha,n-1}}} \cdot s$$

This standard deviation (population value) is to be used to obtain the fatigue limit, where the limits for the probability of failure are taken into consideration.

****

Example

Applying a 90 % confidence interval ($\alpha = 0.1$) and n = 10 (5 failures and 5 run-outs) leads to $\chi^2_{\alpha,n-1} = 4.168$, taken from a table for statistical evaluations (E. Dougherty: Probability and Statistics for the Engineering, Computing and Physical Sciences, 1990).

Hence:

$$S_{90\%} = \sqrt{\frac{n-1}{4.168}} \cdot s = 1.47 \cdot s$$

To be conservative, some authors would consider n to be 5, as the physical number of the used specimens, then $\chi^2_{\alpha,n-1} = 1.064$.

****

3.1 Small Specimen Testing

3.1.1 General

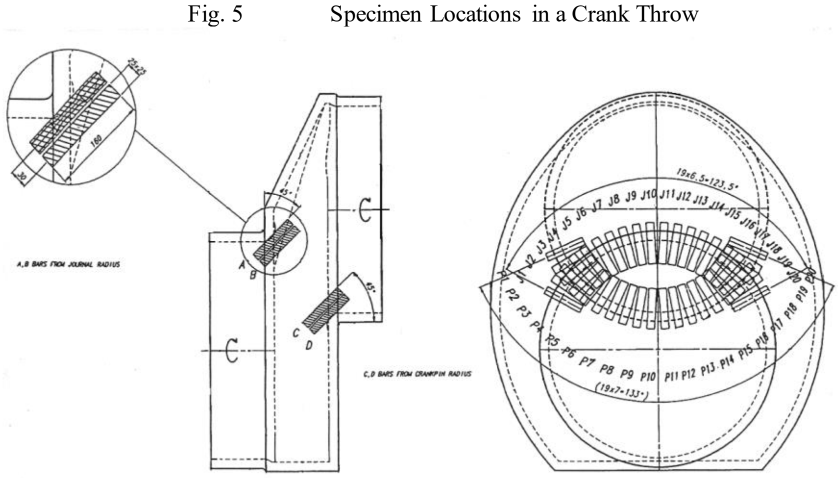
1 In this appendix, a small specimen is considered to be one of the specimens taken from a crank throw.

2 Since the specimens are to be representative for the fillet fatigue strength, they are to be taken out close to the fillets. (*See* **Fig. 5**)

3 It is to be made certain that the principal stress direction in the specimen testing is equivalent to the full-size crank throw. The verification is recommended to be done by utilizing the finite element method.

4 The (static) mechanical properties are to be determined as stipulated by the quality control procedures.

Fig. 5 Specimen Locations in a Crank Throw



3.1.2 Determination of Bending Fatigue Strength

1 It is advisable to use un-notched specimens in order to avoid uncertainties related to the stress gradient influence. Push-pull testing method (stress ratio $R = -1$) is preferred, but especially for the purpose of critical plane criteria other stress ratios and methods may be added.

2 In order to ensure principal stress direction in push-pull testing to represent the full-size crank throw principal stress direction and when no further information is available, the specimen is to be taken at a 45-degree angle as shown in **Fig. 5**. If the objective of the testing is to document the influence of high cleanliness, test samples taken from positions approximately 120 degrees in a circumferential direction may be used. (*See* **Fig. 5**) If the objective of the testing is to document the influence of continuous grain flow (cgf) forging, the specimens are to be restricted to the vicinity of the crank plane.

3.1.3 Determination of Torsional Fatigue Strength

1 If the specimens are subjected to torsional testing, the selection of samples is to follow the same guidelines as for bending above. The stress gradient influence has to be considered in the evaluation.

2 If the specimens are tested in push-pull and no further information is available, the samples are to be taken out at a 45-degree angle to the crank plane in order to ensure collinearity of the principal stress direction between the specimen and the full-size crank throw. When taking the specimen at a distance from the (crank) middle plane of the crankshaft along the fillet, this plane rotates around the pin centre point making it possible to resample the fracture direction due to torsion (the results are to be converted into the pertinent torsional values).

3.1.4 Other Test Positions

1 If the test purpose is to find fatigue properties and the crankshaft is forged in a manner likely to lead to cgf, the specimens may also be taken longitudinally from a prolonged shaft piece where specimens for mechanical testing are usually taken. The condition is that this prolonged shaft piece is heat treated as a part of the crankshaft and that the size is so as to result in a similar quenching rate as the crank throw.

2 When using test results from a prolonged shaft piece, it has to be considered how well the grain flow in that shaft piece is representative for the crank fillets.

3.1.5 Correlation of Test Results

1 The fatigue strength achieved by specimen testing is to be converted to correspond to the full-size crankshaft fatigue strength with an appropriate method (size effect).

2 When using the bending fatigue properties from tests mentioned in **3.1**, it is to be kept in mind that successful continuous grain flow (cgf) forging leading to elevated values compared to other (non cgf) forging, will normally not lead to a torsional fatigue strength improvement of the same magnitude. In such cases it is advised to either carry out also torsional testing or to make a conservative assessment of the torsional fatigue strength. This approach is applicable when using the Gough Pollard criterion. However, this approach is not recognised when using the von Mises or a multi-axial criterion such as Findley.

3 If the found ratio between bending and torsion fatigue differs significantly from $\sqrt{3}$, one is to consider replacing the use of the von Mises criterion with the Gough Pollard criterion. Also, if critical plane criteria are used, it has to be kept in mind that cgf makes the material inhomogeneous in terms of fatigue strength, meaning that the material parameters differ with the directions of the planes.

4 Any addition of influence factors is to be made with caution. If for example a certain addition for clean steel is documented, it may not necessarily be fully combined with a K-factor for cgf. Direct testing of samples from a clean and cgf forged crank is preferred.

4.1 Full-Size Testing

4.1.1 Hydraulic Pulsation

1 A hydraulic test rig can be arranged for testing a crankshaft in 3-point or 4-point bending as well as in torsion. This allows for testing with any R-ratio.

2 Although the applied load is to be verified by strain gauge measurements on plain shaft sections for the initiation of the test, it is not necessarily used during the test for controlling load. It is also pertinent to check fillet stresses with strain gauge chains.

3 Furthermore, it is important that the test rig provides boundary conditions as defined in **3.1** of **Appendix 3**.

4 The (static) mechanical properties are to be determined as stipulated by the quality control procedures.

4.1.2 Resonance Tester

1 A rig for bending fatigue normally works with an R-ratio of -1. **Fig. 6** shows a layout of the testing arrangement.

2 The applied load is to be verified by strain gauge measurements on plain shaft sections. It is also pertinent to check fillet stresses with strain gauge chains.

3 Clamping around the journals is to be arranged in a way that prevents severe fretting which could lead to a failure under the edges of the clamps. If some distance between the clamps and the journal fillets is provided, the loading is consistent with 4-point bending and thus representative for the journal fillets also.

4 In an engine, the crankpin fillets normally operate with an R-ratio slightly above -1 and the journal fillets slightly below -1. If found necessary, it is possible to introduce a mean load (deviate from $R = -1$) by means of a spring preload.

5 A rig for torsion fatigue can also be arranged as shown in **Fig. 7**. When a crank throw is subjected to torsion, the twist of the crankpin makes the journals move sideways. If one single crank throw is tested in a torsion resonance test rig, the journals with their clamped-on weights will vibrate heavily sideways. This sideway movement of the clamped-on weights can be reduced by having two crank throws, especially if the cranks are almost in the same direction. However, the journal in the middle will move more.

6 Since sideway movements can cause some bending stresses, the plain portions of the crankpins are to also be provided with strain gauges arranged to measure any possible bending that could have an influence on the test results.

7 Similarly, to the bending case the applied load is to be verified by strain gauge measurements on plain shaft sections. It is also pertinent to check fillet stresses with strain gauge chains as well.

Fig. 6 An Example of Testing Arrangement of the Resonance Tester for Bending Loading

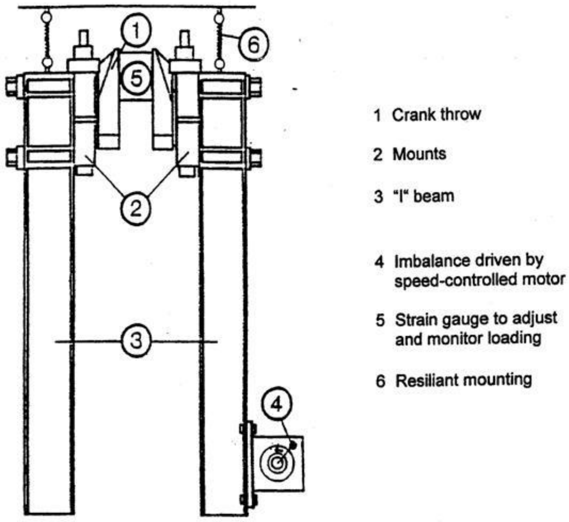
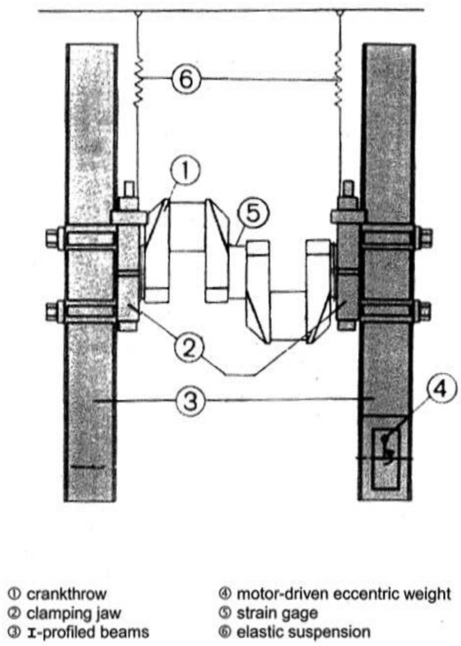


Fig. 7 An Example of Testing Arrangement of the Resonance Tester for Torsion Loading with Double Crank Throw Section



4.1.3 Use of Results and Crankshaft Acceptability

1 In order to combine tested bending and torsion fatigue strength results in calculation of crankshaft acceptability (*See* **1.8** of **Annex 2.3.1**), the Gough-Pollard approach and the maximum principal equivalent stress formulation can be applied for the following cases:

(1) At the crankpin fillet:

$$Q = \left(\sqrt{\left(\frac{\sigma_{BH} + \sigma_{add}}{\sigma_{DWCT}} \right)^2 + \left(\frac{\tau_H}{\tau_{DWCT}} \right)^2} \right)^{-1}$$

where:

σ_{DWCT} : fatigue strength by bending testing

τ_{DWCT} : fatigue strength by torsion testing

for other parameters see **1.3.1-3**, **1.3.2-2** and **1.5, Annex 2.3.1**

(2) Related to crankpin oil bore:

$$Q = \frac{\sigma_{DWOT}}{\sigma_V};\ \sigma_V = \frac{1}{3}\sigma_{BO} \cdot \left[1 + 2\sqrt{1 + \frac{9}{4}\left(\frac{\sigma_{TO}}{\sigma_{BO}}\right)^2}\right]$$

where:

σ_{DWOT} : fatigue strength by means of maximum principal stress from torsion testing

(3) At the journal fillet:

$$Q = \left(\sqrt{\left(\frac{\sigma_{BG} + \sigma_{add}}{\sigma_{DWJT}}\right)^2 + \left(\frac{\tau_G}{\tau_{DWJT}}\right)^2}\right)^{-1}$$

where:

σ_{DWJT} : fatigue strength by bending testing

τ_{DWJT} : fatigue strength by torsion testing

for other parameters see **1.3.1-3**, **1.3.2-2** and **1.5, Annex 2.3.1**

2 In case increase in fatigue strength due to the surface treatment is considered to be similar between the above cases, it is sufficient to test only the most critical location in accordance with the calculation where the surface treatment had not been taken into account.

5.1 Use of Existing Results for Similar Crankshafts

5.1.1 Use of Existing Results

1 For fillets or oil bores without surface treatment, the fatigue properties found by testing may be used for similar crankshaft designs providing:

(1) Material:

(a) Similar material type

(b) Cleanliness on the same or better level

(c) The same mechanical properties can be granted (size versus hardenability)

(2) Geometry:

(a) Difference in the size effect of stress gradient is insignificant or it is considered

(b) Principal stress direction is equivalent. (*See* **3.1**)

(3) Manufacturing:

(a) Similar manufacturing process

2 Induction hardened or gas nitrited crankshafts will suffer fatigue either at the surface or at the transition to the core. The surface fatigue strength as determined by fatigue tests of full-size cranks, may be used on an equal or similar design as the tested crankshaft when the fatigue initiation occurred at the surface. With the similar design, it is meant that a similar material type and surface hardness are used and the fillet radius and hardening depth are within approximately ± 30 % of the tested crankshaft.

3 Fatigue initiation in the transition zone can be either subsurface, i.e. below the hard layer, or at the surface where the hardening ends. The fatigue strength at the transition to the core can be determined by fatigue tests as described above, provided that the fatigue initiation occurred at the transition to the core. Tests made with the core material only will not be representative since the tension residual stresses at the transition are lacking.

4 It has to be noted also what some recent research has shown:

The fatigue limit can decrease in the very high cycle domain with subsurface crack initiation due to trapped hydrogen that accumulates through diffusion around some internal defect functioning as an initiation point. In these cases, it would be appropriate to reduce the fatigue limit by some percent per decade of cycles beyond 10^7. Based on a publication by Yukitaka Murakami "Metal Fatigue: Effects of Small Defects and Non-metallic Inclusions" the reduction is suggested to be 5 % per decade especially when the hydrogen content is considered to be high.

Appendix 3 GUIDANCE FOR CALCULATION OF SURFACE TREATED FILLETS AND OIL BORE OUTLETS

1.1 Introduction

This appendix deals with surface treated fillets and oil bore outlets. The various treatments are explained and some empirical formulae are given for calculation purposes.
Please note that measurements or more specific knowledge is to be used if available. However, in the case of a wide scatter (e.g. for residual stresses) the values are to be chosen from the end of the range that would be on the safe side for calculation purposes.

2.1 Definition of Surface Treatment

"Surface treatment" is a term covering treatments such as thermal, chemical or mechanical operations, leading to inhomogeneous material properties - such as hardness, chemistry or residual stresses - from the surface to the core.

2.2 Surface Treatment Methods

The following list given in **Table 1** covers possible treatment methods and how they influence the properties that are decisive for the fatigue strength.

Table 1 Surface Treatment Methods and the Characteristics They Affect.

Treatment method	Affecting
Induction hardening	Hardness and residual stresses
Nitriding	Chemistry, hardness and residual stresses
Case hardening	Chemistry, hardness and residual stresses
Die quenching (no temper)	Hardness and residual stresses
Cold rolling	Residual stresses
Stroke peening	Residual stresses
Shot peening	Residual stresses
Laser peening	Residual stresses
Ball coining	Residual stresses

Note:
It is important to note that since only induction hardening, nitriding, cold rolling and stroke peening are considered relevant for marine engines, other methods as well as combination of two or more of the above are not dealt with in this appendix. In addition, die quenching can be considered in the same way as induction hardening.

3.1 Calculation Principles

3.1.1 General

1 The basic principle is that the alternating working stresses is to be below the local fatigue strength (including the effect of surface treatment) wherein non-propagating cracks may occur. (*See* also **6.1.2** for details) This is then divided by a certain safety factor.

2 This applies through the entire fillet or oil bore contour as well as below the surface to a depth below the treatment - affected zone - i.e. to cover the depth all the way to the core.

3 Consideration of the local fatigue strength is to include the influence of the local hardness, residual stress and mean working stress.

4 The influence of the “giga-cycle effect”, especially for initiation of subsurface cracks, is to be covered by the choice of safety margin.

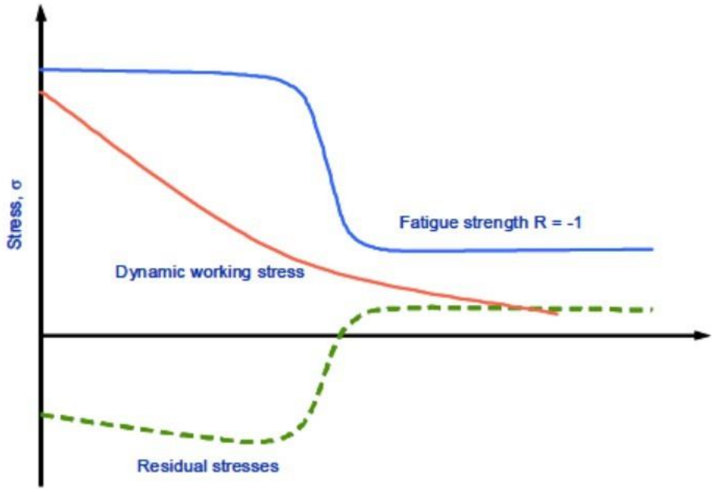
5 It is of vital importance that the extension of hardening/peening in an area with concentrated stresses be duly considered.

6 Any transition where the hardening/peening is ended is likely to have considerable tensile residual stresses. This forms a “weak spot” and is important if it coincides with an area of high stresses.

7 Alternating and mean working stresses are to be known for the entire area of the stress concentration as well as to a depth of about 1.2 times the depth of the treatment. (*See* **Fig. 1**)

8 The acceptability criterion is to be applied stepwise from the surface to the core as well as from the point of maximum stress concentration along the fillet surface contour to the web.

Fig. 1 Stresses as Functions of Depth, General Principles (In case of Induction Hardening)



Note:

The base axis is either the depth (perpendicular to the surface) or along the fillet contour.

3.2 Evaluation of Local Fillet Stresses

3.2.1 Evaluation Based upon FEM

It is necessary to have knowledge of the stresses along the fillet contour as well as in the subsurface to a depth somewhat beyond the hardened layer. Normally this will be found via FEA as described in **Appendix 3**. However, the element size in the subsurface range will have to be the same size as at the surface. For crankpin hardening only the small element size will have to be continued along the surface to the hard layer.

3.2.2 Evaluation Based upon a Simplified Approach

1 If no FEA is available, a simplified approach may be used. This can be based on the empirically determined stress concentration factors (SCFs), as in **1.4** of **Annex 2.3.1** if within its validity range, and a relative stress gradient inversely proportional to the fillet radius. Bending and torsional stresses are to be addressed separately. The combination of these is addressed by the acceptability criterion.

2 The subsurface transition-zone stresses, with the minimum hardening depth, can be determined by means of local stress concentration factors along an axis perpendicular to the fillet surface.

(1) Calculation of the local SCFs $\alpha_{B\text{-local}}$ and $\beta_{B\text{-local}}$ for bending in crankpin and journal fillets is as follows: (*See* **Fig. 2**)

$$\alpha_{B-local} = (\alpha_B - 1)\cdot e^{\frac{-2\cdot t}{R_H}} + 1 - \left(\frac{2\cdot t}{\sqrt{W^2+S^2}}\right)^{\frac{0.6}{\sqrt{\alpha_B}}}$$

$$\beta_{B-local} = (\beta_B - 1)\cdot e^{\frac{-2\cdot t}{R_G}} + 1 - \left(\frac{2\cdot t}{\sqrt{W^2+S^2}}\right)^{\frac{0.6}{\sqrt{\beta_B}}}$$

For parameters see **1.3.1-3** and **1.4** of **Annex 2.3.1**

(2) Calculation of the local SCFs $\alpha_{T\text{-local}}$ and $\beta_{T\text{-local}}$ for torsion in crankpin and journal fillets is as follows: (*See* **Fig. 3**)

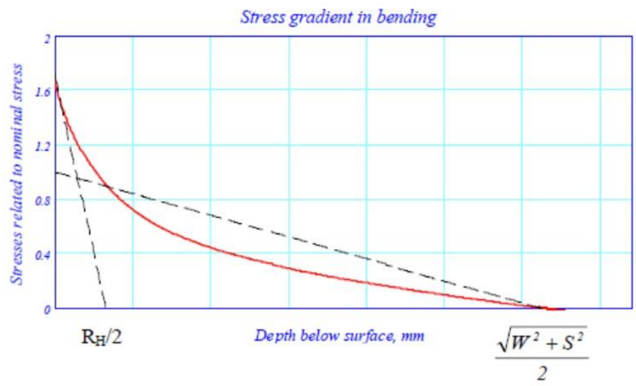
$$\alpha_{T-local} = (\alpha_T - 1) \cdot e^{\frac{-t}{R_H}} + 1 - \left(\frac{2 \cdot t}{D}\right)^{\frac{0.6}{\sqrt{\alpha_T}}}$$

$$\beta_{T-local} = (\alpha_T - 1) \cdot e^{\frac{-t}{R_G}} + 1 - \left(\frac{2 \cdot t}{D_G}\right)^{\frac{0.6}{\sqrt{\beta_T}}}$$

For parameters *see* **1.3.1-3** and **1.4** of **Annex 2.3.1**

3 If the pin is hardened only and the end of the hardened zone is closer to the fillet than three times the maximum hardness depth, FEA is to be used to determine the actual stresses in the transition zone.

Fig. 2 Bending SCF in the Crankpin Fillet as a Function of Depth.



Note:

The corresponding SCF for the journal fillet can be found by replacing R_H with R_G

Fig. 3 Torsional SCF in the Crankpin Fillet as a Function of Depth.

Note:

The corresponding SCF for the journal fillet can be found by replacing R_H with R_G and D with D_G

3.3 Evaluation of Oil Bore Stresses

3.3.1 Evaluation Based upon FEM

1 Stresses in the oil bores can be determined also by FEA.

2 The element size is to be less than 1/8 of the oil bore diameter D_O and the element mesh quality criteria are to be followed as prescribed in **Appendix 1**.

3 The fine element mesh is to continue well beyond a radial depth corresponding to the hardening depth.

4 The loads to be applied in the FEA are the torque and the bending moment, with four-point bending. (*See* **3.1.1** and **3.1.2** of **Appendix 1**)

3.3.2 Evaluation Based upon a Simplified Approach

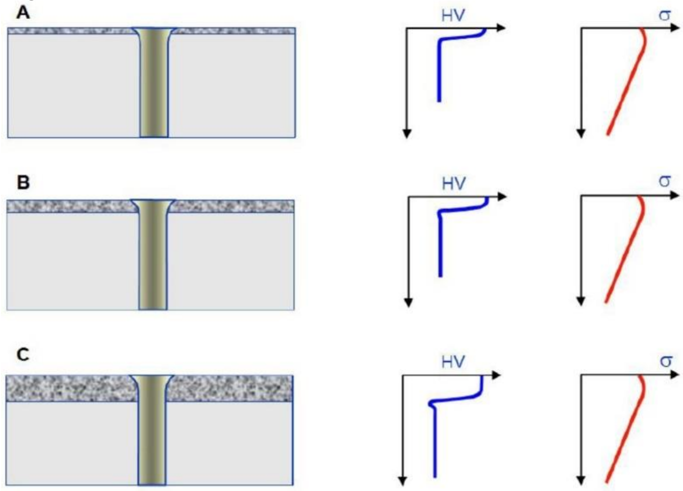
1 If no FEA is available, a simplified approach may be used. This can be based on the empirically determined SCF from **1.3** of **Annex 2.3.1** if within its applicability range.

2 Bending and torsional stresses at the point of peak stresses are combined as in **1.6** of **Annex 2.3.1**.

3 **Figure 4** indicates a local drop of the hardness in the transition zone between a hard and soft material. Whether this drop occurs depends also on the tempering temperature after quenching in the QT process.

4 The peak stress in the bore occurs at the end of the edge rounding. Within this zone the stress drops almost linearly to the centre of the pin. As can be seen from **Fig. 4**, for shallow (A) and intermediate (B) hardening, the transition point practically coincides with the point of maximal stresses. For deep (C) hardening the transition point comes outside of the point of peak stress and the local stress can be assessed as a portion $(1-2tH/D)$ of the peak stresses where tH is the hardening depth.

Fig. 4 Stresses and Hardness in Induction Hardened Oil Holes



5 The subsurface transition-zone stresses (using the minimum hardening depth) can be determined by means of local stress concentration factors along an axis perpendicular to the oil bore surface.

(1) Calculation of the local SCF $\gamma_{B-local}$ for bending in crankpin oil bores is as follows:

$$\gamma_{B-local} = (\gamma_B - 1) \cdot e^{\frac{-4t}{D_o}} + 1$$

For parameters *see* **1.3.1-3** and **1.4** of **Annex 2.3.1**

(2) Calculation of the local SCF $\gamma_{T-local}$ for torsion in crankpin oil bores is as follows:

$$\gamma_{T-local} = (\gamma_T - 1) \cdot e^{\frac{-2t}{D_o}} + 1$$

For parameters *see* **1.3.1-3** and **1.4** of **Annex 2.3.1**

3.4 Acceptability Criteria

The acceptability factors of crankpin fillets, journal fillets and the outlets of crankpin oil bores are to comply with the following criteria, which is specified in **1.8** of **Annex 2.3.1**:

$$Q \geq 1.15$$

This is to be extended to cover also surface treated areas independent of whether surface or transition zone is examined.

4.1 Induction Hardening

4.1.1 General

1 Generally, the hardness specification is to specify the surface hardness range i.e. minimum and maximum values, the minimum and maximum extension in or through the fillet and also the minimum and maximum depth along the fillet contour. The referenced Vickers hardness is considered to be *HV*0.5...*HV*5.

2 The induction hardening depth is defined as the depth where the hardness is 80 % of the minimum specified surface hardness.

3 In the case of crankpin or journal hardening only, the minimum distance to the fillet is to be specified due to the tensile stress at the heat-affected zone as shown in **Fig. 5**.

4 If the hardness-versus-depth profile and residual stresses are not known or specified, one may assume the following:

(1) The hardness profile consists of two layers (*See* **Fig. 6**):
  (a) Constant hardness from the surface to the transition zone
  (b) Constant hardness from the transition zone to the core material
(2) Residual stresses in the hard zone of 200 *MPa* (compression)
(3) Transition-zone hardness as 90 % of the core hardness unless the local hardness drop is avoided
(4) Transition-zone maximum residual stresses (von Mises) of 300 *MPa* (tension)

5 If the crankpin or journal hardening ends close to the fillet, the influence of tensile residual stresses has to be considered. If the minimum distance between the end of the hardening and the beginning of the fillet is more than 3 *times* the maximum hardening depth, the influence may be disregarded.

Fig. 5 Residual Stresses Along the Surface of a Pin and Fillet

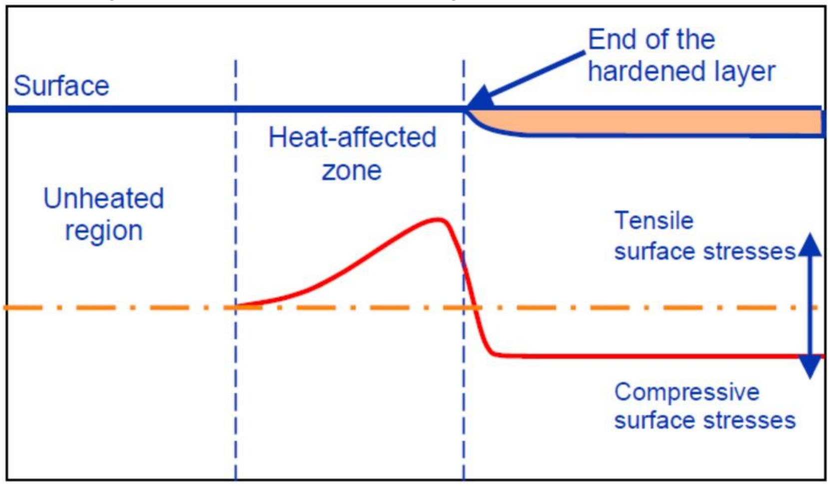
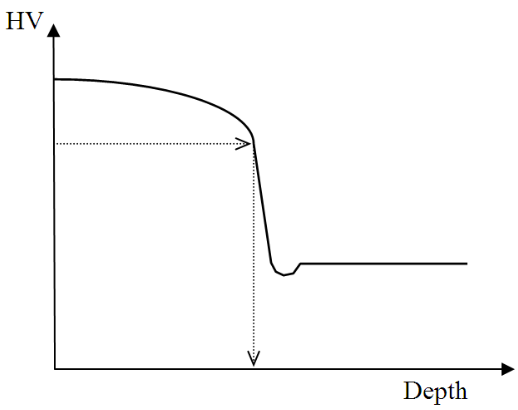


Fig. 6 Typical Hardness as a Function of Depth.



Note:
The arrows indicate the defined hardening depth. Note the indicated potential hardness drop at the transition to the core. This can be a weak point as local strength may be reduced and tensile residual stresses may occur.

4.2 Local Fatigue Strength

4.2.1 General

Induction-hardened crankshafts will suffer fatigue either at the surface or at the transition to the core.

4.2.2 Evaluation Based upon Fatigue Testing

1 The fatigue strengths, for both the surface and the transition zone, can be determined by fatigue testing of full-size cranks as described in **Appendix 2**.

2 In the case of a transition zone, the initiation of the fatigue can be either subsurface (i.e. below the hard layer) or at the surface where the hardening ends.

3 Tests made with the core material only will not be representative since the tensile residual stresses at the transition are lacking.

4.2.3 Evaluation Based upon Calculations

1 The surface fatigue strength can be determined empirically as follows:

$$\sigma_{Fsurface} = 400 + 0.5 \cdot (HV - 400) \ \ [MPa]$$

where

HV : surface Vickers hardness

The equation provides a conservative value, with which the fatigue strength is assumed to include the influence of the residual stress. The resulting value is valid for a working stress ratio of $R = -1$. It has to be noted also that the mean stress influence of induction-hardened steels may be significantly higher than that for QT steels.

2 The fatigue strength in the transition zone, without taking into account any possible local hardness drop, is to be determined by the following:

$$\sigma_{Ftransition,cpin} = \pm K \cdot (0.42 \cdot \sigma_B + 39.3) \cdot \left[0.264 + 1.073 \cdot Y^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{X}} \right]$$

where

$Y = D_G$, $X = R_G$ for journal fillet

$Y = D$, $X = R_H$ for crankpin fillet

$Y = D$, $X = D_O/2$ for oil bore outlet

For parameters *see* **1.4** of **Annex 2.3.1**

The influence of the residual stress is not included in the equation.

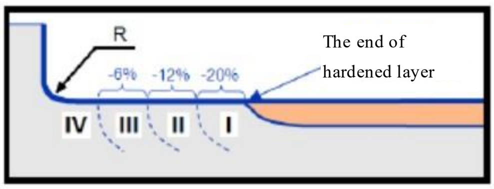
3 For the purpose of considering subsurface fatigue, below the hard layer, the disadvantage of tensile residual stresses has to be considered by subtracting 20 % from the value determined above. This 20 % is based on the mean stress influence of alloyed quenched and tempered steel having a residual tensile stress of 300 *MPa*.

4 When the residual stresses in **-3** are known to be lower, also smaller value of subtraction is to be used. For low-strength steels the percentage chosen is to be higher.

5 For the purpose of considering surface fatigue near the end of the hardened zone - i.e. in the heat-affected zone shown in the **Fig. 5** - the influence of the tensile residual stresses can be considered by subtracting a certain percentage, in accordance with **Table 2**, from the value determined by the above formula.

Table 2 The Influence of Tensile Residual Stresses at a Given Distance from the End of the Hardening towards the Fillet

Area	Distance from the end of the hardening towards the fillet	Ratio
I	0 to 1.0 of the max. hardening depth	20%
II	1.0 to 2.0 of the max. hardening depth	12%
III	2.0 to 3.0 of the max. hardening depth	6%
IV	3.0 or more of the max. hardening depth	0%



5.1 Nitriding

5.1.1 General

1 The hardness specification is to include the surface hardness range (min and max) and the minimum and maximum depth.

2 Only gas nitriding is considered.

3 The referenced Vickers hardness is considered to be HV 0.5.

4 The nitriding depth tN is defined as the depth to a hardness of 50 HV above the core hardness.

5 The hardening profile is to be specified all the way to the core.

6 If this is not known, it may be determined empirically via the following formula:

$$HV(t) = HV_{core} + \left(HV_{surface} - HV_{core}\right) \cdot \left(\frac{50}{HV_{surface} - HV_{core}}\right)^{\left(\frac{t}{t_N}\right)^2}$$

where:

t : The local depth

$HV(t)$: Hardness at depth t

HV_{core} : Core hardness (minimum)

$HV_{surface}$: Surface hardness (minimum)

tN : Nitriding depth as defined above (minimum)

5.2 Local Fatigue Strength

5.2.1 General

It is important to note that in nitrided crankshaft cases, fatigue is found either at the surface or at the transition to the core.

5.2.2 Evaluation Based on Fatigue Testing

The fatigue strength can be determined by tests as described in **Appendix 2**.

5.2.3 Evaluation Based on Calculations

1 Alternatively, the surface fatigue strength (principal stress) can be determined empirically and conservatively as follows:

$$\sigma_{Fsurface} = 450MPa$$

This is valid for a surface hardness of 600 HV or greater.

Note that this fatigue strength is assumed to include the influence of the surface residual stress and applies for a working stress ratio of $R = -1$.

2 The fatigue strength in the transition zone can be determined via the following formula:

$$\sigma_{Ftransition,cpin} = \pm K \cdot (0.42 \cdot \sigma_B + 39.3) \cdot \left[0.264 + 1.073 \cdot Y^{-0.2} + \frac{785 - \sigma_B}{4900} + \frac{196}{\sigma_B} \cdot \sqrt{\frac{1}{X}}\right]$$

where:

$Y = D_G$, $X = R_G$ for journal fillet

$Y = D$, $X = R_H$ for crankpin fillet

$Y = D$, $X = D_O/2$ for oil bore outlet

Note that this fatigue strength is not assumed to include the influence of the residual stresses.

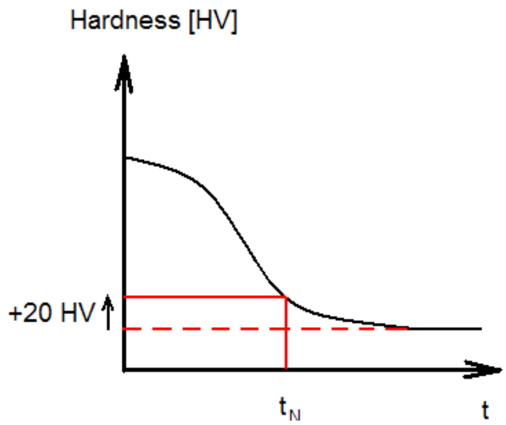
3 In contrast to induction-hardening the nitrited components have no such distinct transition to the core. Although the compressive residual stresses at the surface are high, the balancing tensile stresses in the core are moderate because of the shallow depth.

4 For the purpose of analysis of subsurface fatigue the disadvantage of tensile residual stresses in and below the transition zone may be even disregarded in view of this smooth contour of a nitriding hardness profile.

5 Although in principle the calculation is to be carried out along the entire hardness profile, it can be limited to a simplified approach of examining the surface and an artificial transition point. (*See* **Fig. 7**)

6 This artificial transition point can be taken at the depth where the local hardness is approximately 20 HV above the core hardness. In such a case, the properties of the core material are to be used. This means that the stresses at the transition to the core can be found by using the local SCF formulae mentioned in **3.2.2** or **3.3.2** when inserting $t = 1.2tN$.

Fig. 7 Sketch of the Location for the Artificial Transition Point in the Depth Direction



6.1 Cold Forming

6.1.1 General

1 The advantage of stroke peening or cold rolling of fillets is the compressive residual stresses introduced in the high-loaded area.

2 The fatigue strength has to be determined by fatigue testing (*See* also **Appendix 2**). Such testing is normally carried out as four-point bending, with a working stress ratio of $R = -1$.

3 From these results, the bending fatigue strength - surface - or subsurface-initiated depending on the manner of failure - can be determined and expressed as the representative fatigue strength for applied bending in the fillet.

4 In comparison to bending, the torsion fatigue strength in the fillet may differ considerably from the ratio $\sqrt{3}$ (utilized by the von Mises criterion). The forming-affected depth that is sufficient to prevent subsurface fatigue in bending, may still allow subsurface fatigue in torsion. Another possible reason for the difference in bending and torsion could be the extension of the highly stressed area.

5 The results obtained in a full-size crank test can be applied for another crank size provided that the base material (alloyed Q+T) is of the similar type and that the forming is done so as to obtain the similar level of compressive residual stresses at the surface as well as through the depth. This means that both the extension and the depth of the cold forming are to be proportional to the fillet radius.

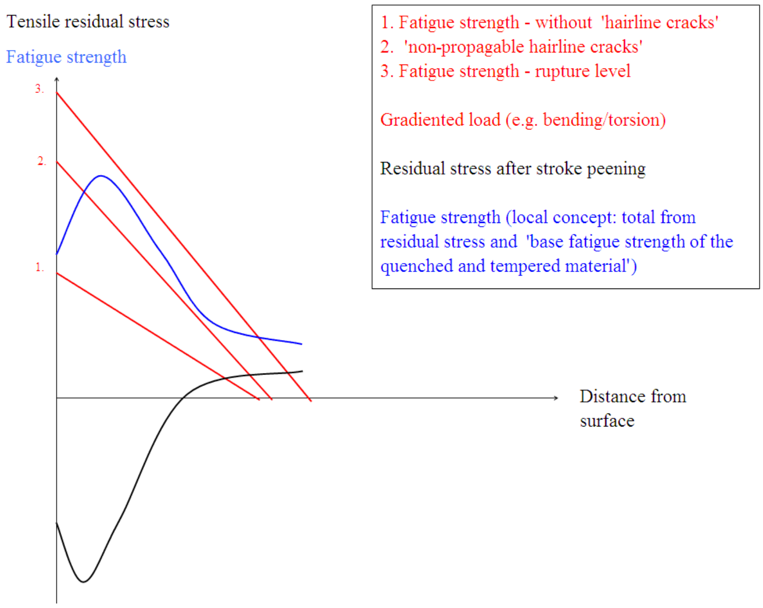
6.1.2 Stroke Peening by Means of a Ball

1 If both bending and torsion fatigue strengths have been investigated and differ from the ratio $\sqrt{3}$, the von Mises criterion is to be excluded.

2 If only bending fatigue strength has been investigated, the torsional fatigue strength is to be assessed conservatively. If the bending fatigue strength is concluded to be $x\,\%$ above the fatigue strength of the non-peened material, the torsional fatigue strength is not to be assumed to be more than 2/3 of $x\,\%$ above that of the non-peened material.

3 As a result of the stroke peening process the maximum of the compressive residual stress is found in the subsurface area. Therefore, depending on the fatigue testing load and the stress gradient, it is possible to have higher working stresses at the surface in comparison to the local fatigue strength of the surface. Because of this phenomenon small cracks may appear during the fatigue testing, which will not be able to propagate in further load cycles and/or with further slight increases of the testing load because of the profile of the compressive residual stress. Put simply, the high compressive residual stresses below the surface “arrest” small surface cracks. (*See* 2. in **Fig. 8**)

Fig. 8 Working and Residual Stresses below the Stroke-peened Surface.



Note:

Straight lines 1…3 represent different possible load stress gradients.

4 In fatigue testing with full-size crankshafts these small "hairline cracks" is to not be considered to be the failure crack. The crack that is technically the fatigue crack leading to failure, and that therefore shuts off the test-bench, is to be considered for determination of the failure load level. This also applies if induction-hardened fillets are stroke-peened.

5 In order to improve the fatigue strength of induction-hardened fillets it is possible to apply the stroke peening process in the crankshafts' fillets after they have been induction-hardened and tempered to the required surface hardness. If this is done, it might be necessary to adapt the stroke peening force to the hardness of the surface layer and not to the tensile strength of the base material.

6 The effect on the fatigue strength of induction hardening and stroke peening the fillets is to be determined by a full-size crankshaft test.

6.1.3 Use of Existing Results for Similar Crankshafts

The increase in fatigue strength, which is achieved by applying stroke peening, may be utilized in another similar crankshaft if all of the following criteria are fulfilled:

(1) Ball size relative to fillet radius within ± 10 % in comparison to the tested crankshaft

(2) At least the same circumferential extension of the stroke peening

(3) Angular extension of the fillet contour relative to fillet radius within ± 15 % in comparison to the tested crankshaft and located to cover the stress concentration during engine operation

(4) Similar base material, e.g. alloyed quenched and tempered

(5) Forward feed of ball of the same proportion of the radius

(6) Force applied to ball proportional to base material hardness (if different)

(7) Force applied to ball proportional to square of ball radius

6.1.4 Cold Rolling

1 The fatigue strength can be obtained by means of full-size crank tests or by empirical methods, if these are applied so as to be on the safe side.

2 If both, bending and torsion fatigue strengths have been investigated, and differ from the ratio $\sqrt{3}$, the von Mises criterion is to be excluded.

3 If only bending fatigue strength has been investigated, the torsional fatigue strength is to be assessed conservatively. If the bending fatigue strength is concluded to be x % above the fatigue strength of the non-rolled material, the torsional fatigue strength is to not be assumed to be more than 2/3 of x % above that of the non-rolled material.

6.1.5 Use of Existing Results for Similar Crankshafts

The increase in fatigue strength, which is achieved applying cold rolling, may be utilized in another similar crankshaft if all of the following criteria are fulfilled:

(1) At least the same circumferential extension of cold rolling

(2) Angular extension of the fillet contour relative to fillet radius within ± 15 % in comparison to the tested crankshaft and located to cover the stress concentration during engine operation

(3) Similar base material, e.g. alloyed quenched and tempered

(4) Roller force to be calculated so as to achieve at least the same relative (to fillet radius) depth of treatment

Appendix 4 GUIDANCE FOR CALCULATION OF STRESS CONCENTRATION FACTORS IN THE OIL BORE OUTLETS OF CRANKSHAFTS THROUGH UTILISATION OF THE FINITE ELEMENT METHOD

1.1 General

The objective of the analysis described in this appendix is to substitute the analytical calculation of the stress concentration factor (SCF) at the oil bore outlet with suitable finite element method (FEM) calculated figures. The former method is based on empirical formulae developed from strain gauge readings or photo-elasticity measurements of various round bars. In cases where these formulae are outside their applicable scope, the FEM-based method is to be used.

The SCF calculated in accordance with the rules set forth in this appendix is defined as the ratio of FEM-calculated stresses to nominal stresses calculated analytically. In use in connection with the present method in **Annex 2.3.1**, principal stresses are to be calculated.

The analysis is to be conducted as linear elastic FE analysis, and unit loads of appropriate magnitude are to be applied for all load cases.

It is advisable to check the element accuracy of the FE solver in use, e.g. by modelling a simple geometry and comparing the FEM-obtained stresses with the analytical solution.

A boundary element method (BEM) approach may be used instead of FEM.

2.1 Model Requirements

The basic recommendations and assumptions for building of the FE-model are presented in **2.1.1**. The final FE-model is to meet one of the criteria in **2.2**.

2.1.1 Element Mesh Recommendations

For the mesh quality criteria to be met, construction of the FE model for the evaluation of stress concentration factors in accordance with the following recommendations is advised:

(1) The model consists of one complete crank, from the main bearing centre line to the opposite side's main bearing centre line.

(2) The following element types are used in the vicinity of the outlets:
  (a) 10-node tetrahedral elements
  (b) 8-node hexahedral elements
  (c) 20-node hexahedral elements

(3) The following mesh properties for the oil bore outlet are used:
  (a) Maximum element size $a = r/4$ through the entire outlet fillet as well as in the bore direction (if 8-node hexahedral elements are used, even smaller elements are required for meeting of the quality criterion)
  (b) Recommended manner for element size in the fillet depth direction:
    i) First layer's thickness equal to element size of a
    ii) Second layer's thickness equal to element size of $2a$
    iii) Third-layer thickness equal to element size of $3a$

(4) The rest of the crank is to be suitable for numeric stability of the solver

(5) Drillings and holes for weight reduction have to be modelled

(6) Submodeling may be used as long as the software requirements are fulfilled.

2.1.2 Material

1 Material properties applied to steels as follows.

Young's Modulus : $E = 2.05 \cdot 10^5 MPa$

Poisson's ratio : $\nu = 0.3$

2 For materials other than steels, reliable values for material parameters have to be used, either as quoted in literature or as measured on representative material samples.

2.2 Element Mesh Quality Criteria

If the actual element mesh does not fulfil any of the following criteria in the area examined for SCF evaluation, a second calculation, with a finer mesh is to be performed.

2.2.1 Principal-stresses Criterion

The quality of the mesh is to be assured through checking of the stress component normal to the surface of the oil bore outlet radius. With principal stresses σ_1, σ_2 and σ_3 the following criterion is to be met:

$$\min(|\sigma_1|, |\sigma_2|, |\sigma_3|) < 0.03 \cdot \max(|\sigma_1|, |\sigma_2|, |\sigma_3|)$$

2.2.2 Averaged/Unaveraged-stresses Criterion

Unaveraged nodal stress results calculated from each element connected to a node is to differ less than 5 % from the 100 % averaged nodal stress results at this node at the location examined.

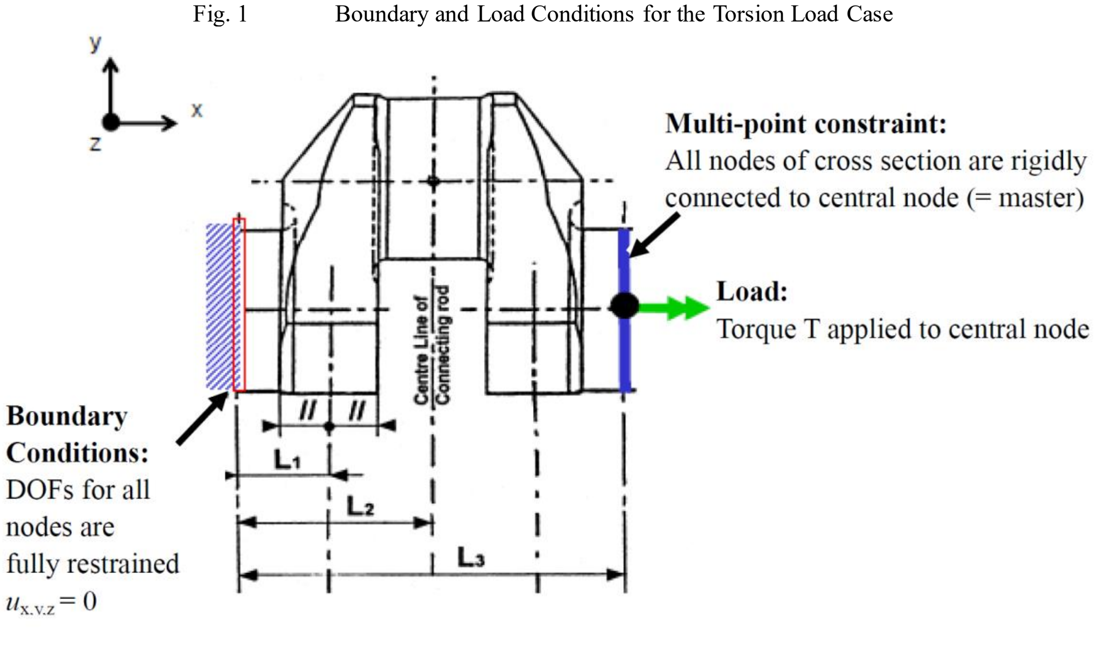
3.1 Load Cases and Assessment of Stress

The following load cases have to be calculated.

3.1.1 Torsion

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 1** where the torque is applied to the central node located at the crankshaft axis.

Fig. 1 Boundary and Load Conditions for the Torsion Load Case



2 For all nodes in an oil bore outlet, the principal stresses are obtained and the maximum value is taken for subsequent calculation of the SCF:

$$\gamma_T = \frac{\max(|\sigma_1|, |\sigma_2|, |\sigma_3|)}{\tau_N}$$

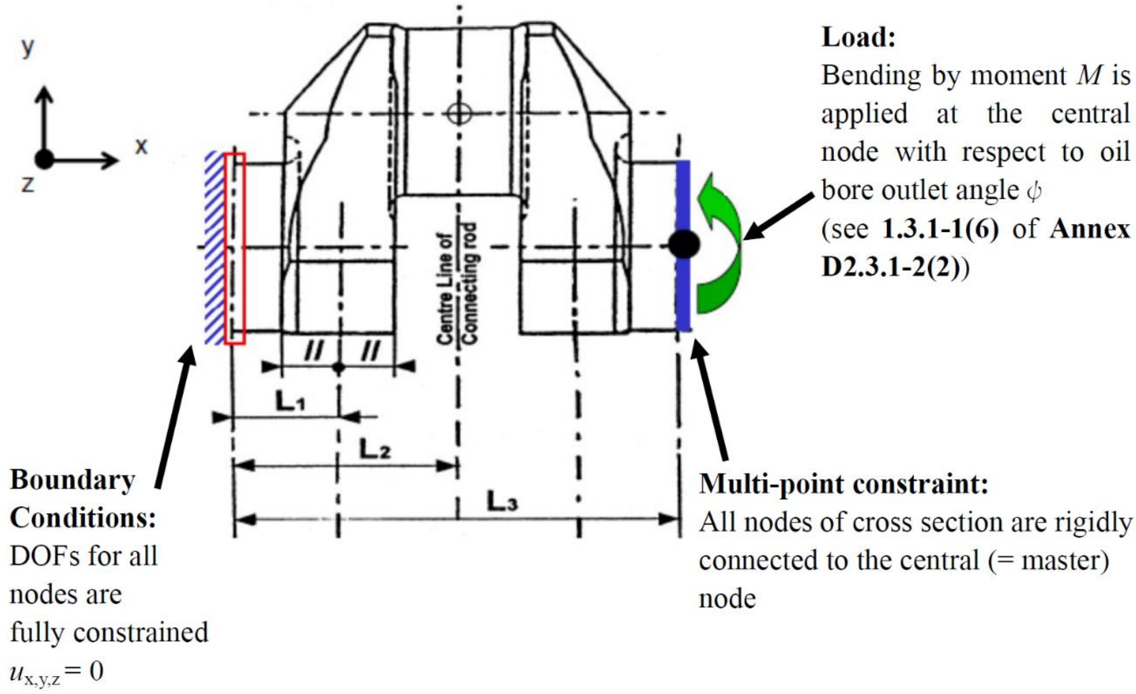
where the nominal torsion stress τ_N referred to the crankpin is calculated as follows (for W_P *see* **1.3.2** of **Annex 2.3.1**) :

$$\tau_N = \frac{T}{W_P}$$

3.1.2 Bending

1 Calculation is to be performed under the boundary and load conditions given in **Fig. 2** where the bending moment is applied to the central node located at the crankshaft axis.

Fig. 2 Boundary and Load Conditions for the Pure Bending Load Case



2 For all nodes in the oil bore outlet, principal stresses are obtained and the maximum value is taken for subsequent calculation of the SCF:

$$\gamma_B = \frac{\max(|\sigma_1|, |\sigma_2|, |\sigma_3|)}{\sigma_N}$$

where the nominal bending stress σ_N referred to the crankpin is calculated as follows (for W_e *see* **1.3.2** of **Annex 2.3.1**):

$$\sigma_N = \frac{M}{W_e}$$

Contents

GUIDANCE FOR THE SURVEY AND CONSTRUCTION OF STEEL SHIPS

Part D MACHINERY INSTALLATIONS

D1 GENERAL

D1.1 General

D1.1.1 Scope

1 In **Part D of the Rules**, "main propulsion machinery" means the following machinery which generates or converts motive power capable of propelling a ship at the speed specified in **2.1.8, Part A of the Rules**:

(1) Reciprocating internal combustion engines (including superchargers)

(2) Steam turbines (including main condensers)

(3) Gas turbines (including combustors)

(4) Generating plants for propulsion and motors for propulsion (excluding **Chapter 18, Part D of the Rules**)

2 Means provided to complement the motive power generated by the main propulsion machinery specified in **-1 that are connected directly to propulsion shafting systems**, are to be included in the shafting system. Any means not connected directly to propulsion shafting systems are to be included in auxiliary machinery essential for main propulsion.

3 The wording "other suitable rules" specified in **1.1.1-2 of the Rules** means the domestic law of the ship's flag state or other rules outside of domestic law which are deemed appropriate by the Flag Administration.

D1.1.4 Modification of Requirements

For those machinery installations specified in **1.1.4, Part D of the Rules** (excluding those specified in other parts of the Rules), some requirements of **Part D of the Rules** may be modified as follows:

(1) Prime movers (including power transmission systems and shafting systems) driving generators, auxiliary machinery essential for main propulsion and auxiliary machinery for manoeuvring and the safety.

(a) Prime movers with an output less than 100 kW

i) Submission of drawings may be omitted.

ii) Materials which comply with the requirements of any national standard may be accepted for the principal components. In this case, materials (excluding valves and pipe fittings) are to be manufactured by a manufacturer approved by the Society.

iii) Shop tests in the presence of the Surveyor may be substituted for manufacturer's tests. In this case, submission or presentation of test records may be required by the surveyor.

(b) Prime movers with an output not less than 100 kW but less than 375 kW

i) Materials used for principal components may be dealt with under the requirements specified in **(a)ii)**.

ii) Hydrostatic tests as well as dynamic balancing tests, overspeed tests and trial runs of turboblowers at the manufacturer may be dealt with under the requirements specified in **(a)iii)**.

(2) Prime movers (including power transmission systems and shafting systems) for auxiliary machinery for cargo handling.

(a) Prime movers with an output less than 375 kW may be dealt with under the requirements of **(1)(a)**.

(b) Prime movers with an output 375 kW or over may be dealt with under the requirements of **(1)(b)**.

(3) Fittings of boilers and pressure vessels of Groups I and II with a design pressure less than 3 MPa or with a nominal diameter less than 100 mm:

Materials which comply with any national standards may be accepted.

(4) For auxiliary machinery essential for main propulsion and auxiliary machinery for manoeuvring and safety:

(a) Hydrostatic tests may be dealt with under the requirements of **(1)(a)iii)**.

(b) For any asterisked auxiliary machinery in **Table D1.1.6-1**, operation tests may also be dealt with under the requirements of **(1)(a)iii)**.

(5) Auxiliary machinery for cargo handling:

(a) In cases where prime mover output is less than 375 *kW*, shop tests may be dealt with under the requirements of **(1)(a)iii)**.

(b) In cases where prime mover output is 375 *kW* or greater, shop tests excluding the operation test may be dealt with under the requirements of **(1)(a)iii)**.

(6) Pipes, valves and pipe fittings of piping systems with both a design pressure less than 1 *MPa* and a design temperature of 230 °C or less:

Hydrostatic tests may be dealt with under the requirements of **(1)(a)iii)** except for those valves and distance pieces directly fitted to the ship's side below the load line.

(7) Piping of Groups I and II, and their respective valves, pipe fittings and valves and pipe fittings which are directly fitted to the shell plating and collision bulkhead:

Materials which comply with any national standards may be accepted for the following **(a)** to **(d)**, except for those cast iron products for valves, seats and distance pieces mounted on the shell plating (including sea chests).

(a) Pipes with both a design pressure less than 1 *MPa* and a design temperature of 230 °C or less

(b) Valves and pipe fittings used for pipes with a nominal diameter less than 100 *mm*

(c) Valves and pipe fittings with both a design pressure less than 3 *MPa* and a design temperature of 230 °C or less

(d) Pipe flanges

(8) Hydraulic piping system excluding those for steering gears and controllable pitch propellers:

(a) For pipes with a nominal diameter less than 100 *mm*, materials which comply with any national standard may be accepted. In this case, materials are to be manufactured by the manufacturer approved by the Society.

(b) In radiographic inspection on butt welded pipe joints, sampling test under the instructions by the Surveyor of the Society may be accepted.

D1.1.6 Terminology

In the Rules, "auxiliaries" are classified as in **Table D1.1.6-1**.

Table D1.1.6-1 Kinds of Auxiliaries

Kind of auxiliary		Auxiliary machinery items
Auxiliary Machinery essential for main propulsion	Auxiliary machinery for cooling systems	Jacket cooling water pumps, Piston cooling water (oil) pumps, Fuel valve cooling water (oil) pumps, Turbocharger cooling water pumps, Circulating water pumps, Cooler cooling water pumps, Generator engine cooling water (oil) pumps, Air compressors cooling water pumps
	Auxiliary machinery for feed water, condensate and draining systems	Boiler water circulating pumps, Condensate pumps, Exhaust gas economizer feed pumps, Drain pumps, Feed water pumps
	Auxiliary machinery for fuel oil systems	F.O. supply (service) pumps, F.O. transfer pumps, Boiler burning pumps, F.O. purifiers
	Auxiliary machinery for lubricating oil systems	Cam shaft L.O. pumps, Turbocharger L.O. pumps, Crosshead L.O. pumps, Reduction gear L.O. pumps, Stern tube L.O. pumps (not applicable for gravitational circulation systems), L.O. purifiers
	Auxiliary machinery for hydraulic systems	Hydraulic oil pumps (pumps to supply hydraulic oil to hydraulic circuits for driving or controlling equipment relevant to main propulsion, e.g., controllable pitch propeller oil pumps)
	Other auxiliary machinery	Vacuum pumps for condensers, Gland steam exhaust fans, Boiler draught fans, Air compressors (excluding air compressors for emergency use), Distilling plants (when distillate is used for main boilers or other essential auxiliary boilers), Others as deemed essential by the Society.
Auxiliary machinery for manoeuvring and safety	Pumps	Bilge pumps (including pumps for oil-water separators*), Ballast pumps, Fire pumps* (including emergency fire pumps), Fuel oil supply pumps for gas combustion units (*GCU*s) of gas-fuelled ships
	Steering-related auxiliary machinery	Steering engines, Side thrusters*, Stabilizers
	Deck machinery	Windlasses, Mooring winches*, Hydraulic pumps used for windlasses, Hydraulic pumps used for mooring winches*
	Ventilating fans, blowers, etc.	Ventilating fans (installed in hazardous areas due to flammable gases or gases harmful to the health of personnel in engine room*, boiler room*, cargo oil pump room of oil tanker), Ventilating fans for cargo oil tanks, Gas-free fans and inert gas blowers of oil tanker, Blower fans for gas combustion units (*GCU*s) of gas-fuelled ships, Others as deemed essential by the Society.
Auxiliary machinery for cargo handling	Cargo handling machinery and gear	Hydraulic pumps used for Cargo handling appliances (items subject to “Rules for the Survey and Construction of Cargo Handling Appliances of Ships”), Hoisting machinery, Operating equipment
	Auxiliary machinery for specific use of oil tanker, ships carrying liquefied gages in bulk and ships carrying dangerous chemicals in bulk	Cargo pumps, Stripping pumps, Tank cleaning pumps, Gas compressors, Pumps used for gas cooling system, Gas refrigerating compressors, Fuel oil supply pumps and blower fans for gas combustion units (*GCU*s) of ships carrying liquefied gases in bulk
	Auxiliary machinery for cargo refrigerating installation	Compressors, Liquid pumps and Condenser cooling pumps used for cargo refrigerating machinery (including items subject to “Rules for the Survey and Construction of Cargo Refrigerating Installation of Ships”)
	Other auxiliary machinery	Others as deemed essential by the Society
Auxiliary machinery for specific use	Cargo handling equipment for specific Use	Unloaders (Shipborne units), Refrigerating machines for heat insulated containers, etc.
	Public working equipment	Dredging equipment, Drilling machines, Pile-driving equipment, etc.
	Fishing equipment	Winches, etc.

	Marine-products processing equipment	Canning/packing equipment, Conveyors, Ice-making machines, etc.
	Equipment for specific operations	Equipment specifically designated by the Society

Remarks:

For those items of auxiliary machinery marked by an asterisk, see **D1.1.4(4)**

D1.3 General Requirements for Machinery Installations

D1.3.1 General

1 The "navigable speed" referred to in **1.3.1-2, Part D of the Rules** means a speed at which the ship is capable of being steered and kept navigable for an extended period of time (period required to get to the nearest port for repairs). Normally, 7 *knots* or a speed corresponding to 1/2 of the speed specified in **2.1.8, Part A of the Rules** at the ship's full loaded draught may be regarded as a navigable speed.

2 The unconventional machinery referred to in **1.3.1-2, Part D of the Rules** is the machinery with novel design features (e.g. gas only engines) specified in **1.1.3, Part D of the Rules**.

3 Examples of starting arrangements for restoring propulsion from a dead ship condition are shown in **Fig. D1.3.1-1** to **Fig. D1.3.1-3**.

4 Dead ship condition means that all machinery installations, including their power supplies, are out of operation and that all auxiliary services, such as compressed air, starting current from batteries, etc., needed to bring these machinery installations back into operation are not available. However, the energy source for starting the emergency generator can be regarded as being available at the dead ship condition.

5 When designing and constructing machinery installations that are adequate for the service for which they are intended in accordance with **1.3.1-1, Part D of the Rules**, the properties (e.g. viscosity, cold flow property) of the fuel oils intended to be used by the machinery installations are to be taken into account, and fuel oil heaters and fuel oil coolers are to be provided when deemed necessary.

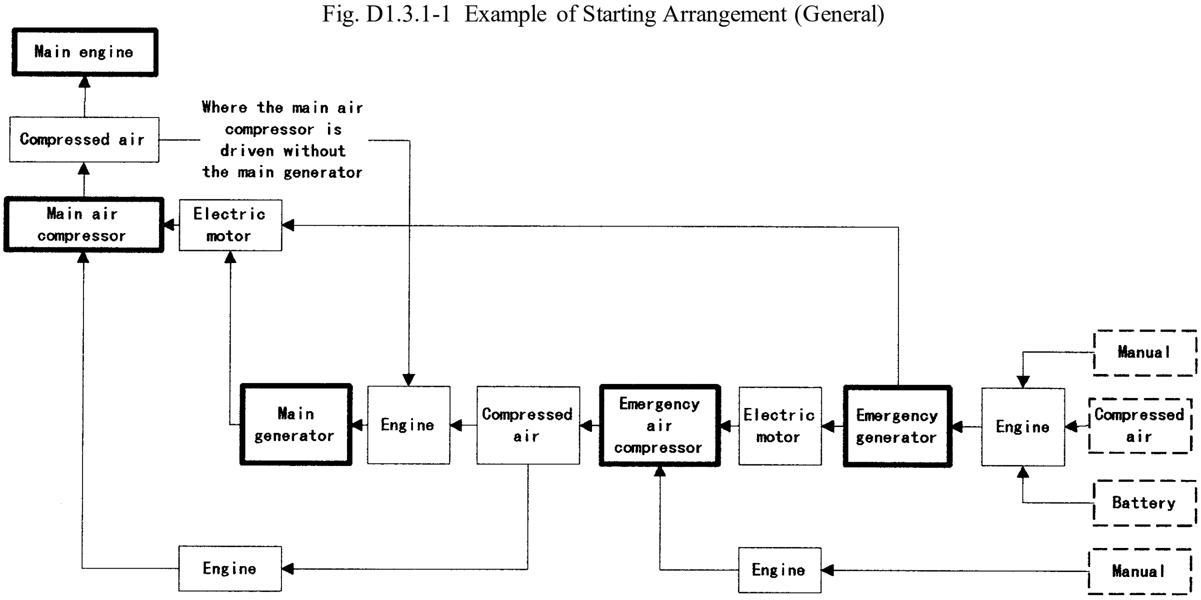
D1.3.7 Communication between the Navigating Bridge and Control Stations for the Speed and Direction of Thrust of the Propellers

1 It is recommended that engine room telegraph under the requirements of **1.3.7(1), Part D of the Rules** is such that it issues alarm upon loss of power supply.

2 The means of communication provided under the requirements of **1.3.7(2), Part D of the Rules** are to be capable of directly indicating orders and responses.

3 Engine room telegraph systems are to be provided independently from any remote control systems of main propulsion machinery on the navigation bridge; however, both systems may use a common handle.

Fig. D1.3.1-1 Example of Starting Arrangement (General)



Note:

Depending on the selected route, an emergency air compressor may not be provided.

Fig. D1.3.1-2 Example of Starting Arrangement (where One of Two Generating Sets is Driven by the Main Engine)

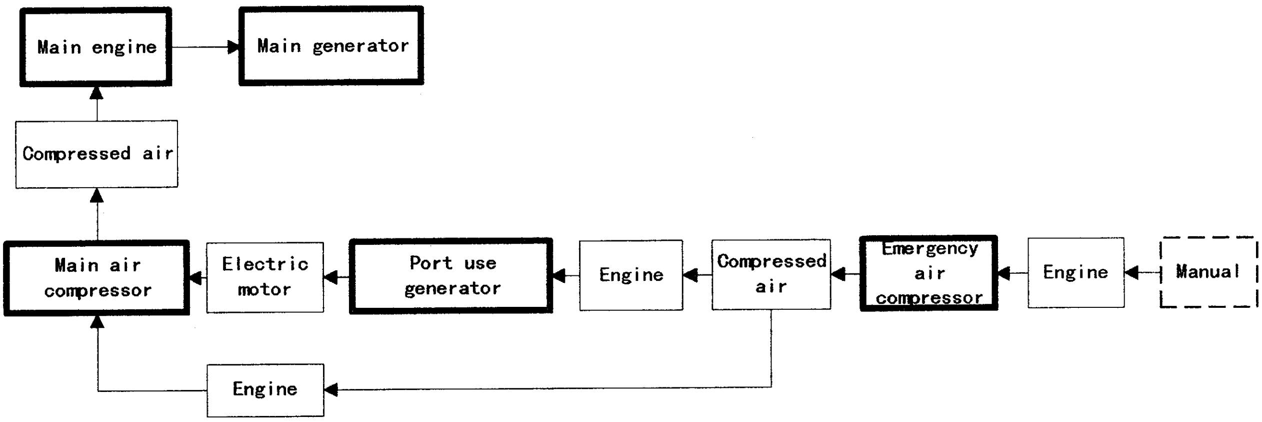
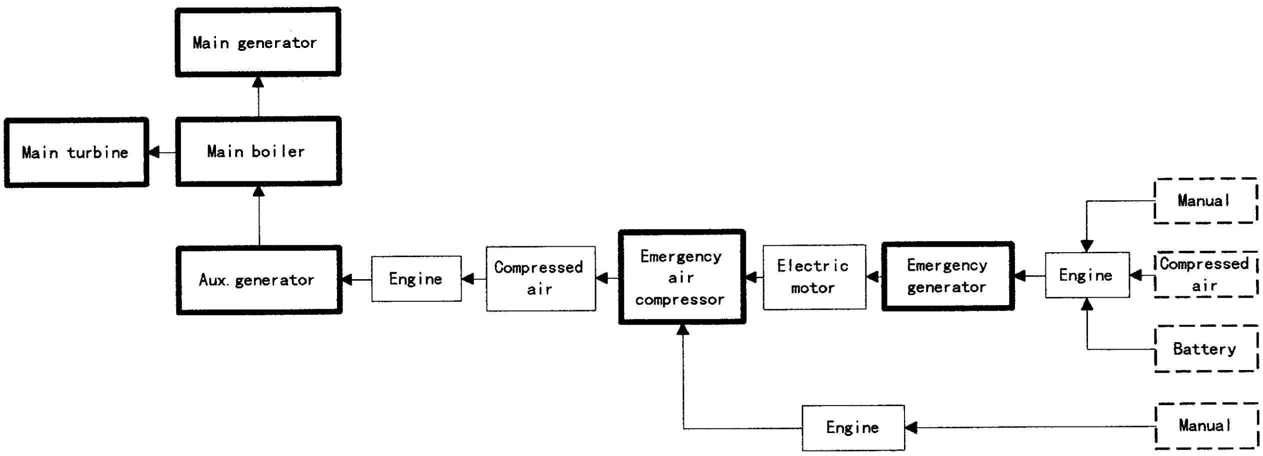


Fig. D1.3.1-3 Example of Starting Arrangement (for Steam Ships)



Note:

The time limit after blackout from a dead ship condition may be interpreted as time until light-off the first boiler.

D1.4 Tests

D1.4.1 Shop Tests

The wording "the survey methods which it considered to be appropriate" in **1.4.1-1, Part D in this Rules** means the survey methods which the Society considers to be able to obtain information equivalent to that obtained through traditional ordinary surveys where the Surveyor is in attendance, notwithstanding any of the requirements in this Part.

D1.4.4 Tests after Installation On Board

In "to be tested to the satisfaction of the Society at an appropriate time before being put into service in order to verify" referred to in **1.4.4-2, Part D of the Rules**, the following items are to be verified:

(1) The location of sea suction and overboard discharge pipes specified in **Chapter 13, Part D of the Rules**;

(2) Prevention of leakages of flammable gases and harmful gases, and prevention of fire;

(3) Insulation of highly heated parts and protection of moving parts; and

(4) Ventilation at the place of operation.

D2 RECIPROCATING INTERNAL CONBUSTION ENGINES

D2.1 General

D2.1.1 General

The wording "as specified separately by the Society" specified in **2.1.1-3, Part D of the Rules** means "in accordance with **Chapter 8, Part 6 of Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**".

D2.1.3 Drawings and Data

For the following data, those represented by two sizes in generic range of turbochargers (i.e. the same components, materials, etc., with the only difference being the size) are acceptable.

(1) The documentation for safe torque transmission specified in **(34)(a), Table D2.1(b), Part D of the Rules**

(2) The operation and maintenance manuals listed in **(34)(c), Table D2.1(b), Part D of the Rules**

D2.1.4 Approval of Reciprocating Internal Combustion Engines

1 In applying **2.1.4, Part D of the Rules**, reference for the approval procedures is to be made to **Fig. D2.1.4-1**.

2 The phrase "design approval is to be obtained as specified separately by the Society" specified in **2.1.4-1(1)(a), Part D of the Rules** means that the design approval and design appraisal are to be obtained in accordance with **Chapter 8, Part 6 of Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

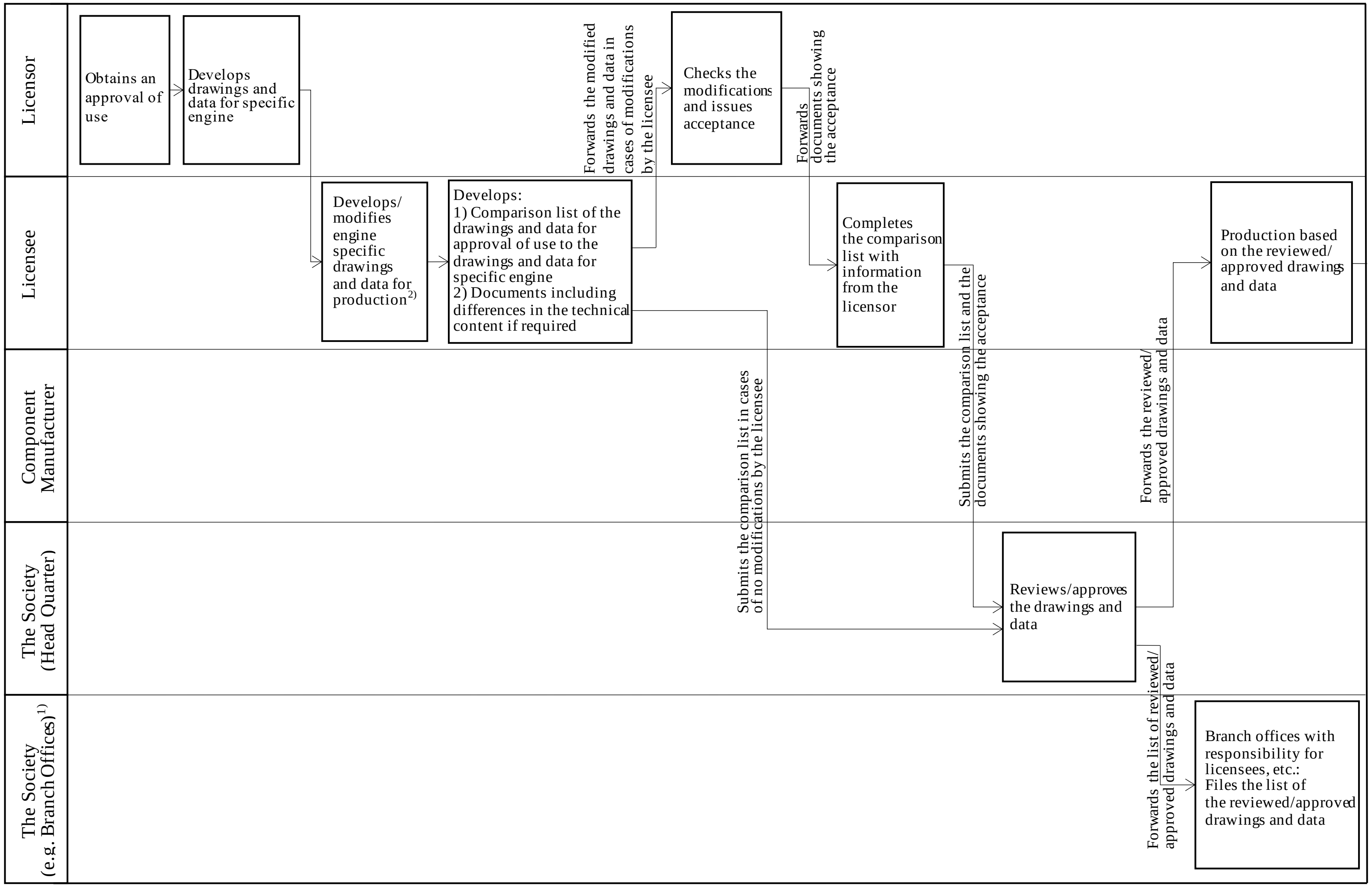
3 The wording "the drawings and data of the reciprocating internal combustion engine whose approval of use has been obtained" specified in **(1)(c)**, **(1)(d)**, **(2)(a)** and **(2)(b)** of **2.1.4-1, Part D of the Rules** means those listed in **8.2.2, Part 6 of Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

4 The wording "as specified separately by the Society" specified in **2.1.4-1(1)(d), Part D of the Rules** means "in accordance with **8.2.2-2, Part 6 of Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**".

5 In applying **2.1.4-1(2)(c), Part D of the Rules**, quality requirements specified by the licensor are to be satisfied.

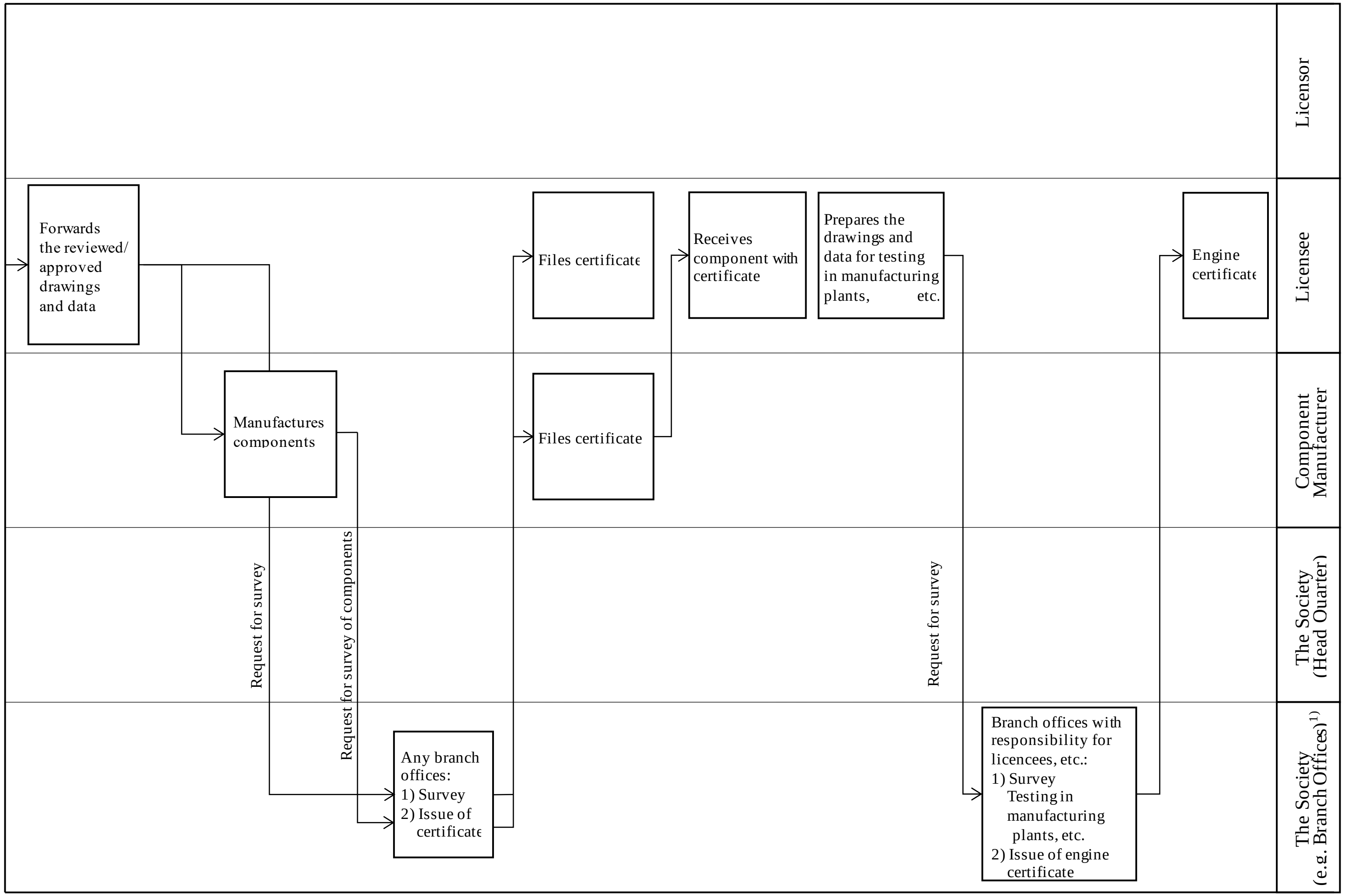
6 The wording "as specified separately by the Society" specified in **2.1.4-1(4)(a), Part D of the Rules** means "in accordance with **8.2.2-4, Part 6 of Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**".

Fig. D2.1.4-1 Flow of Approval of Reciprocating Internal Combustion Engines



1) Branch offices with responsibility for licensees and/or component manufacturers in different locations
2) In cases of modifications by the licensee, refer to **(b)** and **(c)** of **2.1.4-1(2), Part D of the Rules**

Fig. D2.1.4-1 Flow of Approval of Reciprocating Internal Combustion Engines (continued)



D2.3 Crankshafts

D2.3.1 Solid Crankshafts and Semi-Build Crankshafts

1 In applying **2.3.1-4, Part D of the Rules**, solid crankshaft and semi-built crankshaft approvals are to be according to the following.

2 The diameters of crankpins and journals are to be not less than the value given by the following formula:

$$d_c = \left\{\left(M + \sqrt{M^2 + T^2}\right) D^2\right\}^{\frac{1}{3}} K_m K_s K_h$$

where

d_c : Required diameter of crankshaft (mm)

M : $10^{-2} ALP_{max}$

T : $10^{-2} BSP_{mi}$

S : Length of stroke (mm)

L : Span of bearings adjacent to crank measured from centre to centre (mm)

P_{max}: Maximum combustion pressure in cylinder (MPa)

P_{mi} : Indicated mean effective pressure (MPa)

A and B :

Coefficients given in **Table D2.3.1-2(1)** to **D2.3.1-2(4)** for engines having equal firing intervals (in the case of Vee type engines, those with equal firing intervals on each bank.). Special consideration will be given to values A and B for reciprocating internal combustion engines having unequal firing intervals or for those not covered by the Tables.

D : Cylinder bore (mm)

K_m : Value given by the following **(1)** or **(2)** in accordance with the specified tensile strength of the crankshaft material. However, the value of K_m for materials other than steel forgings and steel castings is to be determined by the Society in each case.

(1) In cases where the specified tensile strength of material exceeds 440 N/mm^2

$$K_m = \sqrt[3]{\frac{440}{440 + \frac{2}{3}(T_s - 440)}}$$

where

T_s : Specified tensile strength of material (N/mm^2)

The value of T_s is not to exceed 760 N/mm^2 for carbon steel forgings and 1100 N/mm^2 for low alloy steel forgings.

(2) In cases where the specified tensile strength of material is not more than 440 N/mm^2 but not less than 400 N/mm^2

$K_m = 1.0$

K_s : Value given by the following **(1)**, **(2)**, or **(3)** in accordance with the manufacturing method of crankshafts.

(1) In cases where the crankshafts are manufactured by a special forging process approved by the Society as well as where the product quality is stable and the fatigue strength is considered to be improved by 20 % or more in comparison with that of the free forging process

$$K_s = \sqrt[3]{\frac{1}{1.15}}$$

(2) In cases where the crankshafts are manufactured by a manufacturing process using a surface treatment approved by the Society as well as where the product quality is stable and the fatigue strength is recognized as being superior

$$K_s = \sqrt[3]{\frac{1}{1 + \rho/100}}$$

where

ρ : Degree of improvement in strength approved by the Society relative to the surface hardening (%)

(3) In cases other than **(1)** and **(2)** above

$K_s = 1.0$

K_h : Value given by the following **(1)** or **(2)** in accordance with the inside diameter of the crankpins or journals.

(1) In cases where the inside diameter is one-third or more than that of the outside diameter

$$K_h = \sqrt[3]{\frac{1}{1-R^4}}$$

where

R : Quotient obtained by dividing the inside diameter of a hollow shaft by its outside diameter

(2) In cases where the inside diameter is less than one-third of the outside diameter

$K_h = 1.0$

Table D2.3.1-2(1) Value of Coefficients *A* and *B* for Single Acting In-line Engines

Number of cylinders	2-stroke cycle		4-stroke cycle	
	A	*B*	*A*	*B*
1		8.8		4.7
2		8.8		4.7
3		10.0		4.7
4		11.1		4.7
5		11.4		5.4
6	1.00	11.7	1.25	5.4
7		12.0		6.1
8		12.3		6.1
9		12.6		6.8
10		13.4		6.8
11		14.2		7.4
12		15.0		7.4

Table D2.3.1-2(2) Value of Coefficients *A* and *B* for Single Acting 2-stroke cycle Vee Type Engines with Parallel Connecting Rods

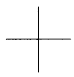
Number of cylinders	Minimum firing interval between two cylinders on one crankpin					
	45°		60°		90°	
	A	*B*	*A*	*B*	*A*	*B*
6	1.05	17.0	1.00	12.6	1.00	17.0
8		17.0		15.7		20.5
10		19.0		18.7		20.5
12		20.5		21.6		20.5
14		22.0		21.6		20.5
16		23.5		21.6		23.0
18		24.0		21.6		23.0
20		24.5		24.2		23.0

Table D2.3.1-2(3) Value of Coefficients A and B for Single Acting 4-stroke cycle Vee Type Engines with Parallel Connecting Rods

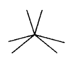
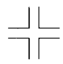
Number of cylinders	Minimum firing interval between two cylinders on one crankpin											
	45°		60°		90°		270°		300°		315°	
	A	B	A	B	A	B	A	B	A	B	A	B
6	1.60	4.1	1.47	4.0	1.40	4.0	1.40	4.0	1.30	4.4	1.20	4.3
8		5.5		5.5		5.5		5.5		5.3		5.2
10		6.7		7.0		6.5		6.5		6.1		5.9
12		7.5		8.2		7.5		7.5		6.9		6.6
14		8.4		9.2		8.5		8.5		7.5		7.3
16		9.3		10.1		9.5		9.5		8.2		7.9
18		10.1		11.1		10.5		10.5		8.8		8.5
20		11.5		14.0		11.5		11.5		9.5		9.2

Table D2.3.1-2(4) Values of Coefficients A and B (In cases of Unequal Firing Intervals)

(1) 4-stroke cycle in-line engines

Number of cylinders	Arrangement of crank	A	B
4		1.25	4.7

(2) 2-stroke cycle vee engines

Number of cylinders	Minimum firing interval between two cylinders on one crankpin	Arrangement of crank	A	B
12	60°		1.00	21.6
				15.0
16				26.3

3 In cases where the diameter of crankpins or journals is less than the required diameter d_c given in **-2** above, consideration will be given in each case on the basis of the stress levels in fillets, the torsional stress levels in crankpins and journals and the material of the crankshaft. In this connection, the stress levels in fillets are to be in accordance with the following:

In cases where the torsional stress in crankpins and journals are evaluated without carrying out a forced vibration calculation including the stern shaftings:

The diameter may be acceptable where the value of equivalent stress amplitude σ_e calculated by the **Annex D2.3.1** "**GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS**" is not more than the allowable stress σ obtained from the formula below with the coefficient shown in **Table D2.3.1-3.**

$\sigma = \sigma_a \cdot f_m \cdot f_s + \alpha$ (N/mm^2)

However, where deemed appropriate by the Society, the diameter in consideration of the allowable stress of crankshafts, including fillet parts, that have been hardened by surface treatment and the resultant stress distribution may be acceptable.

4 The dimensions of crank webs are to comply with the following requirements:

(1) The thickness and breadth of crank webs, the diameters of the crankpins and journals, are to comply with the conditions of the following formula. However, the thickness of crank webs is to be not less than 0.36 *times* the diameter of crankpins and journals. When the actual diameters of the crankpin and journal are larger than the required diameter of the crankshaft as determined by the formula in **-2**, the left side of the following formula may be multiplied by $(d_c/d_a)^3$.

$\{0.122(2.20 - b/d_a)^2 + 0.337\}(d_a/t)^{1.4} \leq 1$

where

b : Breadth of crank web (mm)

d_a : Actual diameter of crankpin or journal (mm)

t : Thickness of crank web (mm)

(2) The radius in fillets at the junctions of crank webs with crankpins or journals is to be not less than 0.05 *times* the actual diameter of crankpins or journals, respectively.

5 In cases where the dimensions of crankwebs fail to meet the requirements specified in **-4(1)** above, consideration will be given in accordance with the following:

(1) The dimensions of the crankwebs may be acceptable in cases where the actual diameters of crankpins and journals are not less than the required diameter d_c calculated by **-2** by replacing M and T with those specified below.

In this case, the dimensions are to be within the following ranges;

$0 \leq q/r \leq 1,\ -0.3 \leq h/d \leq 0.4,\ 8 \leq d/r \leq 27$

$1.1 \leq b/d \leq 2.1,\ 0.2 \leq t/d \leq 0.56$

$M = 10^{-2} A P_{\max} L \alpha_{KB}/5$

$T = 10^{-2} B P_{mi} S \alpha_{KT}/1.8$

where

α_{KB} : Stress concentration factor for bending, as specified below;

$\alpha_{KB} = 4.84 f_1 f_2 f_3 f_4 f_5$

$f_1 = 0.420 + 0.160\sqrt{d/r - 6.864}$

$f_5 = 1 + 81[0.769 - (0.407 - h/d)^2]$

$\times (q/r)(r/d)^2$

$f_3 = 0.285(2.2 - b/d)^2 + 0.785$

$f_4 = 0.444(d/t)^{1.4}$

$f_5 = 1 - [(h/d + 0.1)^2/(4t/d - 0.7)]$

$\cdots (t/d \geq 0.36)$

$= 1 - 1.35(h/d + 0.1)^2$

$\cdots (t/d < 0.36 \text{ and } h/d > -0.1)$

$= 1 \quad \cdots (t/d < 0.36 \text{ and } h/d \leq -0.1)$

α_{KT} : Stress concentration factor for torsion, as specified below;

$\alpha_{KT} = 1.75 g_1 g_2 g_3$

$g_1 = 31.6(0.152 - r/d)^2 + 0.67$

$g_2 = 1.04 + 0.317h/d$

$g_3 = 1.31 - 0.233b/d$

d : actual diameter of crankpin or journal (mm)

r : radius in fillet (mm)

q : recess (mm)

h : overlap between crankpin and journal (mm)

$h = (d_p + d_j - S)/2$

(2) In cases where the dimensions of the crankwebs fail to meet the requirements even after applying **(1)** above, the acceptance criteria specified below may be used:

In cases where the torsional stresses in crankpins and journals are evaluated without carrying out a forced vibration calculation including the stern shaftings:

The dimensions may be acceptable in cases where the value of the equivalent stress amplitude σ_e calculated by the **Annex D2.3.1** "**GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS**" is not more than the allowable stress σ obtained from the formula below with the coefficient shown in **Table D2.3.1-3**.

$\sigma = \sigma_a \cdot f_m \cdot f_s + \alpha$ (N/mm^2)

However, where deemed appropriate by the Society, the dimensions in consideration of the allowable stress of crankshafts, including fillet parts, that have been hardened by surface treatments and the resultant stress distribution may be acceptable.

6 The dimensions of the crankweb breadth b, crankweb thickness t, radius in fillet r and recess q used for **-4** and **-5** above are to be in accordance with the following (*See* **Fig. D2.3.1-1**):

(1) As for "b", the breadth on the perpendicular bisector of the line between the crankpin centre and journal centre is to be used.

(2) As for "t", the thickness at the same section specified in **(1)** is to be used. In this case, the recess q need not be accounted in the thickness even when it is provided.

(3) As for "r", the radius connecting to the crankpin or journal is to be used when a composite radius is provided.

7 Semi-built crankshafts are to be in accordance with **D2.3.2**.

Table D2.3.1-3 Coefficient of Allowable Stress at Fillet

<table>
<tr><td rowspan="5">σ_a (N/mm^2)</td><td rowspan="2">Stroke cycle of engine</td><td rowspan="2">Type of crankshaft</td><td colspan="3">Shaft diameter ϕ [1] (mm)</td></tr>
<tr><td>$d \geq 200$</td><td>$200 > d \geq 100$</td><td>$100 > d$</td></tr>
<tr><td rowspan="2">2-cycle</td><td>Semi-built-up</td><td>54</td><td>——</td><td>——</td></tr>
<tr><td>Solid</td><td>74</td><td>142-0.34d</td><td rowspan="2">108</td></tr>
<tr><td>4-cycle</td><td>Solid</td><td>83</td><td>133-0.25d</td></tr>
<tr><td>f_m</td><td colspan="5">$1 + \frac{2}{3}\left(\frac{T_s^{(2)}}{440} - 1\right)$</td></tr>
<tr><td rowspan="3">f_s</td><td colspan="5">Manufacturing method</td></tr>
<tr><td>Ordinary method</td><td colspan="2">Method (1) for K_s specified in D2.3.1-2</td><td colspan="2">Method (2) for K_s specified in D2.3.1-2</td></tr>
<tr><td>1</td><td colspan="2">1.15</td><td colspan="2">1+ ρ [3]/100</td></tr>
<tr><td rowspan="3">α (N/mm^2)</td><td colspan="5">Main bearing material</td></tr>
<tr><td colspan="3">White metal</td><td colspan="2">Aluminum or kelmet</td></tr>
<tr><td colspan="3">0</td><td colspan="2">10</td></tr>
</table>

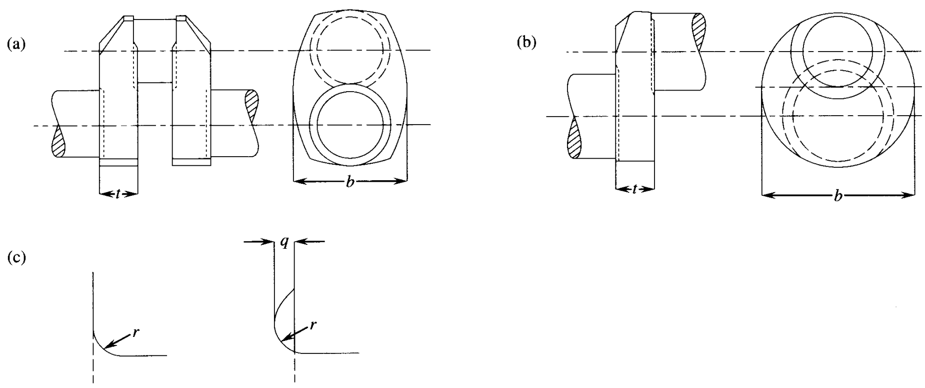
Notes:

(1) d is to be the actual diameter of crankpin or journal, whichever is larger.

(2) T_s signifies the minimum specified tensile strength (N/mm^2) of the crankshaft materials.
The limit of T_s for computing f_m is to be in accordance with the requirements in **D2.3.1-2**.

(3) ρ signifies the degree of strength improvement (%) approved by the Society relative to surface hardening.

Fig. D2.3.1-1 Dimensions for Webs of Solid Crankshafts



D2.3.2 Built-up Crankshafts

1 In applying **2.3.2, Part D of the Rules**, built-up crankshaft approval is to be in accordance with the followings.

2 The dimensions of crankpins and journals of built-up crankshafts are to comply with the following requirements in **(1)** and **(2)**:

(1) The diameters of crankpins and journals are to comply with the requirements in **D2.3.1-2**.

(2) The diameters of axial bores in journals are to comply with the following formula:

$$D_{BG} \leq D_S \cdot \sqrt{1 - \frac{4000 \cdot S_R \cdot M_{\max}}{\mu \cdot \pi \cdot D_S^2 \cdot L_S \cdot \sigma_{SP}}}$$

D_{BG} : Diameter of axial bore in journal (mm)

D_S : Journal diameter at the shrinkage fit (mm)

S_R : Safety factor against slipping (a value not less than 2 is to be taken)

M_{max} : Absolute maximum torque at the shrinkage fit ($N \cdot m$)

μ : Coefficient for static friction (a value not greater than 0.2 is to be taken)

L_S : Length of shrinkage fit (mm)

σ_{SP} : Minimum yield strength of material used for journal (N/mm^2)

3 The wording "maximum torque at the shrinkage fit" in **-2(2)** above means, in principle, $M_{T\max}$ shown in **1.3.2-1 of the Annex 2.3.1 Part D of the Rules** "CALCULATON METHOD OF CRANKSHAFT STRESS".

4 The dimensions of crank webs are to comply with the following requirements in **(1)** and **(2)**:

(1) The thickness of crank webs in way of the shrinkage fit is to comply with the following formula:

$$t \geq \frac{C_1 T D^2}{C_2 {d_h}^2} \frac{1}{\left(1 - \frac{1}{{r_s}^2}\right)}$$

$$t \geq 0.525 d_c$$

where

t : Thickness of crank web measured parallel to the axis (mm)

C_1 : 10 for 2-stroke cycle in-line engines / 16 for 4-stroke cycle in-line engines

T : Same as given in **D2.3.1-2**

D : Cylinder bore (mm)

C_2 : $12.8\alpha - 2.4\alpha^2$, but in the case of a hollow shaft, C_2 is to be multiplied by $(1 - R^2)$

$$\alpha = \frac{Shrinkage\ allowance(mm)}{d_h} \times 10^3$$

R : Quotient obtained by dividing the inside diameter of the hollow shaft by its outside diameter

d_h : Diameter of the hole at shrinkage fit (mm)

$$r_s = \frac{External\ diameter\ of\ web(mm)}{d_h}$$

d_c : Required diameter of crankshaft determined by the formula in **D2.3.1-2** (mm)

(2) The dimensions in fillets at the junctions of crank webs with crankpins of semi-built-up crankshafts are to comply with the requirements in **D2.3.1-4**.

5 In cases of built-up crankshafts, the value of α used in **-4(1)** is to be within the following range:

$$\frac{1.1Y}{225} \leq \alpha \leq \left(\frac{1.1Y}{225} + 0.8\right) \frac{1}{1 - R^2}$$

where

Y : Specified yield point of crank web material (N/mm^2)

R : Quotient obtained by dividing the inside diameter of the hollow shaft by its outside diameter

However, when the specified yield point of the crank web exceeds 390 N/mm^2 or the value obtained by the following formula is less than 0.1, the value used for α is to be approved by the Society.

where

$$\frac{S - d_p - d_j}{2 d_p}$$

S : Length of stroke (mm)

d_p : Diameter of the crankpin (mm)

d_j : Diameter of the journal (mm)

6 In cases where the dimensions of crankwebs fail to meet the requirements in **-4(1)**, they may be acceptable provided that either the following **(1)** or **(2)** is satisfied.

(1) In cases where the maximum torque at the shrinkage fit is evaluated without carrying out a forced vibration calculation including the stern shaftings:

$$d_h^2 t P_m \geq C T D^2$$

where

C : 103 for 2-stroke cycle in-line engines

165 for 4-stroke cycle in-line engines

P_m : Surface pressure at shrinkage fit, as given by the following formula

$$P_m = Y\left\{\log_e K + \frac{1}{2}\left(1 - \frac{K^2}{r_s^2}\right)\right\}(1 - R^2)$$

$$K = 0.9\sqrt{\frac{206\alpha}{Y} + 0.25}$$

(2) In cases where the maximum torque at the shrinkage fit is evaluated by carrying out a forced vibration calculation including the stern shaftings:

$$\alpha \geq \frac{4 \times 10^3 S_R M_{T\max}\left(1 - \frac{R^2}{r_s^2}\right)}{\pi \mu E d_h^2 t\left(1 - \frac{1}{r_s^2}\right)(1 - R^2)}$$

where

$M_{T\max}$: Maximum torque at shrinkage fit, as shown in **1.3.2-1 of the Annex 2.3.1 Part D of the Rules** "CALCULATION METHOD OF CRANKSHAFT STRESS" ($N \cdot m$)

E : Modulus of longitudinal elasticity (N/mm^2)

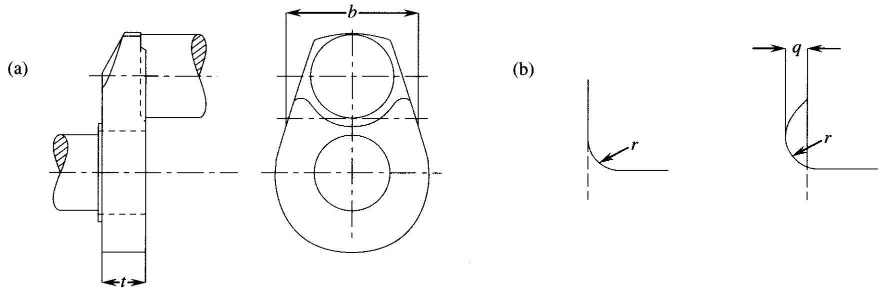
7 In cases where **-4(1)** is applied and where **D2.3.1-4** and **-5** is applied in accordance with **-4(2)** the dimensions of the crankweb breadth b, crankweb thickness t, radius in fillet r and recess q above are to be in accordance with the following (*See* **Fig. D2.3.2-1**):

(1) As for "b", the breadth on the line perpendicularly intersected to the line between the crankpin centre and journal centre and tangent to the crankpin is to be used.

(2) As for "t", the thickness at the same section specified in **(1)** is to be used. In this case, the recess q need not be accounted in the thickness even when it is provided, and the ring around the shrinkage hole is not to be included in the thickness.

(3) As for "r", the radius connecting to the crankpin or journal is to be used when a composite radius is provided.

Fig. D2.3.2-1 Dimensions for Webs of Semi-built-up Crankshafts



D2.3.3 Shaft Couplings and Coupling Bolts

1 The wording "as deemed appropriate by the Society" in **2.3.3-1, Part D of the Rules** is the value calculated by the formula in **D2.3.1-2** when the values of K_m, K_s and K_h are replaced with 1.0 (mm).

2 The wording "to be of sufficient strength" in **2.3.3-2, Part D of the Rules** means to be in accordance with the following **(1)** or **(2)**:

(1) The thickness of shaft coupling flanges at the pitch circle of the bolt holes is to be not less than the diameter of the bolts determined by the formula in **2.3.3-1, Part D of the Rules** by using 440 N/mm^2 for T_b. The radius at the fillet transition between the flange and shaft is to be not less than 0.08 *times* the shaft diameter. In this case, the fillet is not to be recessed in way of the

bolt heads and nuts.

(2) Detailed calculation sheets for the strength of couplings (for the procedures and contents of these calculations, the following **(a)** to **(f)** are to be considered as standards) are to be submitted to the Society for approval. In this case, it is to be verified that the thickness of the coupling flange is larger than the diameter of the bolts determined by the formula in **2.3.3-1, Part D of the Rules** using the tensile strength of the bolt material assumed to be equivalent to the tensile strength of the crankshaft material.

(a) With the procedures specified in the following **(b)** to **(f)**, it is to be verified that the stress at the coupling is less than the allowable value. As the stress value in this case, comparisons are to be made by applying appropriate safety factors for yield points for bending stress, bending fatigue limits, yield points for torsional stress and torsional fatigue limits of the crankshaft material considering four types of stress, such as the maximum bending stress, fluctuating bending stress, the maximum torsional stress and fluctuating torsional stress.

(b) The maximum bending moment and fluctuating bending moment of this portion are to be determined in accordance with the requirements specified in the **Annex D2.3.1** "GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS" or **Annex 2.3.1, Part D of the Rules** "CALCULATION METHOD OF CRANKSHAFT STRESS." Mean torque of this portion is to be determined.

(c) Torsional vibratory torque is to be determined by inverse operations from the allowable torsional vibratory stress value, which is to be taken as the fluctuating torque value.

By adding the fluctuating torque value, thus determined, to the mean torque value determined in the preceding sub-paragraph **(b)**, the sum is to be taken as the maximum torque value. (When the torsional vibratory torque value at this portion can be accurately determined through detailed torsional vibration calculations, the calculated torque may be used as the torsional vibratory torque value.)

(d) From the maximum bending moment and fluctuating bending moment of this portion, and the rigidity of the crankshaft, deflection angles of the crankshaft for respective cases are to be determined.

(e) Bending moments in magnitudes that cause the coupling flange of the crankshaft to assume the respective deflection angles determined in the preceding sub-paragraph **(d)** are to be determined, and the maximum bending stress and fluctuating bending stress of this portion are to be determined by dividing above by the section modulus of the coupling flange.

(f) Respective tangential forces are to be determined by dividing the maximum torque value and fluctuating torque value determined in the preceding sub-paragraph **(c)** by the diameter of the crankshaft at the root of the coupling flange. The maximum torsional stress and fluctuating torsional stress are to be determined by dividing the above tangential forces by the sectional area of the coupling flange (crankshaft diameter $\times \pi \times$ flange thickness) at the root, and by multiplying the stress concentration factor.

D2.4 Safety Devices

D2.4.1 Speed Governors and Overspeed Protective Devices

In applying **2.4.1-5(1)(c), Part D of the Rules**, the following throwing-on methods are considered acceptable.

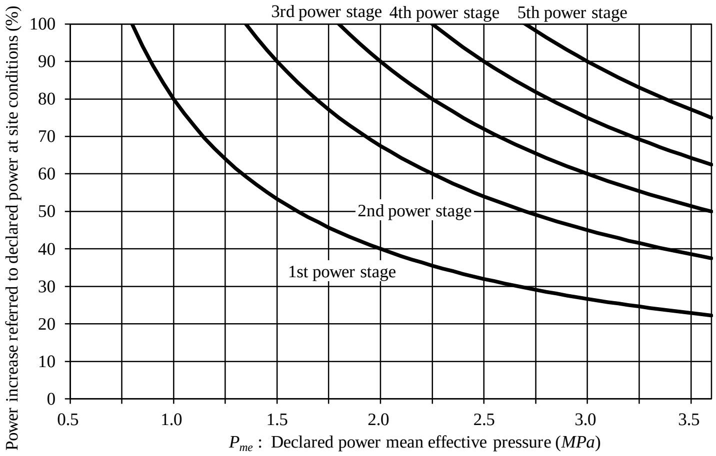
(1) Four-stroke diesel engines with mean effective pressures of 1.35 MPa or more (Refer to **Fig. D2.4.1**)

Total throw-on loads at the 1st power stage	(%) = $80/P_{me}$
Total throw-on loads at the 2nd power stage	(%) = $135/P_{me}$
Total throw-on loads at the 3rd power stage	(%) = $180/P_{me}$
Total throw-on loads at the 4th power stage	(%) = $225/P_{me}$
Total throw-on loads at the 5th power stage	(%) = $270/P_{me}$
Total throw-on loads at the 6th power stage	(%) = 100

(2) Gas-fuelled engines

Methods of throwing-on in steps decided by mutual agreement between manufacturer and user.

Fig. D2.4.1 Reference values for maximum possible sudden power increases as a function of brake mean effective pressure (P_{me}) at declared power (four-stroke diesel engines)



D2.4.3 Protection against Crankcase Explosion

1 The wording "explosion relief valves of approved type" in **2.4.3-1, Part D of the Rules** means those valves approved by the Society in accordance with **Chapter 10, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

2 The total volume of the stationary parts within the crankcase may be discounted in estimating the crankcase gross volume specified in **2.4.3-1, Part D of the Rules**. Rotating and reciprocating components are to be included in the gross volume.

3 The installation and maintenance manual specified in **2.4.3-1(5), Part D of the Rules** is to contain the following information:

(1) Description of valve with details of function and design limits

(2) Copy of type test certification

(3) Installation instructions

(4) Maintenance in service instructions to include testing and renewal of any sealing arrangements

(5) Actions required after a crankcase explosion

D2.4.5 Crankcase Oil Mist Detection Arrangements

1 The wording "devices as deemed appropriate by the Society" specified in **2.4.5-1, Part D of the Rules** means to the types of temperature monitoring devices for main bearings, crankpin bearings and crosshead bearings approved by the Society or equivalent devices.

2 The wording "crankcase oil mist detection arrangements required to be fitted to engines are to be approved type" stipulated in **2.4.5-2, Part D of the Rules** refers to crankcase oil mist detection arrangement approved in accordance with **Chapter 6, Part 7 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

D2.5 Associated Installations

D2.5.3 Starting Arrangements

The wording "effective arrangement to prevent the accumulation of combustibles" specified in **2.5.3-1(5), Part D of the Rules** means a construction of the starting air manifold from which combustibles can be easily (e.g. the starting air manifold has a gentle slope and a well at the lowermost position) discharged through a drain pipe which is normally open.

D2.6 Tests

D2.6.1 Shop Tests

1 The wording "a procedure deemed appropriate by the Society" in **2.6.1-2(6)(c), Part D of the Rules** means the tests specified in **8.5.2-2(10), Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

2 The wording "a procedure deemed appropriate by the Society" in **2.6.1-3(5), Part D of the Rules** means the tests specified in

8.3, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use.

3 In cases where the manufacturer has a quality system deemed appropriate by the Society, dynamic balancing tests specified in **2.6.1-6, Part D of the Rules** for category *B* turbochargers may be substituted by manufacturer tests. In such cases, the submission or presentation of test records may be required by the Society.

4 In cases where the manufacturer has a quality system deemed appropriate by the Society, the overspeed tests specified in **2.6.1-7, Part D of the Rules** for categories *B* turbochargers may be substituted for by manufacturer tests. In such cases, the submission or presentation of test records may be required by the Society.

5 The wording "a procedure deemed appropriate by the Society" in **2.6.1-7, Part D of the Rules** means the tests specified in **Chapter 11, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

D3 STEAM TURBINES

D3.4 Tests

D3.4.1 Shop Tests

Test runs of steam turbines for main propulsion machinery, including performance tests for safety devices, are to be carried out under Society approved test procedures.

D4 GAS TURBINES

D4.1 General

D4.1.3 Drawings and Data

"Documents containing strength considerations made for principal components" referred to in **4.1.3(2)(h), Part D of the Rules** are to include the following **(1)** and **(2)** documents:

(1) documents showing that mechanical stresses acting on principal components are clear based upon the results of stress analysis or from experimental values, etc. and it is ensured that there is an adequate safety factor for such stresses against the fatigue limit.

(2) documents showing that it has been verified for principle components on which mechanical stresses, thermal stresses, creeps, relaxations, etc. or any combination thereof is applied and that stresses corresponding to differential stresses between those in the static condition of the gas turbine at ambient temperature and those in the condition in which the gas turbine is operating at the maximum continuous output.

D4.2 Materials, Construction and Strength

D4.2.2 Construction and Installations

1 The restart of gas turbines used as main propulsion machinery specified in **4.2.2-4, Part D of the Rules** does not require an automatic restarting function.

2 The phrase "installed so that (...) does not endanger persons and machinery in the vicinity of the gas turbine" specified in **4.2.2-6, Part D of the Rules** means that the following **(1)** to **(3)** are, as far as possible, to be located outside of the plane of high speed rotating parts of the gas turbine, taking into account those cases where the casing is unable to contain a blade or another principal component, or any debris in the event of the loss of a blade or the failure of such a principle component.

(1) Fuel oil, lubricating oil and other systems having a fire risk

(2) Fire detection and alarm systems as well as fire extinguishing systems

(3) Areas normally manned in the compartment where the gas turbine is installed

D4.4 Associated Installations

D4.4.2 Exhaust Gas Arrangement, etc.

The "other hot surfaces" referred to in **4.4.2-3, Part D of the Rules** means, for example, surfaces of piping systems supplying compressed air extracted from a part between compressors or the compressor outlet, or high temperature gas extracted from the turbine inlet or a part between turbines; to the outside.

D4.4.3 Starting Arrangements

1 In cases where the "means" specified in **4.4.3-1, Part D of the Rules** is automatic, fuel oil systems, lubricating oil systems and cooling systems, etc. are to be designed so that they can be operated sequentially according to a pre-determined programme when the engine starts or stops. Regarding the sequence and operation related to these systems, attention is to be paid to the following **(1)** to **(7)**:

(1) Lubricating oil pumps are to be in operation before the starting-up and after the stopping of any engine. However, this requirement may be dispensed with in cases where the engine is equipped with roller bearings and the lubricating oil pumps are being driven by the engine.

(2) Combustion chambers are to be pre-purged by a sufficient volume of air before ignition.

(3) The opening of the main fuel valve is not to precede the ignition spark.

(4) The ignition period of each burner (after the main fuel valve has been opened, in the event of ignition failure, the amount of time until the valve is closed.) is not to exceed a pre-determined length of time. The engine starting trial is to be halted in cases where the engine does not start within such a pre-determined time period.

(5) Excessive fuel is not to be supplied to combustion chambers during ignition.

(6) After shutting off the fuel valves, a suitable measure is to be taken to prevent any abnormal combustion or ignition trouble at times of restarting. For example, this could be achieved by opening the drain valves located at positions between the fuel oil shut off valve and the fuel nozzle.

(7) Starting devices are to be disconnected from gas generators after their running becomes self-sustaining.

2 In cases where the "reservoirs" specified in **4.4.3-2(2), Part D of the Rules** are utilized for the "purging" specified in **4.4.3-1(2), Part D of the Rules**, the total capacity of the reservoirs is to be such that a capacity necessary for the purging is added.

D4.4.5 Fuel Oil Arrangements

The "sufficient consideration" referred to in **4.4.5-1, Part D of the Rules** means that provisions such as those in accordance with the following **(1)** and **(2)** are made.

(1) At least two filters are to be fitted in the fuel oil supply lines to the gas turbine and be so arranged that any filter may be cleaned without interrupting the supply of filtered fuel oil to the gas turbine.

(2) Fuel treatment systems, including filtration and centrifuging devices, are to be provided so as to control the level of water and particulate contamination within the limits specified by the manufacturer of the gas turbine.

D4.5 Tests

D4.5.1 Shop Tests

Manufacturers of gas turbines are to submit shop trial plans, according to *JIS B* 8041 or *ISO* 2314 as a standard, for Society approval and all shop trials are to be carried out in accordance with such approved plans.

D5 POWER TRANSMISSION SYSTEMS

D5.2 Materials and Construction

D5.2.3 General Construction of Gearings

The words "enough" and "sufficient" referred to in **5.2.3, Part D of the Rules** mean being designed in accordance with national or international standards such as *JIS*.

D5.2.4 General Construction of Power Transmission Systems other than Gearings

1 The wording "having sufficient strength against transmitted power" in **5.2.4-1, Part D of the Rules** means complying with the following requirements in addition to the requirements in **8.2.4-3, Part D of the Rules**:

Flexible couplings used in main propulsion shafting systems are to withstand the torque (T) calculated by the following formula:

$$T = 3.0 \times 10^4 \frac{H}{N_0}$$

where

T : Maximum allowable torque of flexible couplings ($N \cdot m$)

H : Maximum continuous output of main propulsion machinery (kW)

N_0 : Rotational speed (rpm) of flexible couplings at maximum continuous output of main propulsion machinery

2 The wording "heating due to hysteresis" in **5.2.4-1, Part D of the Rules** means to give consideration to any heat built-up by power loss of rubber, etc. with respect to the strength of rubber couplings.

3 The wording "deemed appropriate by the Society" in **5.2.4-2, Part D of the Rules** means as follows:

(1) Electromagnetic couplings are to be constructed in a manner that permits the periodical inspection of the clearance of the magnetic circuits, and are to be provided with gauges of adequate graduations necessary for inspection.

(2) Electromagnetic couplings are at least to be of drip-proof construction; and, in cases other than the enclosed-type constructions, means are to be provided to prevent operators from coming into contact with any rotating units as well as preventing any foreign matter from entering the inside.

4 The wording "to be constructed so that inspections can be performed as easily as possible" in **5.2.4-4, Part D of the Rules** means to be constructed so that an external inspection of rubber elements and a measurement of surface hardness or permanent deformation may be easily made. For this purpose, ships are to be provided with gauges for measuring surface hardness or permanent deformation of rubber elements.

D5.3 Strength of Gears

D5.3.1 Application

In the case of bevel gear, the wording "deemed appropriate by the Society" in **5.3.1, Part D of the Rules** means as follows:

(1) The bending stress at the root sections of gear teeth and limiting tooth surface stress are to be according to ISO standards or as deemed appropriate by the Society.

(2) Strength of the interior of gear teeth

The Vickers hardness (HV) of the interior of gear teeth is not to be less than the value calculated by the following formula. However, this requirement does not apply to bevel gears for which the tip diameter (outer end) is smaller than 1,100 mm:

If $\frac{z}{w} < 0.79$ then $\frac{z}{w}$ is to be taken as 0.79.

$$HV = 1.11 S_H p \left[\frac{z}{w} - \frac{\left(\frac{z}{w}\right)^2}{\sqrt{1 + \left(\frac{z}{w}\right)^2}} \right]$$

HV : Vickers hardness

S_H : Safety factor for contact stress, is to comply with the requirements in **1.6.3-9 of the Annex 5.3.1 Part D of the Rules** "CALCULATION OF STRENGTH OF ENCLOSED GEARS".

p : Real hertzian stress (*MPa*). The upper limit of the value of p used in this calculation is to be 1,500 *MPa.*

$$p = AS_c$$

S_c: Contact stress (*MPa*), to be calculated according to *ISO 10300* standards.

A: If S_c is calculated according to *ISO 10300* standards, then the coefficients are to be determined, in consideration of analysis results, by the Society on a case by case basis. In addition, if S_c is calculated according to *ISO 10300* standards, A is to taken as 1.32

w: Half the hertzian contact width (*mm*), to be calculated by the following formula:

$$w = \frac{p\rho_C}{56300}$$

$$\rho_C = \frac{\rho_1 \rho_2}{\rho_1 + \rho_2}$$

$$\rho_1 = 0.5d_{vn1}\sin\alpha_n$$

$$\rho_2 = 0.5d_{vn2}\sin\alpha_n$$

$$d_{vn1} = d_{m1}\frac{\sqrt{1+u^2}}{u}\frac{1}{\cos^2\beta_{vb}}$$

d_{m1} : Mean pitch diameter of pinion (*mm*)

u : Gear ratio

$$\beta_{vb} = \arcsin(\sin\beta_m \cos\alpha_n)$$

β_m : Mean spiral angle

α_n : Normal pressure angle

$$d_{vn2} = u^2 d_{vn1}$$

z : Depth from teeth surface to evaluation point (*mm*)

D5.3.3 Allowable Tangential Loads for Bending Strength

Tangential loads for the bending strength of internal cylindrical gears with involute teeth are to comply with the following conditions:

$$P_{MCR} \leq 47.6(K_1 S_b - K_2)K_3 m_n$$

Symbols in the formula are the same as in **5.3.3, Part D of the Rules**.

D5.3.4 Tangential Loads for Surface Strength

Tangential loads for the surface strength of internal cylindrical gears with involute teeth, except for any reversing gears for astern operation, are to comply with the following conditions:

$$P_{MCR} \leq 9.81(K_1 S_s - K_2)K_3 K_4 \frac{i}{i-1} D_1$$

Symbols in the formula are the same as in **5.3.4, Part D of the Rules**.

D5.4 Gear Shafts and Flexible Shafts

D5.4.1 Gear Shafts

The word "sufficient" in **5.4.1-1(2), Part D of the Rules** means being designed in accordance with national or international standards such as *JIS*.

D5.4.3 Couplings and Coupling Bolts

The word "sufficient" in **5.4.3, Part D of the Rules** means being designed in accordance with national or international standards such as *JIS*.

D6 SHAFTINGS

D6.1 General

D6.2 Materials, Construction and Strength

D6.2.4 Propeller Shafts and Stern Tube Shafts

1 As for the diameter of propeller shaft Kind 2 or stern tube shafts Kind 2 made of carbon steel or low alloy steel, the wording "to be deemed appropriate by the Society" means to calculate the required diameter by the following formula:

$$d_s = 100k_3\sqrt[3]{\frac{H}{N_0}}$$

d_s : Required diameter of propeller shaft (mm)

H : Maximum continuous output of main propulsion machinery (kW)

N_0 : Number of revolutions of shaft at maximum continuous output (rpm)

k_3 : Factor concerning shaft design, given in **Table D6.2.4-1**

2 The value of k_3 for propeller shafts and stern tube shafts made of stainless steel forgings, etc. other than those indicated in the **Table D6.4, Part D of the Rules** which is for k_3 specified in **6.2.4-2, Part D of the Rules**, is to be in accordance with **Table D6.2.4-2**. Furthermore, this requirement may be applied to propeller shafts Kind 2 and stern tube shafts Kind 2.

Table D6.2.4-1 Values of k_3

	Application	k_3
1	The portion from the big end of the tapered part of a propeller shaft (in the case of a flange connected propeller, the forward end of the of the flange) to the forward end of the after most stern tube bearing or to 2.5 d_s, whichever is larger	1.33
2	Excluding any portion specified in 1 above, the portion in the direction toward the bow side up to the forward end of the forward stern tube sealing assembly	1.21(1)
3	The portion between the forward end of the forward stern tube sealing assembly and the intermediate shaft coupling	1.21(2)

Notes :

(1) The diameter of the boundary portion is to be reduced with either a smooth taper or a blending radius nearly equal to the change in diameter.

(2) The diameter may be reduced in accordance with **6.2.4.-3, Part D of the Rules**.

Table D6.2.4-2 Values of k_3

Application		Shaft material	
		Austenitic stainless steel with 0.2 % proof stress not less than 205 N/mm^2	Precipitation hardened martensite stainless steel with 0.2 % proof stress not less than 400 N/mm^2
1	The portion from the big end of the tapered part of a propeller shaft (in the case of a flange connected propeller, the forward end of the flange) to the forward end of the after most stern tube bearing or to 2.5 d_s, whichever is larger	1.28	1.05
2	Excluding the portion shown in 1 above, the portion in the direction toward the bow side up to the forward end of the forward stern tube sealing assembly	1.16(1)	0.94(1)
3	The portion between the forward end of the forward stern tube sealing assembly and the intermediate shaft coupling	1.16(2)	0.94(2)

Notes :

(1) The diameter of the boundary portion is to be reduced with either a smooth taper or a blending radius nearly equal to the change in diameter.

(2) The diameter may be reduced in accordance with **6.2.4-3, Part D of the Rules**.

D6.2.6 Detailed Evaluation for Strength

1 In cases where stresses acting on the shaft simultaneously satisfy the conditions below, the shaft diameter may be accepted as being equivalent to those required by the formulae specified in **Chapter 6, Part D of the Rules**.

$$\beta_m \tau_m + \beta_t \tau_D \leq \frac{\tau_y}{S_y}$$

$$\beta_t \tau_D \leq \frac{\tau_f'}{S_f}$$

τ_m : Mean torsional stress acting on the shaft. In cases where mean bending stress acts simultaneously, τ_{me} given by the following formula is to be used: (N/mm^2)

$$\tau_{me} = \sqrt{\tau_m^2 + \frac{1}{3}\sigma_m^2}$$

τ_{me} : Equivalent mean torsional stress (N/mm^2)

σ_m : Mean bending stress (N/mm^2)

β_m : Notch factor against static stress

τ_D : Alternating torsional stress acting on the shaft. In cases where alternating bending stress acts on the shaft simultaneously, τ_{De} given by the following formula is to be used: (N/mm^2)

$$\tau_{De} = \sqrt{\tau_D^2 + \frac{1}{3}\left(\frac{\beta_b}{\beta_t}\sigma_D\right)^2}$$

τ_{De} : Equivalent alternating torsional stress (N/mm^2)

σ_D : Alternating bending stress (N/mm^2)

β_b : Notch factor for bending stress

β_t : Notch factor for torsional stress

τ_y : Torsional yielding stress of shaft material (N/mm^2)

S_y : Safety factor for yielding

τ_f' : Torsional fatigue strength of the shaft material acting under the mean stress τ_m (or τ_{me}) (N/mm^2)

S_f : Safety factor for fatigue

2 Fatigue strength and yielding stress of the shaft material in **-1** above is to be determined on a case by case basis by the Society, with consideration being given to materials, heat treatments, surface treatments, etc. on the basis of information and data submitted by the applicant. In addition, safety factors for fatigue and yielding are to be determined on a case by case basis by the Society with consideration being given to the purpose of the shaft, working conditions, etc.

3 The diameters of propeller shafts of vessels whose main propulsion machinery falls under one of classes of high speed engines which are reciprocating internal combustion engines may be in accordance with **(1)** to **(3)** below. However, in special cases, such as the vessels are intended to navigate in rough sea frequently, special considerations for the features affecting to the strength are to be paid.

(1) The definition of "high speed engine"

The term "high speed engine" in this sub-paragraph is defined as those engines simultaneously complying with the following conditions:

$$\frac{Sn^2}{1.8\times10^6}\geq 90$$

$$\frac{\pi d_j n}{6.0\times10^4}\geq 6$$

S : Length of stroke (mm)

n : Number of revolutions at maximum continuous output of engine (rpm)

d_j : Diameter of crank journal (mm)

(2) Required diameter of propeller shafts

The diameter of propeller shafts is to be not less than the value determined by the following formula:

$$d_s = 100k\cdot\sqrt[3]{\frac{H}{N_0}}$$

d_s : Required diameter of propeller shaft (mm)

H : Maximum continuous output of main propulsion machinery (kW)

N_0 : Number of revolutions of shaft at maximum continuous output (rpm)

k : Factor given in **Table D6.2.6-1**. For those propeller shafts Kind 1 or stern tube shafts Kind 1 which are made of carbon steel or low alloy steel with tensile strength exceeding 400 N/mm^2, the factor k may be multiplied by the following factor K_{m1}.

$$K_{m1} = \sqrt[3]{\frac{560}{T_s+160}}$$

T_s : Specified tensile strength (N/mm^2)

(3) Torsional vibration

The torsional vibration for those shafting systems, to which these requirements applied, is to comply with the requirement **D8.2.6-2**.

Table D6.2.6-1 Values of k

Carbon steel or low alloy steel		$KSUSF316$ $KSUS316\text{-}SU$	$KSUSF316L$ $KSUS316L\text{-}SU$	Precipitation hardened martensite stainless steel
Kind 1	Kind 2			
1.00	1.05	1.03	1.08	0.85

D6.2.7 Corrosion Protection of Propeller Shafts and Stern Tube Shafts

1 Shafts effectively protected against corrosion caused by water using a means approved by the Society in **6.2.7-1(1), Part D of the Rules** are to be either of the following **(1)** to **(4)**:

(1) Shafts effectively protected from any contact with water in ships having oil lubricated stern tube bearings (including shaft bracket bearings when used) equipped with approved sealing devices.

(2) Shafts effectively protected from any contact with water by continuous copper alloy sleeves fitted onto the shafts by shrinkage fit in ships having water lubricated stern tube bearings (including shaft bracket bearings when used).

(3) Shafts fitted with shrunk-on copper alloy sleeves in cases where they are supported by stern tube bearings (including shaft bracket bearings when used) and covered with rubber or other synthetic resin materials so that they may be effectively protected from any contact with water in ships having water lubricated stern tube bearings.

(4) Shafts of other designs specially approved by the Society.

2 The wording "corrosion resistant materials approved by the Society" in **6.2.7-1(3), Part D of the Rules** means those materials which have been subjected to approval tests specified in **2.4.2-5, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use** and then which obtain type approval of use of machinery and equipment as a corrosion resistant material for propeller shafts or stern tube shafts. In addition, *KSUSF*316, *KSUSF*316*L*, *KSUS*316-*SU* or *KSUS*316*L*-*SU* used for the propeller shafts exceeding 200 *mm* in diameter are also to be in accordance with this requirement to obtain type approval of use of machinery and equipment as a corrosion resistant material for propeller shafts or stern tube shafts.

D6.2.10 Stern Tube Bearings and Shaft Bracket Bearings

1 The wording "provisions specified elsewhere" in **6.2.10-1(1)(a)i), Part D of the Rules** means the following **(1)** and **(2)** in principle:

(1) Shaft alignment calculations are to be carried out in accordance with the requirements in **Annex 6.2.13, Part D of the Rules** "CALCULATION OF SHAFT ALIGNMENT".

(2) For improving the lubricating condition of the bearing, the following measures are to be taken:

(a) A lubricating oil inlet is to be provided at the aft end of the bearing to ensure the forced circulation of the lubricating oil.

(b) Either of the following devices to measure stern tube bearing metal temperature at the aft end bottom along with high temperature alarms (with a preset value of 60 °C or below) is to be provided:

i) Two or more temperature sensors embedded in the metal; or

ii) An embedded temperature sensor, replaceable from inboard the ship, and a spare temperature sensor. In this case, the replacement of such sensors according to procedures submitted beforehand is to be demonstrated.

(c) Low level alarms are to be provided for lubricating oil sump tanks.

2 The wording "provisions specified elsewhere" in **6.2.10-1(1)(b)ii), Part D of the Rules** means the following **(1)** and **(2)** in principle:

(1) Nominal bearing pressure, etc. calculated in accordance with **Annex 6.2.13, Part D of the Rules** "CALCULATION OF SHAFT ALIGNMENT" are to be within the allowable limits specified in the Type Approval Certificate.

(2) The measures for lubricating condition specified in **-1(2)** are to be taken.

3 The wording "provisions specified elsewhere" in **6.2.10-1(2)(b), Part D of the Rules** means the following **(1)** and **(2)** in principle:

(1) Nominal bearing pressure is to be within the allowable limit specified in the Type Approval Certificate.

(2) Forced lubrication using water pumps is to be adopted and a low flow alarm is to be provided at the lubricating water inlet.

D6.3 Tests

D6.3.2 Tests after Installation On Board

The wording "the requirements specified otherwise by the Society" means the following:

(1) Confirmation tests for the optical or laser sighting of the shaft center (only in cases where a shipyard has not had a sufficient record of experience in new ship building Society classified, or in cases where the very first ship of its type, shape or size is being built at the shipyard)

(2) Confirmation tests for measuring sag and the gaps between shaft coupling flanges (an alternative test may be acceptable for those couplings in which it is difficult to carry out this measurement.)

(3) Confirmation tests for jacking up the shaft near bearings (only in cases where the shafts are coupled before launching)

(4) Confirmation tests at sea trials to check that crankshaft hot deflection values are within engine manufacturer recommended ranges (confirmation of the measuring record is acceptable instead.)

D7 PROPELLERS

D7.2 Construction and Strength

D7.2.1 Thickness of Blade

1 With respect to the wording "as deemed appropriate by the Society" specified in Note (1) for **Table D7.2 of the Rules**, the value is to be 0.6 in cases where the materials used for propellers are grey cast iron, and the values are to be determined on a case by case basis by the Society in cases where the material is some other specific material.

2 In accordance of **7.2.1-1 of the Rules**, the following **(1)** to **(3)** are to be applied to Δw and w when using experimental data taken from model ships which are 6 *m* or more in length.

(1) Δw and w are to be calculated using a wake distribution for those model ship or be calculated using a method deemed equivalent thereto.

(2) The wake distribution is to be converted into the full scale value, using a suitable method.

(3) w is to be calculated by averaging the square measure of the wake distribution inside the propeller's circumference.

3 When applying **7.2.1-4, Part D of the Rules**, the standard method of detailed calculation of a propeller blade thickness is shown as follows:

(1) The hydraulic forces on a propeller blade during a propeller rotation are calculated by the lifting-surface theory, and the stresses on the propeller blade are calculated by structural analysis using the hydraulic forces. The wake distribution used for the calculation of the hydraulic forces is to be experimental data taken from a sister vessel or a model ship (data is to be corrected appropriately to the actual ship's scale). In cases where such data is not known, the data shown in **Fig. D7.2.1-2** or **Table D7.2.1-3** may be used for high speed craft ($C_b \leq 0.6$), excluding those with unconventional stern constructions (such as multi-shafting arrangements), instead.

(2) The stress ratio σ_a/σ_m is calculated using the maximum value of stress amplitude σ_a and mean stress σ_m (both of which are calculated by the method given in **(1)**) on each radius location of the propeller blade during propeller rotation.

(3) Mean stress at maximum continuous revolutions $(\sigma_m)_{MCR}$ is calculated by the method given in **(1)**. Absorbed output of propellers, or the torque factor K_Q (calculated from the hydraulic force calculation), is to be equal to the output of main propulsion machinery at the maximum continuous revolutions, or the torque factor K_Q ($=H/30\,N_0^3D^5$) at maximum continuous revolutions.

In place of that calculation, $(\sigma_m)_{MCR}$ may also be calculated by the following formula:

$$(\sigma_m)_{MCR} = 100\frac{K_1HK}{ZN_0lt_a^2K_2}$$

t_a : Actual thickness of propeller blade (*cm*)

Other parameters : As specified in **7.2.1, Part D of the Rules**.

(4) $(\sigma_m)_{MCR}$ is calculated by following formula using the value of σ_a/σ_m determined by **(2)**.

$$(\sigma_a)_{MCR} = (\sigma_m)_{MCR} \times (\sigma_a/\sigma_m)$$

(5) In consideration of the effects of heavy weather, the stresses on the propeller blade are calculated by the following formulae:

$$\sigma_m = (\sigma_m)_{MCR} \times S$$
$$\sigma_a = (\sigma_a)_{MCR} \times S$$

S : As specified in **7.2.1, Part D of the Rules**.

(6) In cases where the stress amplitude σ_a and the mean stress σ_m calculated in **(5)** satisfy the following allowable stresses of propeller materials, these stresses are to be considered compliant with **7.2.1 of the Rules**:

$\sigma_a \leq k_1 - k_2\sigma_m$ (*N/mm*^{2})

$\sigma_m \leq k_3$ (*N/mm*^{2})

k_1, k_2 and k_3: As specified in **Table D7.2.1-2**.

Fig. D7.2.1-1 Standard Wake Distribution for High Speed Craft

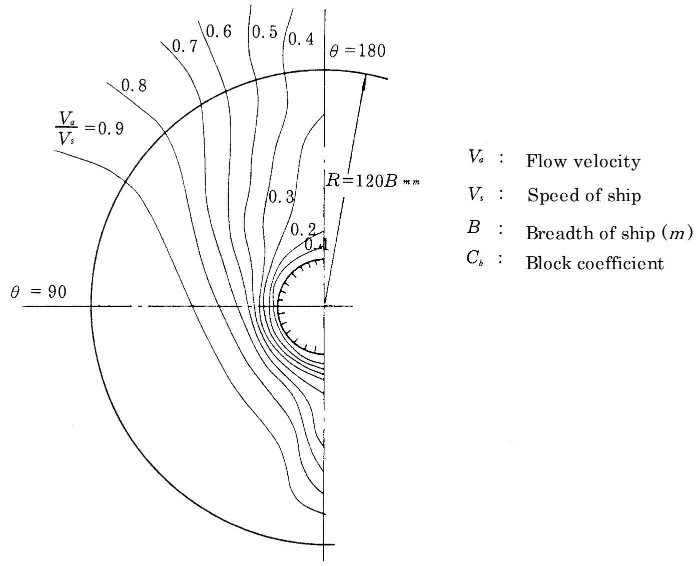


Table D7.2.1-1 Standard Wake Distribution for High Speed Craft (V_a/V_s)

Rotation angle of propeller (θ)	Radius of propeller (mm)					
	36B	48B	60B	78B	96B	114B
0	0.390	0.520	0.560	0.660	0.820	0.930
10	0.406	0.610	0.706	0.841	0.924	0.984
20	0.444	0.688	0.799	0.929	0.981	1.000
30	0.490	0.750	0.850	0.950	1.000	1.000
40	0.509	0.788	0.874	0.959	1.000	1.000
50	0.520	0.807	0.889	0.964	1.000	1.000
60	0.526	0.810	0.900	0.970	1.000	1.000
70	0.521	0.807	0.889	0.966	1.000	1.000
80	0.505	0.784	0.873	0.952	0.996	1.000
90	0.490	0.740	0.850	0.930	0.980	1.000
100	0.439	0.682	0.808	0.907	0.949	0.976
110	0.389	0.618	0.748	0.866	0.914	0.947
120	0.350	0.550	0.670	0.800	0.870	0.910
130	0.303	0.469	0.571	0.703	0.796	0.849
140	0.262	0.398	0.473	0.599	0.696	0.765
150	0.230	0.340	0.390	0.500	0.580	0.660
160	0.201	0.284	0.333	0.403	0.417	0.533
170	0.184	0.240	0.278	0.317	0.361	0.401
180	0.180	0.220	0.240	0.270	0.290	0.320

Note:

B : Breadth of ship (m)

Table D7.2.1-2 Values of k_1,k_2 and k_3

Materials	k_1	k_2	k_3
*KHBsC*1	32.4	0.22	61.6
*KHBsC*2			
*KAlBC*3	37.3	0.22	70.8
*KAlBC*4	32.4	0.22	61.6

D7.2.2 Controllable Pitch Propellers

1 The wording "documents deemed appropriate by the Society" specified in **7.2.2-2, Part D of the Rules** means the following documents (**(1)** to **(5)**).

(1) Calculation sheet for the load sharing factor k of the bolt

where

$$k = \frac{k_b}{k_b + k_f}$$

k_b : rigidity of bolt tension

k_f : rigidity of flange compression

(2) Static stress and dynamic stress acting on the bolt

(3) Specifications of bolt material (including the manufacturing process)

(4) Endurance limit curve of the bolt (both in air and in sea water)

(5) Securing method of the bolt

2 When fitting blade fixing bolts according to the requirements of **7.2.2-5, Part D of the Rules**, blades are to be fitted securely to pitch control gears by giving all of the fixing bolts an adequate initial fitting force. It is to be regarded as standard practice that the initial fitting force complies with the following condition;

$$\frac{1.3}{n}\left(\frac{AK_3}{L} + F_c\right) < T_0 < 0.55\sigma_0 d^2$$

where

T_0 : initial fitting force (N)

σ_0 : Yield strength or 0.2 % proof strength of bolt material (N/mm^2)

Other symbols are the same as in the formula shown in **7.2.2-2, Part D of the Rules**.

3 Installation of back-up hydraulic oil pumps specified in **7.2.2-8 of the Rules** may be omitted for ships with propeller pitch fixing devices that can work in normal service conditions and that can be easily changed from ahead to astern while using these devices.

D7.3 Force Fitting of Propellers

D7.3.1 Pull-up Length

In the provision of coefficient "c" used in the calculation of tangential force F_v, specified in **7.3.1-1, Part D of the Rules**, the wording "the satisfaction of the Society", means determining "c" in accordance with **(2)** below using maximum torque Q_{max} as derived from **(1)** below:

(1) Q_{max}, which is the value of the maximum torque acting on the propeller's fitted portion, is to be determined by measurements or precise estimation complying with the following **(a)** or **(b)**, and approved by the Society:

(a) In cases where Q_{max} is determined by measuring, the measurements are to be carried out on a sister ship (complete same design including the main engine, shafting system and so on) under a fully loaded condition at the time of the astern tests required under the provision of item 2, **Table B2.11, Part B of the Rules**.

(b) In cases where Q_{max} is determined by estimation, the estimation method is to be verified with an estimation error not exceeding 10 % when compared with the results of actual measurements taken at the time of the astern tests.

(2) When using a value for Q_{max} determined above, the coefficient "c" in **7.3.1-1, Part D of the Rules** is to be 1.2 or the value given by following formula, whichever is greater:

$$c = 5.08 \times \frac{N_0 Q_{max}}{H} \times 10^{-3}$$

N_0, H : Same as those specified in 7.2.1-1, Part D of the Rules

Q_{max} : Maximum torque acting on the propeller's fitted portion under every operation conditions including transient conditions, such as crash astern, derived from **(1)** above (N-m)

D7.3.2 Propeller Boss

Heating temperature is not to exceed 100℃ in cases where the propeller boss is drawn out of the propeller shaft.

D7.4 Tests

D7.4.1 Shop Tests

The value of the unbalanced mass, measured during static balancing testing of propellers at the time of manufacture, is not to exceed the value determined by the following formula:

$$m = \frac{3.6M \times 10^{-3}}{D\left(\frac{N_0}{100}\right)^2}$$

where

m : unbalanced mass converted from a value given on the outer circumference of the propeller (kg)

M : mass of the propeller (kg)

D : diameter of the propeller (m)

N_0 : number of revolutions of the propeller shaft at the maximum continuous output (rpm)

D7.4.2 Tests after Installation On Board

1 In cases where propellers are force fitted onto propeller shafts by hydraulic force, the confirmation of the pull-up length specified in 7.4.2, Part D of the Rules is to be made assuming that the true relative starting point is the point where the pull-up load equals zero on an approximate line drawn through the measured points plotted on a chart of the relationship between pull-up length and load.

2 In the force fitting test for keyless propellers, it is to be confirmed that the pull-up length measured according to -1. above is between the upper and lower limits specified in 7.3.1-1, Part D of the Rules, and that the apparent coefficient of friction derived from following formula is not less than 0.1 and below 0.2.

$$\mu_r = \frac{K\frac{K_E}{S} - tan\alpha}{1 + K\frac{K_E}{S}tan\alpha}$$

μ_r : Apparent coefficient of friction derived from the results of force fitting tests

K : Rate of fitting force to pull-up length derived from the results of force fitting tests used for dry-fitting methods (N/mm)

K_E, S, α : Same as those specified in 7.3.1-1, Part D of the Rules

3 In cases where propellers are force fitted onto propeller shafts with the use of key, the standard pull-up length is generally as follows;

$$L_4 = \frac{2d_p}{tan\,\alpha} \times 10^{-4}$$

L_4 : Standard pull-up length (mm)

d_p : Diameter of propeller shaft (cone part large end) (mm)

α : Half-angle of the taper at the propeller shaft cone part (deg)

D8 TORSIONAL VIBRATION OF SHAFTINGS

D8.2 Allowable Limit

D8.2.2 Intermediate Shafts, Thrust Shafts, Propeller Shafts and Stern Tube Shafts

1 The allowable limit of the torsional vibration stress for propeller shafts Kind 1 made of approved corrosion-resistant materials or propeller shafts Kind 2 is to be calculated by the following formulae in place of the formulae for τ_1 shown in **8.2.2-1(1), Part D of the Rules**.

$$\tau_1 = A - B\lambda^2 \quad (0 \leq \lambda \leq 0.9)$$

$$\tau_1 = C \quad (0.9 < \lambda)$$

where

λ : ratio of the number of revolutions to the number of maximum continuous revolution

A, B, C : constants dependent on shaft materials, given in **Table D8.2.2-1**.

2 For ships that fall under any of the following, the allowable limits of torsional vibration stress on the intermediate shafts, thrust shafts, propeller shafts and stern tube shafts are to be calculated by applying the values of C_K given in the following **Table D8.2.2-2** to the formula specified in **8.2.2-1(1), Part D of the Rules**.

(1) Ships in which steam turbines, or gas turbines are used as main propulsion machinery (excluding electric propulsion ships)

(2) Ships in which reciprocating internal combustion engines are used as main propulsion machinery (excluding electric propulsion ships), having slip couplings such as electro-magnetic couplings or fluid couplings between engine and propulsion shafting; or

(3) Electric propulsion ships

Table D8.2.2-1 Values of A, B and C

	Carbon steel or low alloy steel without effective protection against seawater corrosion	Austentic stainless steel with 0.2% proof stress not less than 205 N/mm^2	Precipitation hardened martensite stainless steel with 0.2% proof stress not less than 400 N/mm^2
A	32.4	40.7	61.1
B	24.6	30.5	47.3
C	12.5	16.0	22.8

Note:

Values for materials other than those given above are to be determined by the Society on a case by case basis.

Table D8.2.2-2 Values of C_K

Intermediate shafts			Thrust shafts		Propeller shafts and stern tube shafts
Integral flange couplings	Shrinkfit couplings	Keyways	On both sides of the thrust collar	In way of axial bearings where a roller bearings is used as a thrust bearing	—
0.75	0.75	0.45	0.65	0.65	0.35

Note:

Values of C_K other than those given above are to be determined by the Society on a case by case basis.

D8.2.4 Power Transmission Systems

The wording “the provisions specified elsewhere” in **8.2.4-3, Part D of the Rules** means the following with respect to rubber couplings.

(1) When the number of revolutions is within the range of 80 % to 105 % of the number of maximum continuous revolutions, the torsional vibration torque amplitude is not to exceed the T_1 value shown below:

$$T_1 : 2.5 \times 10^3 \times \frac{H}{N_0}$$

where

T_1 : Allowable limit of torsional vibration torque amplitude within the speed range of 80 % to 105 % of the number of maximum continuous revolutions ($N \cdot m$)

H : Maximum continuous output of engine (kW)

N_0 : Number of revolutions of rubber couplings at the maximum continuous output of engine (rpm)

However, in cases where some margin exists between the allowable mean torque for a rubber coupling and its actual mean torque, T_1 may be multiplied by the coefficient F calculated from the following formula:

$$F = T_n / T_m$$

where

T_n : Allowable mean torque approved for the rubber coupling concerned ($N \cdot m$)

T_m : Mean torque obtained by the maximum continuous output of an engine and rotational speed of the coupling corresponding to the number of maximum continuous revolutions of an engine ($N \cdot m$)

(2) When the number of revolutions is within the range of 80 % and below the number of maximum continuous revolutions, the torsional vibration torque amplitude is not to exceed the value for T_2 shown below. In cases where the torsional vibration torque amplitude exceeds the value of T_1 specified in **(1)** above, a barred speed range for avoiding continuous operation as specified in **D8.3.1** is to be provided.

$$T_2 = 8 \times T_1$$

where

T_2 : Allowable limit of torsional vibration torque amplitude at an engine speed within the range of 80 % and below the number of maximum continuous revolutions ($N \cdot m$)

(3) In cases where rubber couplings, whose torque is transmitted in the direction of shearing rubber elements, are used in main propulsion shafting driven by reciprocating internal combustion engines having outputs of 3,500 kW or more, the main propulsion shafting is to comply with the following **(a)** and **(b)**:

(a) Audible and visual alarms, which come into action when the vibratory torque exceeds the allowable limit T_1 of torsional vibration torque specified in **(1)** above, are to be provided at main control stations. However, in cases where no vibratory torque is likely to exceed the allowable limit T_1, within the operational speed range, as the result of the torsional vibration calculation specified in **(b)**, the requirements may be dispensed with.

(b) Means are to be provided to show the safe operation range by torsional vibration calculations under a one cylinder cut condition.

D8.2.6 Detailed Evaluation for Strength

1 In cases where the torsional stresses acting on the shafts satisfy the conditions specified in **D6.2.6-1** and **-2**, alternating torsional stress τ_D in the said conditions may be used for determining the allowable limit of torsional vibration stress in lieu of τ_1 specified in **Chapter 8, Part D of the Rules**.

2 In cases where the diameter of shafts are determined in accordance with **D6.2.6-3**, allowable limit of torsional vibration stress τ_1 and τ_2 are to be calculated in accordance with the following:

(1) When the number of revolutions is within the range of 80% to 105% of the number of maximum continuous revolutions, the torsional vibration allowable limit τ_1 is to be calculated by the following formulae:

$$\tau_1 = A - B\lambda^2 \quad (\lambda \le 0.9)$$

$$\tau_1 = C \quad (0.9 < \lambda)$$

τ_1 : Allowable limit of torsional vibration stresses for the range of $0.8 < \lambda \le 1.05$ of the number of maximum continuous revolutions (N/mm^2)

λ : Ratio of the number of revolutions to the number of maximum continuous revolutions

A, B, C : Constants dependent on shaft materials, given in **Table D8.2.6-1.**

For a propeller shaft Kind 1 made of carbon steel or low alloy steel with a specified tensile strength exceeding 400 N/mm^2, the values determined by above formulae may be multiplied by the coefficient K_{m2} given below:

$$K_{m2} = \frac{T_s + 160}{560}$$

T_s : Specified tensile strength of the shaft material (N/mm^2)

(2) When the number of revolutions is within the range of 80% and below, the allowable limit of torsional vibration stresses τ_2 is to be calculated by the following formula. In cases where torsional vibration stress exceeds τ_1, the barred speed range specified in **8.3, Part D of the Rules** is to be imposed.

$$\tau_2 = 2.3\tau_1$$

τ_2 : Allowable limit of torsional vibration stresses for the range of $\lambda \leq 0.8$ of the number of maximum continuous revolutions (N/mm^2)

τ_1 : Value calculated by the formula for $\lambda \leq 0.9$ in **(1)** above (N/mm^2)

λ: Ratio of the number of revolutions to the number of maximum continuous revolutions

Table D8.2.6-1 Values of A, B and C

	Carbon steel or low alloy steel		Austentic stainless steel		Precipitation hardened martensite stainless steel
	Shaft Kind 1	Shaft Kind 2	*KSUSF*316 *KSUSF*316-*SU*	*KSUSF*316L *KSUSF*316*L*-*SU*	
A	24.5	21.0	26.4	24.4	39.6
B	24.3	20.0	27.1	25.3	39.0
C	4.8	4.8	4.5	3.9	8.1

Note:

Values for materials other than those given above are to be determined by the Society on a case by case basis.

D8.3 Barred Speed Range

D8.3.1 Barred Speed Range for Avoiding Continuous Operation

In cases where torsional vibration torque amplitude exceeds the allowable limit T_1 specified in **8.2.4(1), Part D of the Rules** above, the barred speed range for avoiding continuous operation specified in **8.3.1, Part D of the Rules** is to be calculated by replacing τ_1 with T_1.

D9 BOILERS, ETC. AND INCINERATORS

D9.1 General

D9.1.3 Drawings and Data to be Submitted

The operating Instructions specified in **9.1.3(2)(c) of the Rules** are to include the following information:

(1) Feed water treatment and sampling arrangements

(2) Operating temperatures (exhaust gas and feed water temperatures)

(3) Operating pressures

(4) Inspection and cleaning procedures

(5) Records of maintenance and inspections

(6) The need to maintain adequate water flows through economizers under all conditions

(7) Periodical operational checks of safety devices to be carried out by operating personnel and to be documented accordingly

(8) Procedures for using exhaust gas economizers in dry conditions

(9) Procedures for the maintenance and overhaul of relief valves

D9.2 Materials and Welding

D9.2.1 Materials

1 The pressure parts of boilers in **9.2.1-1, Part D of the Rules** which are required to use materials complying with the requirements given in **Part K of the Rules,** include all those shown in **Fig. D9.2.1-1** such as: nozzles welded to the boiler drum, manhole rings, stiffeners (except for those used for screwing fittings), flanges attached to nozzles (except for those used for connecting piping), manhole covers, cleaning hole covers, inspection hole covers, etc. .

2 The boiler fittings "whose dimensions and conditions of service have been approved by the Society" referred to in **9.2.1-2, Part D of the Rules** means the fittings specified in **D1.1.4**. In addition, the wording "standards recognized by the Society" referred to in **9.2.1-2, Part D of the Rules** means national or international standards such as *JIS*.

D9.2.4 Non-destructive Tests for Cast Steels

The criteria for non-destructive tests of cast steels used for boiler drums exposed to internal pressure are to be in accordance with the following:

(1) The testing method of radiographic tests is to be in accordance with *JIS G* 0581. Any cracks or insufficient fusion are to be judged unacceptable. Gas and blowholes, sand spots, inclusions, and internal shrinkage are judged acceptable in cases where the defects are classified as Grade 1 in accordance with the above standard.

(2) The testing method of magnetic particle tests is to be in accordance with *JIS Z* 2320-1 to -3. The result of a magnetic particle test is acceptable, provided the following **(a)** to **(d)** are complied with:

  (a) There are no magnetic particle indications due to surface cracks.

  (b) The maximum length of linear magnetic particle indications is 4 *mm* or less.

  (c) The major axis length of circular magnetic particle indications is 4 *mm* or less.

  (d) The point total specified in **Table D9.4.2** with respect to the type of magnetic particle indications is 12 or less within an area of 2500 mm^2 for scattering magnetic particles.

(3) Defects judged unacceptable by **(1)** or **(2)** may be repaired. Welding for such repairs is to be in accordance with the requirements specified in **5.1.11, Part K of the Rules**.

Table D9.4.2 Points for Scattering Magnetic Particle Indication

Magnetic particle pattern of defect	Magnetic particle indications of 2 *mm* or less	Magnetic particle indications of more than 2 *mm*, 4 *mm* or less
Linear	3	6
Circular	1	2

D9.3 Design Requirements

D9.3.4 Boilers of Unusual Shape

The "analysis results as deemed appropriate by the Society" specified in **9.3.4, Part D of the Rules** means structural strength analysis by a strength assessment such as FEM.

D9.3.7 Consideration for Soot Fire

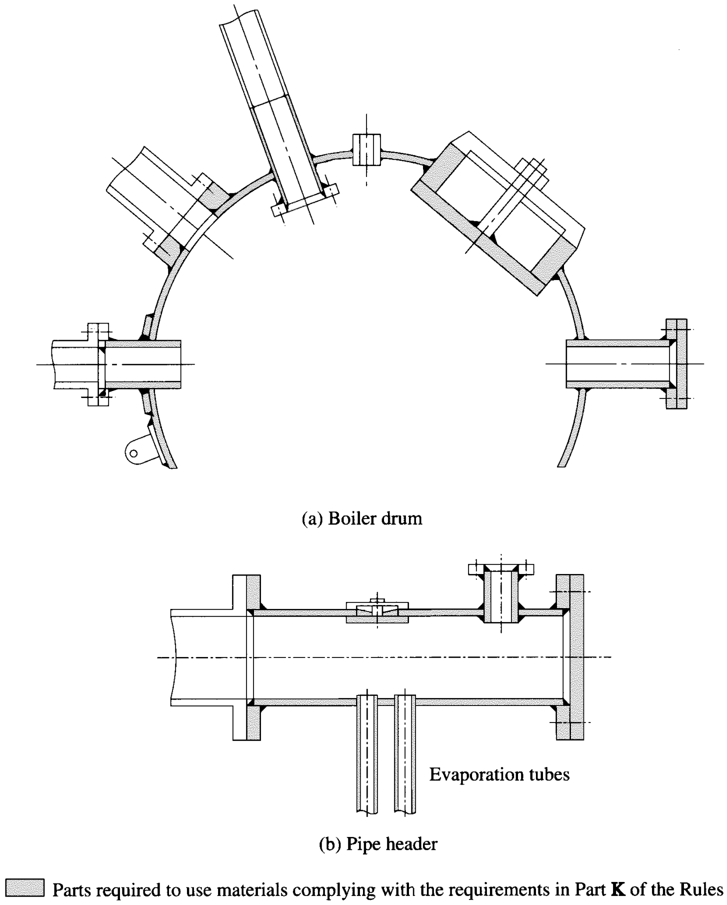
The "consideration" specified in **9.3.4, Part D of the Rules** means (but is not limited to) arrangements for soot cleaning such as the soot blowers with cleaning holes.

D9.4 Allowable Stress and Efficiency

D9.4.1 Allowable Stress

In cases where materials other than those specified in **Table D9.2 of the Rules** are used, the values for allowable stress may be in accordance with any national standard or **Table D12.3(1)** and **(2) of the Rules**, if applicable.

Fig. D9.2.1-1 Materials Used for the Pressure Parts of Boilers



D9.5 Calculations of Required Dimensions of Each Member

D9.5.6 Required Thickness of Flat Plates with Stays or Other Supports

1 In cases where the required thickness at portions including water tube holes of the tube plates of dry combustion cylindrical boilers are calculated by the formula specified in **9.5.6-1, Part D of the Rules**, the value of 'C_5' in the formula for those supports adjacent to water tube holes is to be divided by the square of the rate of strength reduction obtained by the following formula:

$$\eta = \frac{p - 0.5d}{p}$$

where

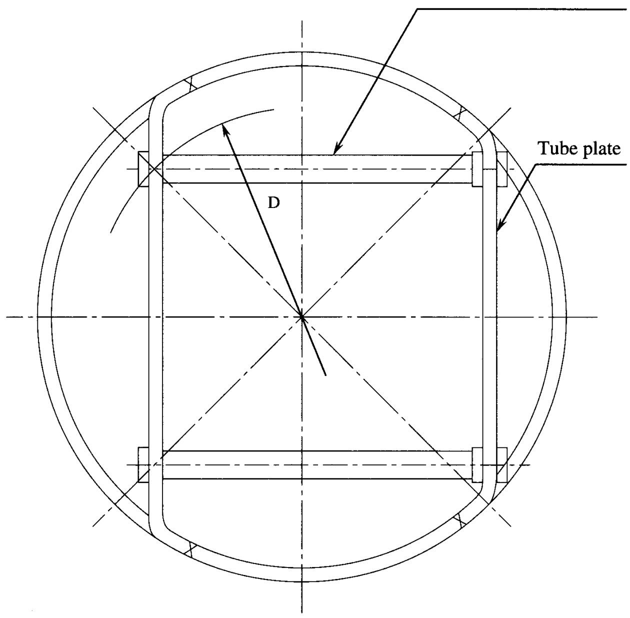
η : rate of strength reduction

p : pitch of water tube holes (mm)

d : diameter of water tube (mm)

2 The standards "deemed appropriate the Society" specified in **9.5.6-1, Part D of the Rules** means national or international standards such as *JIS*.

Fig. D9.5.6-1 Definition of "*D*" for Vertical Boilers with Horizontal Smoke Tubes



3 In **9.5.6-4, Part D of the Rules**, '*D*' in the case of vertical boilers with horizontal smoke tubes is to be taken as specified in **Fig. D9.5.6-1**. Also, 'd_s' in the formula is to be the mean of the hole diameters of contiguous tube holes if such tube hole diameters are different as in those cases where stay tubes and smoke tubes are alternately arranged.

D9.5.8 Required Thickness of Plain Cylindrical Furnaces

The required thickness of plain cylindrical furnaces supported with stays or other members is to be calculated by the formula specified in **9.5.8, Part D of the Rules** by regarding the effective length between the stays as '*L*' in that formula.

D9.5.9 Required Thickness of Hemispherical Furnaces without Stays or Other Supports

Among those cases where "because the part is of an unusual shape" referred to in **9.3.4-1, Part D of the Rules**, the required thickness of dished furnace plates fitted with flue tubes is to be determined by the formula below:

$$T_r = \frac{PR_f}{f} + 1$$

where

T_r : required thickness of the dished furnace plate fitted with flue tube (mm)

P : design pressure (MPa)

R_f : outer radius at the centre of furnace plate (mm)

f : allowable stress specified in **9.3.1, Part D of the Rules** (N/mm^2)

D9.5.12 Required Diameter of Stays

1 The boundaries for net areas supported by one stay or stay tube are, according to the adjacent support conditions, to be determined as follows:

(1) In cases where such adjacent supports are stays or stay tubes, boundaries are considered to be perpendicular bisectors of the lines connecting both support points.

(2) In cases where such adjacent supports are curved flanges or welded joints, boundaries are considered to be the locus of the centres of the inscribed circles to those support points in question and the commencement of curvatures or the inside of those welding parts specified in **9.5.6-2, Part D of the Rules**.

2 At the corner parts of smoke tube nests, calculations may be carried out by regarding net areas as the half of the sum of two areas supported by stays or stay tubes adjacent each other.

D9.6 Manholes, Other Openings for Nozzles, etc. and their Reinforcements

D9.6.1 Manholes, Cleaning Holes and Inspection Holes

The required thickness of manhole covers is to be determined by the formula below. However, note that the thickness at the centre is not to be made 14 *mm* or less. In cases where grooves are provided at the periphery of manholes, the thickness of such parts may be reduced to 2/3 of that of the central area.

$$t = \frac{b}{2c}\sqrt{\frac{100P}{f}}$$

where

t : required thickness of manhole cover (mm)

P : design pressure (MPa)

f : allowable stress specified in **9.3.1, Part D of the Rules** (N/mm^2)

b : length of minor axis of manhole (mm)

c : value given in **Fig. D9.6.1-1**

In **Fig. D9.6.1-1**, 'a' stands for the length of the major axis of manholes (mm), and in cases where b/a is 1, 'c' is to be 9.

In the case of corrugated manhole covers, 'a' and 'b' are to be taken as shown in **Fig. D9.6.1-2**.

Fig. D9.6.1-1 Values of *c*

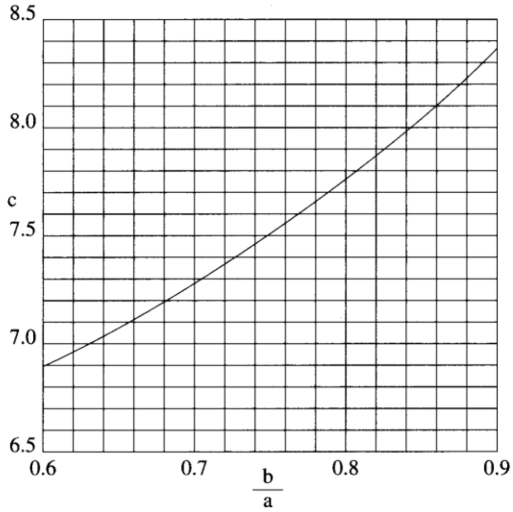
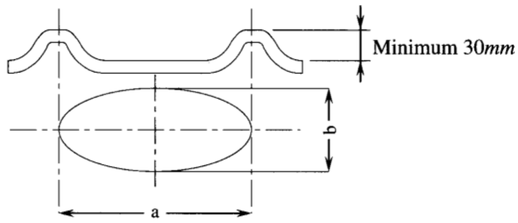


Fig. D9.6.1-2 '*a*' and '*b*' of Corrugated Manhole Covers



D9.9 Fittings, etc.

D9.9.1 Materials of Fittings

The "special cast iron" in **9.9.1-2(3), Part D of the Rules** means cast iron having those mechanical properties specified in *JIS G* 5705 (Malleable Iron Castings, *FCBM*32, *FCBM*34 and *FCBM*35), *JIS G* 5502 (Spherical Graphite Iron Castings, *FCD*400) or the equivalent thereto.

D9.9.2 Construction of the Fittings

The wording "recognized standards" in **9.9.2-1, Part D of the Rules** means any national standards. In cases where standards are not specified for marine boilers, those for boilers for land use such as *ASME* or *ANSI* may be referred to.

D9.9.3 Safety Valves and Relief Valves

1 In the calculation of **9.9.3-5(2), Part D of the Rules**, the correction coefficient '*C*' given in **Table D9.9.3-1** may be used in the formula in place of $\sqrt{V_H/V_S}$.

2 In **9.9.3-12, Part D of the Rules**, the springs of safety valves or relief valves are to be installed capable of being compressed to at least 1/10 of the diameters of valve seats. Furthermore, permanent sets of such springs under solid conditions at room temperatures for a period of 10 *minutes* are not to exceed 1% of their free height.

3 The wording "an accumulation test deemed appropriate by the Society" specified in **9.9.3-15, Part D of the Rules**, means tests such that in cases where safety valves blow under the maximum firing conditions of boilers with the stop valves closed except for the valves for steam supplies to machinery necessary to the operation of such boilers.

D9.9.7 Burning Systems

In cases where one complete spare unit which can be replaced in a short period of time is provided for those draught fans specified in **9.9.7-2, Part D of the Rules**, the provision of alternative means specified therein may be omitted.

D9.10 Tests

D9.10.1 Shop Tests

1 In hydrostatic tests for boilers, the test pressure for the hydrostatic tests of boiler tubes and connecting pipes after completion of welding and assembly may be modified to 1.25 times design pressure, in cases where parts or members such as drums, headers, and others are individually subjected to hydrostatic tests at test pressures 1.5 times design pressures.

2 The hydrostatic tests of desuperheaters installed within the water drums or steam drums of boilers are to be carried out at those test pressures specified below. In either case, test pressures are not to be less than 2.0 *MPa*.

(1) 1.5 times or more of the design pressures of boilers in cases where steam stop valves are provided at the inlets of desuperheaters.

(2) 1.5 times or more of the assumed pressure differences in cases where stop valves are provided only at the outlets of desuperheaters.

D9.11 Construction etc. of Small Size Boilers

D9.11.2 Materials, Construction, Strength and Accessories of Small Boilers

The wording "recognized standard" referred to in **9.11.2-1, Part D of the Rules** means the requirements of national or international standards such as *JIS*.

D9.12 Construction of Thermal Oil Heaters

D9.12.3 Safety Devices, etc. for Thermal Oil Heaters Directly Heated by the Exhaust Gas of Engines

The wording "Fixed fire extinguishing and cooling systems as deemed appropriate by the Society" in **9.12.3-7, Part D of the Rules** means combinations of fixed gas fire-extinguishing systems and systems for cooling heating coils, headers, casings, etc., and heater themselves such as water-spray. Fixed fire extinguishing cooling systems can be water-drenching systems able to discharge copious amounts of water. In such cases, the suitable means for collection and drainage, to prevent any water from flowing into reciprocating internal combustion engines, are to be provided on exhaust ducting below heaters, and such drainage is to be led to suitable places.

D9.13 Incinerators

D9.13.3 Construction and Fittings

The wording "to be subject to the recognition of the Society" in **9.13.3(3)(c), Part D of the Rules** signifies that means are provided to prevent the passage of any exhaust gases into other boilers, thermal oil heaters and incinerators, and furthermore, means are also provided to enable operators to verify their operating conditions.

Table D9.9.3-1(a) Coefficient of Correction C for Steam Temperature and Pressure (1)

Absolute pressure	Temperature (℃)									
(*MPa*)	200	220	240	260	280	300	320	340	360	380
0.5	0.996	0.972	0.951	0.931	0.913	0.896	0.879	0.864	0.849	0.835
1	0.981	0.983	0.960	0.938	0.919	0.901	0.884	0.868	0.853	0.838
1.5	0.976	0.970	0.972	0.947	0.925	0.906	0.888	0.872	0.856	0.841
2		0.967	0.964	0.955	0.932	0.912	0.893	0.876	0.860	0.845
2.5			0.961	0.961	0.937	0.918	0.898	0.880	0.863	0.848
3			0.962	0.957	0.949	0.924	0.903	0.885	0.867	0.851
4				0.958	0.954	0.934	0.915	0.894	0.875	0.857
5					0.955	0.953	0.927	0.904	0.884	0.865
6					0.962	0.953	0.941	0.911	0.891	0.872
7						0.958	0.954	0.924	0.901	0.881
8						0.967	0.956	0.937	0.912	0.888
9							0.962	0.957	0.926	0.897
10							0.971	0.961	0.936	0.909

Table D9.9.3-1(b) Coefficient of Correction C for Steam Temperature and Pressure (2)

Absolute pressure	Temperature (℃)										
(*MPa*)	400	420	440	460	480	500	520	540	560	580	600
0.5	0.822										
1	0.825										
1.5	0.828										
2	0.830	0.817	0.804	0.792	0.780	0.768					
2.5	0.833	0.819	0.806	0.793	0.782	0.770					
3	0.836	0.822	0.808	0.795	0.783	0.774	0.763	0.718	0.742	0.730	0.721
4	0.841	0.826	0.813	0.799	0.787	0.775	0.763	0.755	0.744	0.735	0.725
5	0.848	0.832	0.817	0.803	0.790	0.778	0.766	0.755	0.747	0.737	0.723
6	0.854	0.838	0.822	0.808	0.794	0.781	0.769	0.758	0.747	0.739	0.729
7	0.861	0.844	0.827	0.812	0.798	0.785	0.772	0.761	0.749	0.739	0.731
8	0.868	0.850	0.833	0.817	0.802	0.789	0.776	0.763	0.752	0.741	0.731
9	0.876	0.856	0.838	0.822	0.807	0.792	0.779	0.766	0.754	0.743	0.733
10	0.883	0.863	0.844	0.827	0.811	0.796	0.782	0.769	0.757	0.745	0.735

Note:

Pressures in this table correspond to $(1.03P + 0.1)$ in the calculation formula.

D10 PRESSURE VESSELS

D10.2 Materials and Welding

D10.2.1 Materials

The interpretation of the meaning of the pressure parts of pressure vessels specified in **10.2.1-1, Part D of the Rules,** which are required to use materials complying with the requirements in **Part K of the Rules,** is to be based on the information given in **D9.2.1** of this Guidance.

D10.2.6 Non-destructive Testing for Cast Steels and Cast Irons

The criteria for non-destructive tests in cases where cast steels are used for shells of Group I or Group II pressure vessels are to be in accordance with the following:

(1) The testing method of radiographic tests is to be in accordance with *JIS G* 0581. Any cracks or insufficient fusion are to be judged unacceptable. Gas and blowholes, sand spots, inclusions, and internal shrinkage are judged acceptable in cases where the defects are classified as Grade 1 in accordance with the above standard. Gas and blowholes, sand spots, and inclusions of Grade 2 are judged acceptable in cases where the thickness of Group II pressure vessels in defect area is 25 *mm* or more.

(2) The testing method and the criteria for acceptable defects detected by magnetic particle tests are to be in accordance with those in **D9.2.4(2)**.

(3) The testing method of liquid penetrant testing is to be in accordance with *JIS Z* 2324. The criteria for acceptable defects detected by liquid penetrant testing is to be in accordance with those in **D9.2.4(2)**.

(4) Defects judged unacceptable by **(1)**, **(2)**, or **(3)** may be repaired. Welding for such repairs is to be in accordance with the requirements specified in **5.1.11, Part K of the Rules**.

D10.3 Design Requirements

D10.3.2 Design Loads

When designing pressure vessels, the load or external force specified in **10.3.2-1** and **-2, Part D of the Rules** is to be taken into account in the cases specified below:

(1) In cases where the static head of contained fluid cannot be disregarded in the strength calculation of **10.5, Part D of the Rules**.
In this case, the value of P_0+P_1 (P_0: design pressure, P_1: static head of contained fluid) is to be used as the internal pressure at the part to be examined in the strength calculation concerned to the membrane stress.

(2) In cases where the internal pressure may become lower than the external pressure during in service.
In this case, the calculation procedure is to be taken in accordance with Section VIII, Division 2, Appendix 2 of the “Boiler and Pressure Vessel Code” of *ASME*.

(3) In cases where dynamic loads caused by ship’s motions can be considered.
In this case, the dynamic loads are to be estimated under the conditions of inclination angle specified in **Table D1.1, Part D of the Rules**, and also the calculation procedure is to be taken in accordance with Section VIII, Division 2, Appendix 4 of the “Boiler and Pressure Vessel Code” of *ASME*.

(4) In cases where fatigue due to thermal stress cannot be disregarded.
In this case, the calculation procedure is to be taken in accordance with Section VIII, Division 2, Appendix 5 of the “Boiler and Pressure Vessel Code” of *ASME*.

(5) In cases where loads from fittings cannot be disregarded.
In this case, the calculation procedure is to be taken in accordance with Section VIII, Division 2, Appendix 4 and 5 of the “Boiler and Pressure Vessel Code” of *ASME*.

(6) In cases where loads of hydraulic test pressure cannot be disregarded.
In this case, the measure specified in **(1)** is to be applied, or the test pressure is to be taken such that any calculated value of primary general membrane stress due to test pressure is not more than 90% (135% for local membrane stress) of the yield point

or proof stress of the material at the test temperature.

D10.9 Tests

D10.9.1 Shop Tests

1 Pressure vessels for which hydraulic tests are considered necessary by the Society, as specified in **10.9.1-2(1)(b), Part D of the Rules,** are pressure vessels such as those given below:

(1) Pressure vessels in cases where the product of the design pressure (*MPa*) and internal capacity (m^3) exceeds 1.0

(2) Heat exchangers such as fresh water coolers, lubricating oil coolers, hydraulic oil coolers, lubricating oil heaters, fuel oil heaters, condensers, feed water heaters, air coolers, etc., and air tanks such as control air tanks, etc. which are necessary for the operation of the following installations as well as other essential pressure vessels:

(a) Main propulsion machinery and shafting;

(b) Boilers and thermal oil installations (main boilers, essential auxiliary boilers, and other boilers and thermal oil installations used for any fuel oil heating necessary for the operation of main propulsion machinery or cargo heating that is continuously required); or

(c) Electric generators and auxiliaries (excluding auxiliary machinery for specific use, etc.) and their prime movers.

2 Notwithstanding the requirements in **10.9.1, Part D of the Rules**, hydrostatic tests of heat exchangers fitted to engines having cylinder bores of 300 *mm* or less may be omitted (see **Table D2.6 of the Rules**).

D11 WELDING FOR MACHINERY INSTALLATIONS

D11.2 Welding Procedure and Related Specifications

D11.2.1 Approval of Welding Procedure and Related Specifications

The "detailed data" referred to in **11.2.1-3, Part D of the Rules** are, in general, to be as follows:

(1) Outline of plant facilities and equipment (outline of plant installations, type and number of important welding machines, outline of facilities for heat treatment and installations for test and inspection)

(2) Qualifications and number of welders

(3) Production records of conspicuous welded constructions

(4) Data covering the welding quality control system and working process standards

(5) Welding procedures intended to be tested, and type or name of product to which such welding procedures are to be applied

(6) Maximum plate thickness and maximum pipe wall thickness of products referred to above, kind and specification of material

(7) Draft proposal for welding procedure qualification tests (type of welding machine, welding rod, type of flux, welding conditions, welding procedures including preheating and post weld heat treatment are to be specified in the proposal. Also, type of test, sampling procedure of test specimens and dimensions of test specimens are to be specified.)

D11.2.2 Execution of Tests

1 Approval tests for welding procedures and related specifications that fall under **11.2.2(1), Part D of the Rules** are to comply with the following requirements. For items not specified in the following requirements, **4.1.3** and **4.2** to **4.6, Part M of the Rules** are to be applied correspondingly. In cases where it is difficult to meet the above requirements, approval tests are to be as deemed appropriate by the Society.

(1) Selection of welding consumables

In general, a welding consumable for which the requirements related to strength (i.e. yield point or proof stress and tensile strength) of deposited weld metal is higher than strength of base metals and which resemble to base metals in the chemical composition is to be selected.

(2) Tests for butt welded joints

(a) The kinds of tests, the areas subjected to tests and the number of specimens is to be in accordance with the requirements specified in **Table D11.2.2-1**.

(b) The values of minimum mean absorbed energy are to comply with the requirements of base metals. In addition, testing temperatures are to be lower than the testing temperatures required for base metals.

(c) The Vickers hardness measured by hardness tests is, as a standard, to comply with the values specified in **Table D11.2.2-2** depending on the requirements related to the yield point or proof stress of base metals.

(3) Tests for fillet weld joints, T-joints with full penetration and T-joints with partial penetration

(a) For the number of specimens for hardness tests, the requirements specified in **Table D11.2.2-1** are to be applied correspondingly.

(b) The Vickers hardness measured by hardness tests is to be in accordance with **(2)(c)**.

2 Approval tests for welding procedures and related specifications that fall under **11.2.2(2), Part D of the Rules** are to be comply with the following requirements. For items not specified in the following requirements, **4.1.3** and **4.2** to **4.6, Part M of the Rules** are to be applied correspondingly.

(1) Test assemblies

Test assemblies are to be of the same or equivalent material used in the actual welding work. Additionally, the thickness of the test assemblies is, in principle, to be equal to the maximum thickness of the materials to be used in the actual welding work.

(2) Tests for butt welded joints

(a) The kinds of tests, areas subjected to tests and the number of specimens is to be in accordance with the requirements specified in **Table D11.2.2-3**.

(b) Test specimens are to be collected in accordance with **Fig. D11.2.2-1**.

(c) Minimum mean absorbed energy values for impact tests are to be in accordance with **-1 (2)(b)**.

(d) The Vickers hardness measured by hardness tests is to be in accordance with **-1 (2)(c)**.

(e) Notwithstanding **(a)** above, in principle, creep tests or high temperature tensile tests are to be added as a reference in cases where deemed necessary by the Society for the welding work of components used at temperatures higher than 1/2 of the melting points (absolute temperature) of base metals or the welding consumables, whichever is lower. Creep tests are to be performed in accordance with *ISO* 204, *JIS Z* 2271 or equivalent standards, and high temperature tensile tests are to be performed in accordance with *ISO* 6892-2, *JIS Z* 0567 or equivalent standards.

(3) Tests for fillet weld joints

The kinds of tests to be conducted are finished inspections, macro-structure inspections, hardness tests and fracture tests.

3 Approval tests for welding procedures and related specifications that fall under **11.2.2(3), Part D of the Rules** are to be comply with **-2**. However, the Society may require other test conditions or other kinds of tests where deemed necessary.

Table D11.2.2-1 Kinds of Tests, Areas Subjected to Tests and Number of Specimens
(Approval Tests for Welding Procedures and Related Specifications Applicable to Welding Work for Windlasses)

Kind of test	Areas subjected to tests or number of specimens
Visual inspection	Whole length of welding joints
Non-destructive inspection	Whole length of welding joints (1)
Tensile test	2
Bend test	4(2)
Impact test (sets)	3–8(3)(4)
Macro-Structure inspection	1
Hardness test	1(5)

Notes:

(1) Internal inspections by radiographic examination or ultrasonic examination, and surface inspections by magnetic particle examination or liquid penetrant examination are to be carried out.

(2) Two specimens are to be taken from the root bend and face bend respectively. However, the root and face bends may be substituted for by four side bends for plates and pipes (or tubes) with a thickness of 12 *mm* or more regardless of **Table D11.2.2-1**.

(3) Impact tests may be omitted when welding base metals which have no requirements related to testing temperature during impact tests and minimum mean absorbed energy.

(4) **Fig. M4.2** and **Fig. M4.3, Part M of the Rules** are to be applied correspondingly to the position of the notch of the impact test specimen.

(5) Hardness tests may be omitted when welding austenitic stainless steels or materials for which requirements related to yield point or proof stress is less than 355 N/mm^2.

Table D11.2.2-2 Requirements of Hardness Test

Requirements related to yield point or proof stress of a base metal(1) (N/mm^2)	Vickers hardness (*HV*10)
420 or less	350 max
more than 420 but 690 or less	420 max

Note:

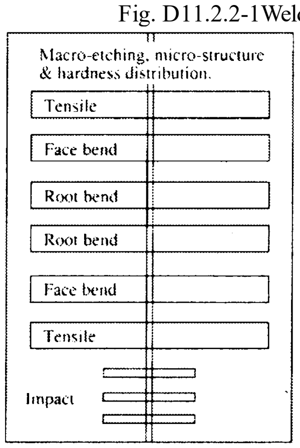
(1) In cases where the value is more than 690 N/mm^2, Vickers hardness is to be a value deemed necessary by the Society.

Table D11.2.2-3 Kinds of Tests, Areas Subjected to Tests and Number of Specimens
(Approval Tests for Welding Procedures and Related Specifications Applied to Welding Work for Boilers, Pressure Vessels, Principal Components of Prime Movers, etc.)

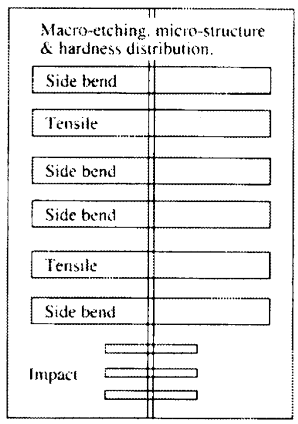
Kinds of test	Areas subjected to tests or number of specimens
Visual inspection	Whole length of welding joints
Radiographic examination	Whole length of welding joints
Tensile test	2
Bend test	4(1)
Impact test (sets)	3(2) (3)
Macro-Structure inspection	1(4)
Microscopic examinations	- (4)
Hardness test	1

Notes:
(1) In cases where it is difficult to collect a specified number of test specimens due to tube diameter, the number of each bend test specimen may be reduced by half.
(2) Impact tests may be omitted when welding base metals which have no requirements related to testing temperature during impact tests or minimum mean absorbed energy.
(3) Impact test specimen notch position is to be at the center of weld, on the fusion line and in the heat affected zone.
(4) To be conducted at the center of weld, on the fusion line and in the heat affected zone.

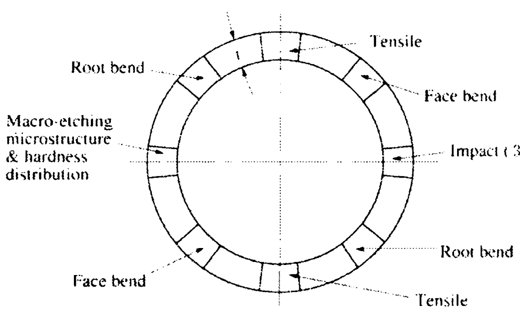
Fig. D11.2.2-1Welding Procedure Qualification Test Assemblies



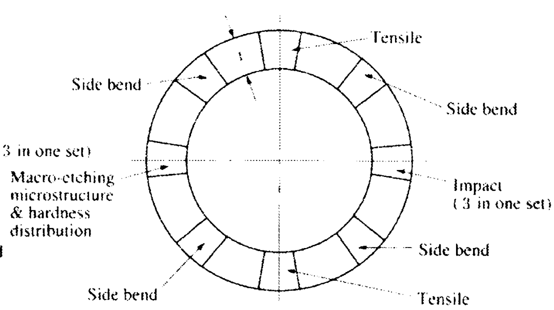
Test Assembly up to 20 mm in Thickness



Test Assembly over 20 mm in Thickness



Test Assembly for Pipes up to 20 mm in Thickness



Test Assembly for Pipes over 20 mm in Thickness

D11.2.3 Range of Approval

1 For welding procedures and related specifications that fall under **11.2.1-1(1)(a), Part D of the Rules**, the range of approval related to the kind of base metal is to be in accordance with the following requirements; however, **4.1.4, Part M of the Rules** is to be correspondingly applied for ranges of approval other than that for the kind of base metal. In cases where it is difficult to meet the above requirements, the ranges of approval are to be as deemed appropriate by the Society.

(1) In cases where the approval test is conducted using the materials specified in **Part K of the Rules** as base metals

(a) The welding procedures may be considered applicable to the materials specified in **Part K of the Rules** in accordance with **4.1.4, Part M of the Rules**.

(b) In addition to **(a)**, Where approved by the Society, the welding procedures may be considered applicable to materials not specified in **Part K of the Rules** in accordance with the group and the subgroup of base metals during approval tests as well as the range of approval related to the kind of base metals specified in *ISO* 15614-1.

(The group and the subgroup of materials are to be in accordance with *ISO/TR* 15608. This also applies throughout the rest of requirement **-1**.)

(c) With respect to **(b)**, in cases where a welding heat input is greater than 50 *kJ/cm*, the welding procedures may be considered applicable only to materials in the same group and the same subgroup as the materials to which the welding procedures are considered applicable in **4.1.4, Part M of the Rules**.

(2) In cases where the approval test is conducted using materials not specified in **Part K of the Rules** as a base metal

(a) Where approved by the Society, the welding procedures may be considered applicable to materials not used as the test assembly in accordance with the group and the subgroup of base metals during approval tests as well as the range of approval related to kind of base metals specified in *ISO* 15614-1.

(b) With respect to **(a)**, in cases where a welding heat input is greater than 50 *kJ/cm*, the welding procedures may be considered applicable only to materials in the same group and in the same or one subgroup below that of the test assembly.

2 For the welding procedures and related specifications that fall under **11.2.2(2)** and **(3), Part D of the Rules**, the upper limit of the range of approval related to thickness is, in principle, to be a value same as the thickness of test assembly; however, **4.1.4, Part M of the Rules** is to be correspondingly applied for the range of approval for the thickness.

D11.3 Post Weld Heat Treatment

D11.3.1 Procedure of Post Weld Heat Treatment

The wording "specially considered by the Society" specified in **11.3.1-2, Part D of the Rules** means as follows:

(1) The temperature to be maintained in the post weld heat treatment is to be as given in **Table D11.3.1-1**.

(2) The requirements in **11.3.1-1, Part D of the Rules** apply to procedures of heat treatments other than post weld heat treatments for alloy steel referred to above.

Table D11.3.1-1 Post-welding Temperature to be Maintained

Kind of steel	Minimum temperature to be maintained (℃)
2-2.5*Ni* steel 3.5*Ni* steel	600

Note:
For 5-9*Ni* steel, the post weld heat treatment is not required.

D11.4 Welding of Boilers

D11.4.3 Post Weld Heat Treatment

The wording "welded parts specially approved by the Society" referred in **11.4.3-1(3), Part D of the Rules** means the parts complying with the following conditions:

(1) Plate material is *KP*42, *KP*46 or *KP*49, and plate thickness is 19 *mm* or less.

(2) The Charpy *U*-notch impact test value of the base metal and welded joint through the use of test specimen *U*4 at a temperature of 0℃ is to be 27.5 *J* or more. Regarding the values of welded joints, an impact test is to be added to the production weld tests in order to verify the values of welded joints.

(3) The joints between shells and end plates or tube plates are to be butt weld.

D11.4.4 Production Weld Tests

The wording "bend test jig which deemed appropriate by the Society" referred to in **11.4.4-4(2), Part D of the Rules** means those bend test jigs specified in **Fig.M3.1**, **Fig.M3.2** and **Fig.M3.3**, **Part M of the Rules** or equivalent thereto.

D11.4.5 Non-destructive Testing for Longitudinal and Circumferential Joints

1 The criteria for evaluating radiographic testing are as follows:

(1) Classification of defects

Defects shall be classified into 4 types in accordance with **Table D11.4.5-1**.

(2) Type 1 defects

Type 1 defects are to be rejected, if the score of a particular defect exceeds the value of the acceptable score specified in **Table D11.4.5-2** according to thickness of the base metal. However, the score of one defect is determined on the basis of the axis length of the defect shown in **Table D11.4.5-3**. In cases where the axis length of a defect is shorter than the value specified in **Table D11.4.5-4**, the score may be uncounted. The score of two or more defects is to be sum of the scores for each defect in the sight of test field.

(3) Type 2 defects

Type 2 defects are to be rejected, if the length of a defect exceeds the value of the acceptable score specified in **Table D11.4.5-5** according to thickness of the base metal. The length of defect is to be determined by measuring the length of a defect. However, in cases where the defects are present in a row and the mutual distance between the defects does not exceed the length of the larger defect, the length of all defects including the spaces between them is to be measured as the length of the defect specified in **Table D11.4.5-5**.

(4) Type 3 defects

Any type 3 defect is to be rejected.

(5) Type 4 defects

The acceptable criteria and score of defects are to be according to the requirements specified in **(2)** (in this case, "type 1 defect" is to be read as "type 4 defect"). However, in cases where the type 4 defects coexistent with the type 1 defects in the sight of the test field, the score of defect is to be the sum of both scores.

2 The wording "other appropriate non-destructive testing" referred to in **11.4.5-8, Part D of the Rules** means the following **(1)** or **(2)**:

(1) The radiographic testing to be carried out in accordance with *ISO* 17636. The criteria and others that are not specified in the *ISO* are to be in accordance with **11.4.5, Part D of the Rules** and **D11.4.5-1**. In cases of the radiographic testing using no radiograph film, the testing plan is to be submitted to and approved by the Society, prior to the testing.

(2) The ultrasonic testing to be carried out in accordance with **11.4.6, Part D of the Rules** and **D11.4.6-2**. In this case, **8.1.2-5, Part M of the Rules** is to be applied.

Table D11.4.5-1 Classification of Defects

	Kind of defects
Type 1	Round blow holes and similar defects
Type 2	Elongated slag inclusions, pipes, incomplete penetration, incomplete fusion, and similar defects
Type 3	Cracks and similar defects
Type 4	Tungsten inclusions

Table D11.4.5-2 Acceptable Criteria for Type 1 Defects

Thickness of base metal (*mm*)	10 or less	More than 10, 25 or less	More than 25, 50 or less	More than 50
Score of one defect	3	6	12	15
Test field of vision (*mm*)	10×10		10×20	

Table D11.4.5-3 Score of Type 1 Defects

Axis length of one defect (*mm*)	1.0 or less	More than 1.0, 2.0 or less	More than 2.0, 3.0 or less	More than 3.0, 4.0 or less	More than 4.0, 6.0 or less	More than 6.0, 8.0 or less	More than 8.0
Score	1	2	3	6	10	15	25

Table D11.4.5-4 Maximum Axis Length of Type 1 Defects for an Uncountable Score

Thickness of base metal (*mm*)	Axis length of one defect (*mm*)
20 or less	0.5
More than 20, 50 or less	0.7
More than 50	1.4% of thickness of base metal

Table D11.4.5-5 Acceptable Criteria for Type 2 Defects

Thickness of base metal (*mm*)	Length of defect (*mm*)
12 or less	3
More than 12, 48 or less	1/4 of the base metal thickness
More than 48	12

D11.4.6 Non-destructive Testing for Other Welds

1 The wording "important welds other than those specified in **11.4.5 of the Rules**" referred to in **11.4.6, Part D of the Rules** means, for example, the following parts with a plate thickness of 6 *mm* or more:

(1) Welds between flat end plates or cover plates and shell plates

(2) Welds between furnaces or ogee rings and shell plates

(3) Welds for manholes

(4) Welds for nozzles

2 The standards for ultrasonic testing are to be in accordance with the following:

(1) The testing method is to be in accordance with *JIS Z* 3060 (1994) or equivalent thereto.

(2) Any indicated defect length according to L-line sensitivity specified in the method in **(1)**, which exceeds the value given in **Table D11.4.6-2** with respect to plate thickness, is not acceptable. Two or more defects existing at a same depth, separated by an interval shorter than the length of the largest defect are to be regarded as a continuous defect which includes the interval between them.

3 The standards for magnetic particle tests are to be in accordance with the following:

(1) The testing method is to be in accordance with *JIS Z* 2320-1 to -3 or equivalent thereto.

(2) The result of a magnetic particle test is acceptable, provided the following **(a)** through **(d)** are complied with.

(a) There are no magnetic particle indications due to surface cracks.

(b) The maximum length of linear magnetic particle indications is 2 *mm* or less.

(c) The major axis length of circular magnetic particle indications is 2 *mm* or less.

(d) The point total specified in **Table D11.4.6-3** with respect to the type of magnetic particle indications is 6 or less within an area of 2500 mm^2 for scattering magnetic particles.

4 The standards for liquid penetrant tests are to be in accordance with the following:

(1) The testing method is to be in accordance with *JIS Z* 2343 or equivalent thereto.

(2) The criteria for acceptable defects detected by liquid penetrant testing are to be in accordance with those specified in **-3(2)**.

5 The criteria for acceptable defects detected by the radiographic testing are to be in accordance with **D11.4.5-1**.

6 The wording "another appropriate method" referred to in **11.4.6-2, Part D of the Rules** means those specified in **D11.4.5-2(1)**.

Table D11.4.6-2 Acceptable Criteria for Indicated Defect Length

Plate thickness t (mm)	Length of defect (mm)
$t < 12$	3
$12 \leq t < 48$	$t/4^{(1)}$
$48 \leq t$	12

Note:

(1) t is the plate thickness on the open edge side of the base material. However, in cases where the value of the thickness of the base material and value of the thickness of the section at the butt welding are different, the lesser of the two values is to be taken as the plate thickness.

Table D11.4.6-3 Points for Scattering Magnetic Particle Indication

Magnetic particle pattern of defect	Magnetic particle indications of 2 mm or less
Linear	3
Circular	1

D11.5 Welding of Pressure Vessels

D11.5.3 Stress Relieving

1 The mechanical stress relieving is, in principle, to be given by hydraulic means in cases where the applicable plate thickness is to be 40 mm or less.

2 The required conditions for omitting stress relieving in cases where material having superior notch toughness is used are to be as specified below:

(1) The base metal is to be of steel plate with a rule required impact test value of 47.1 J or more by the use of test specimens $U4$ at a temperature of 0 ℃.

(2) The impact test value of welds in the production weld tests is not to be less than the rule required value of the base metal at a temperature of 0 ℃.

(3) The plate thickness of the material is to be 40 mm or less.

D11.5.5 Non-destructive Testing for Welded Joints

1 The criteria for acceptable defects detected by the radiographic testing are to be in accordance with **D11.4.5-1.**

2 The wording "another appropriate method" referred to in **11.5.5-3, Part D of the Rules** means those specified in **D11.4.5-2(1)**.

3 The ultrasonic testing is to be in accordance with **D11.4.6-2.**

D11.5.6 Non-destructive Testing for Other Welded Parts

1 The criteria for acceptable defects detected by the radiographic testing are to be in accordance with **D11.4.5-1**.

2 The testing procedures and criteria for acceptable defects detected by the ultrasonic testing are to be in accordance with **D11.4.6-2**.

3 The testing procedures and criteria for acceptable defects detected by the testing are to be in accordance with **D11.4.6-3** and **D11.4.6-4** respectively for magnetic particle testing and liquid penetrant testing.

D11.6 Welding of Piping

D11.6.5 Non-destructive Testing

1 When the non-destructive testing specified in **11.6.5, Part D of the Rules** is carried out, test plans are to be submitted to the Society for approval prior to testing in accordance with **2.1.7-1(2), Part B of the Rules**.

2 The criteria for acceptable defects detected by the radiographic testing are to be in accordance with **D11.4.5-1**.

3 The ultrasonic testing is to be in accordance with **D11.4.6-2**.

4 The magnetic particle testing and liquid penetrant testing are to be in accordance with **D11.4.6-3** and **D11.4.6-4** respectively.

5 The "when the conditions are such that a comparable level of weld quality is assured" referred to **11.6.5-1(1)(d), Part D of the Rules** means, for example, such case wheres well-documented quality assurance procedures and records are available to enable the Society to assess the ability of the manufacturer to produce satisfactory automatic welds consistently.

6 The wording "another appropriate method" referred to in **11.6.5-4, Part D of the Rules** means those specified in **D11.4.5-2(1)**.

D12 PIPES, VALVES, PIPE FITTINGS AND AUXILIARIES

D12.1 General

D12.1.5 Service Limitations for Materials

1 The wording "deemed acceptable by the Society" in **12.1.5-1(4), Part D of the Rules** means, for example, as follows:

(1) For control oil piping used to control each valve in machinery spaces, copper alloy pipes may be used.

(2) For instrumentation piping for saturated steam piping with a design pressure of 1.6 *MPa* or less, copper pipes and copper alloy pipes may be used.

(3) For thermal oil piping used for heat tracing of F.O. piping in machinery spaces, which are provided with appropriate protection, copper pipes may be used.

2 The wording "where deemed as appropriate by the Society after consideration has been given to their construction and purpose" in **12.1.5-2(3)(b), Part D of the Rules** means in cases where cast iron valves, which have a rigid construction and a design breaking pressure of not less than 10 times the maximum working pressure, are provided in hydraulic piping systems to control the direction of the flow.

3 The wording "where deemed as appropriate by the Society after consideration has been given to their construction and purpose" in **12.1.5-2(4), Part D of the Rules** means in cases where cast iron valves, which have a rigid construction and a design breaking pressure of not less than 5 times the maximum working pressure, are provided in hydraulic piping systems to control the direction of the flow.

D12.1.6 Use of Special Materials

1 The wording "requirements specified otherwise" in **12.1.6, Part D of the Rules** means as follows.

(1) In cases where rubber hoses, Teflon hoses or nylon hoses are used for the following pipes, materials approved in accordance with **Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use** are to be used.

(a) Pipes of Group I or Group II

(b) Pipes likely to cause fire or flooding in cases where they rupture

(2) Only plastic pipes (including vinyl pipes) approved by the Society in accordance with **Chapter 6, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use** are to be used.

(3) In cases where aluminum alloy pipes are used; the following requirements are to be complied with:

(a) As a rule, aluminum alloy pipes are to be in accordance with the requirements of the code deemed appropriate by the Society, and are to be of seamless drawn pipes or seamless extruded pipes.

(b) Aluminum alloy pipes are not to be used for any of the following applications:

i) As a rule, pipes with a design temperature exceeding 150 °C.

ii) Any pipes which penetrates either an "*A*-Class division" or a "*B*-Class division."

iii) Piping in which the use of copper alloy pipes is prohibited by **Table D12.2, Part D of the Rules**.

(c) The required thickness of aluminum alloy pipes subject to internal pressure are to be in accordance with the following requirements:

Pipe thickness is to be determined using the formula in **12.2.1-1, Part D of the Rules**. In this case, allowable stress (f) is to be the smallest of the following values. However, in cases where the design temperature is not in the creep region of the material, no consideration needs to be given to the value of f_3.

$$f_1 = \frac{R_{20}}{4.0},\quad f_2 = \frac{E_t}{1.5},\quad f_3 = \frac{S_R}{1.6}$$

where

R_{20} : Specified minimum tensile strength (N/mm^2) of the material at room temperature (less than 50 °C)

E_t : 0.2 % proof stress (N/mm^2) of the material at design temperature

S_R : Mean value of creep breaking stress (N/mm^2) of the material after 100,000 *hours* at design temperature

4 For tankers and ships carrying dangerous chemicals in bulk, aluminum alloy pipes and aluminized pipes are not to be used in any of the hazardous areas defined in **4.2.3-1** or **4.2.3-2, Part H of the Rules** of those ships intended to carry crude oil and petroleum products having a flashpoint not exceeding 60°C and having a Reid vapour pressure below atmospheric pressure or other liquid cargo having similar fire hazards. However, aluminized pipes may be permitted in the following hazardous areas mentioned above:

(1) inside ballast tanks;
(2) inside inerted cargo tanks; and
(3) hazardous areas on the open deck, in cases where the pipes are appropriately protected from accidental impact.

D12.2 Thickness of Pipes

D12.2.2 Minimum Thickness of Pipes

1 In cases where the requirement for minimum thickness of the corrosion resistant alloy steel pipes in **12.2.2-1, Part D of the Rules** is applied, the minimum thickness of any stainless steel pipes used for cargo oil pipes is to be the value specified in **S5.1.6-1, Part S of the Guidance**.

2 The "fresh water pipes" in **Table D12.6(1), Part D of the Rules** means those fresh water pipes used for boiler feed water and drinking water. For other fresh water pipes, notwithstanding the requirements specified in **Table D12.6(1), Part D of the Rules**, those pipes with the thickness of Ⓕ specified in **Table D12.6(2), Part D of the Rules** may be applied to those pipes. However, pipes with a thickness less than 6 *mm* may not be used. The hot water pipes used for heating oil tanks are not regarded as fresh water pipes.

3 The minimum wall thickness for pipes whose nominal diameter is more than 450 *mm* in **Table D12.6(2)**, **Part D of the Rules** is to be in accordance with national or international standards. In this case, the wall thickness is not to be less than the minimum value indicated in the appropriate column of **Table D12.6(2)** for those pipes of 450 *mm* in nominal diameter.

D12.3 Construction of Valves and Pipe Fittings

D12.3.1 General

Rubber seated butterfly valves are to be dealt with under the following requirements:

(1) Application

Rubber seated butterfly valves (hereinafter referred to as the "butterfly valves") may, in principle, not be used for the following applications:

(a) Outlet valves fitted to tanks, carrying flammable or combustible liquids (*e.g.*, fuel oil, crude oil, etc.) and subjected to liquid head, installed in engine rooms or other areas susceptible to fire.
(However, butterfly valves may be used for those valves installed within the cargo oil tanks or as outlet valves leading to the pump rooms of oil tankers.)

(b) Valves in piping systems with a design pressure exceeding 1.6 *MPa*

(c) Valves in piping systems with a design temperature exceeding 70°C

(d) Valves in piping systems handling special liquids other than water and oil

(e) Valves in the flammable oil piping systems within engine rooms in cases where they have such a construction that the internal rubber lining is extended to the abutting face of a flange in order to be used as a gasket.

(2) Construction

The construction of the butterfly valves is to conform to the following requirements:

(a) A stopper, which can be engaged at designated "Open" and "Shut" positions, is to be provided.

(b) Valves serving at an intermediate valve disc position are to be able to maintain their position when locked and the locking system is not to be loosened between "Open" and "Shut" by vibrations, mechanical impacts or liquid flows, etc.

(c) The valve is to be able to be operated by a single person.

(d) Means are to be provided to indicate valve disc position.

(e) Valve stems are to be of sufficient strength and valve discs are to be fitted to valve stems in such a way of that there is no possibility of loosening.

(f) Sufficient consideration is to be given to the corrosion resistance and wear resistance properties of all materials used for the main parts of valves.

(g) Butterfly valves used as seawater suction valves or overboard discharge valves are, in principle, to be of a flange type.

(3) Product markings of

The butterfly valve is to be marked with the following items at a conspicuous place on the product:

(a) Design pressure

(b) Valve box material

(c) Nominal diameter

(d) Name of manufacturer

D12.3.3 Mechanical Joints

1 The wording "type approved by the Society" referred to in **12.3.3-2, Part D of the Rules** means one whose approval of use is obtained in accordance with **Chapter 9, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

2 Details of the pressure referred to in **12.3.3-5, Part D of the Rules** are specified in **9.3.2(4)** of **Chapter 9, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

3 The wording "standards separately specified by the Society" referred to in **12.3.3-7, Part D of the Rules** refers to **Chapter 9, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

4 The wording "where deemed necessary by the Society" referred to in **(2)** and **(4)** as well as **(6)** to **(8)** of **12.3.3-7, Part D of the Rules** is in accordance with **Table 6.9-1** of **Chapter 9, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

D12.3.4 Flexible Hose Assemblies

1 The wording " approved by the Society" referred to in **12.3.4-2, Part D of the Rules** means one whose approval is obtained in accordance with **2.4.2-11, Chapter 2, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

2 The wording "exposed open decks" in **12.3.4-3, Part D of the Rules** means "open decks" as defined in **9.2.3-2(10), Part R of the Rules**, excluding spaces in the cargo areas of tankers, ships carrying liquefied gases in bulk and ships carrying dangerous chemicals in bulk defined in **3.2.6, Part R**, **1.1.4(6), Part N** and **1.3.1(4), Part S of the Rules**.

3 The wording "where specially approved by the Society" referred to in **12.3.4-3(3)(a) of the Rules** refers to the use of materials such as Teflon and nylon which are unable to be reinforced. The hoses, however, are to have external wire braid protection as practicable.

D12.4 Connection and Forming of Piping Systems

D12.4.1 Welding of Piping Systems

1 The welding work for a pipe referred to in **12.4.1-1, Part D of the Rules** includes butt welded joints of direct connection, slip-on sleeve welded joints and welded joints between pipes and pipe flanges, etc.

2 The "standards recognized by the Society" referred to in **12.4.1-2(1), Part D of the Rules** means national or international standards such as *JIS* and *ISO*.

3 The "deposited weld metal tests" referred to in **12.4.1-2(2), Part D of the Rules** means the deposited metal test specified in **Chapter 6, Part M of the Rules** or an equivalent test. The tests are to be carried out at the same time as the tests for approval of welding procedures and related specifications.

D12.4.2 Direct Connection of Pipe Lengths

1 The "standards recognized by the Society" specified in **12.4.2-2(1), Part D of the Rules** refers to, for example, *JIS B* 2316.

2 The "toxic media" specified in **12.4.2-2(2)(b)** and **-3(2)(b), Part D of the Rules** refers to, for example, media categorized as toxic gases or toxic substances by the *IMDG* code, as defined in Chapter VII, Regulation 1.1 of the *SOLAS*.

3 The "standards recognized by the Society" specified in **12.4.2-3(1), Part D of the Rules** refers to, for example, *JIS B* 2301, *JIS B* 2302, *JIS B* 2308, *ASME* B31.1 and *ASME* B31.3.

4 An example of "the Society may allow use for pipes specified in **(e)** or **(f)** after considering the service" specified in **12.4.2-3(2), Part D of the Rules** is that of the fixed local application fire-fighting system specified in **10.5.5, Part R of the Rules**. In cases such as this, all of the relevant joints are to be in compliance with recognized national and/or international standards.

D12.4.3 Connection of Pipes with Pipe Fittings

1 The following pipe joints may be used as those "deemed appropriate by the Society" referred to in **12.4.3-1, Part D of the**

Rules:

(1) Types of pipe joints with bell-mouthed pipe ends as shown in Fig. D12.4.3-1 may be used for pipes in Group III and pipes in Group I or II with a design pressure of 1.0 *MPa* or less and with a nominal diameter of 50 *A* or less.

(2) One side welded flange joints shown in Fig. D12.4.3-2 may be used for drinking water piping, scupper piping and sanitary piping located above the load line as well as overflow piping, air vent piping, exhaust gas piping, gas vent piping of crank chambers, exhaust steam piping and foam fire extinguishing agent discharge piping having open ends. Furthermore, pipes in Group III which are used in ways other than those given above may be used for pipes with a nominal diameter of 40*A* or less except for those pipes used for flammable oils.

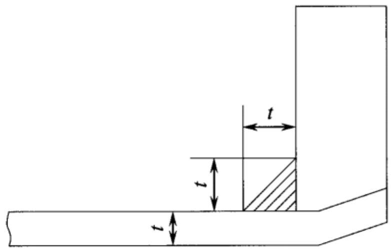
2 In cases where non-ferrous metal valves and fittings are soldered to non-ferrous metal piping referred to in 12.4.3-2, Part D of the Rules, the procedures for soldering copper pipes and pipe flanges are to be as shown below:

(1) The portion to be soldered is to be provided with a suitable molten pool, and the pipe end is to be bell-mouthed.

(2) Fillet welding is not recommendable for connecting copper pipes with pipe flanges. However, this recommendation may be waived when a special soldering method such as silver soldering or *TIG* welding is applied.

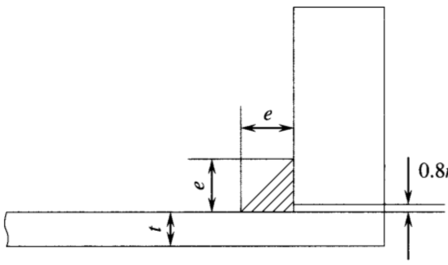
(3) Copper pipes connected by soldering may be used in cases where the design temperature is 200℃ or below.

Fig. D12.4.3-1 Flange Joint (Bell-mouth)



t: required thickness of pipe

Fig. D12.4.3-2 Flange Joint (One-side Welded)



e: 1.4*t*
t: required thickness of pipe

D12.5 Construction of Auxiliary Machinery and Storage Tanks

D12.5.1 General

1 Plate thickness of fuel oil storage tanks

The "small tanks" specified in 12.5.1-2, Part D of the Rules means fuel oil storage tanks with a capacity of 1,000 *litres* or less.

D12.6 Tests

D12.6.1 Shop Tests

1 Testing of pipe joints of a butt welded type and pipe joints of a slip-on sleeve welded type (such as elbows, reducers, tees, bends and sockets, etc.)

(1) Materials and tests of pipe joints of a butt welded type and pipe joints of a slip-on sleeve welded type used for Group I or II pipes are to be in accordance with the following:

(a) Materials

i) Materials for pipe joints are to comply with the requirements in **Part K** (see **D1.1.4(7)**).

ii) Notwithstanding the requirement given in **i)**, materials complying with international or national standards such as *ISO*, *JIS*, etc. may be used for pipe joints for which hot forming or heat treatment is carried out during the manufacturing process, provided that they receive approval of use from the Society in accordance with **Chapter 12, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

(b) Material tests

Pipe joints for which hot forming or heat treatment is carried out during the manufacturing process are to be subject to the following tests after hot forming or heat treatment.

i) Tensile tests, bend tests and flattening tests, etc. for the pipes as specified in **Part K** are to be carried out pipe joints of the preceding **(a)i)** and **ii)** selected at random.

(c) Production weld tests

In cases where welding is carried out during the manufacturing process, **Chapter 11, Part D of the Rules** is to be applied mutatis mutandis to pipe joints.

(d) Visual inspections and dimension inspections

Visual inspections and dimension inspections are to be carried out on pipe joints after hot forming or heat treatment.

(e) Omission of surveyor attendance

i) A Society surveyor need not be present during the tests specified in **(b)** to **(d)** in cases where the pipe joints are intended to be used for pipes with a nominal diameter less than 100 *mm*, or for pipes with a design pressure less than 3 *MPa* and a design temperature less than 230°C.

ii) With respect to pipe joints other than those specified in **i)** and **(1)(a)ii),** a Society surveyor need not be present during the tests specified in **(b)** to **(d)** when the requirements in **Chapter 4, Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use** are satisfied.

iii) With regard to the pipe joints specified in **(1)(a)ii),** a Society surveyor need not be present during the tests specified in **(b)** to **(d)**.

(2) Certificates

In cases where a surveyor is not present during testing in accordance with the requirements in preceding **(1)(e)ii)** and **(1)(e)iii)**, records for the material tests, production weld tests, visual inspections and dimension inspections specified in the preceding **-1(1)** are to be submitted to the Society. Such reports may be used in lieu of approval certificates if they are signed by a Society surveyor and stamped with the Society's seal of approval.

(3) Marking of pipe joints which have passed relevant tests

Pipe joints, except for those in preceding **(1)(e)i),** are to be marked with the Society's brand as well as information such as material used, dimensions, name of manufacturer etc. by means of stencils or metal tags (for small-diameter pipe joints).

2 The Society may waive the presence of the Surveyor at the hydrostatic tests required by in **12.6.1-2** and **-3, Part D of the Rules** for small bore pipes (less than about 15 *mm*), depending on the application.

3 The term "integral" referred to in **12.6.1-2** and **-3, Part D of the Rules** means, for example, welded fittings.

4 The "non-integral" referred to in **12.6.1-5, Part D of the Rules** means one which does not fall under that specified in **-3** above.

5 The wording "standards recognized by the Society" referred to in **12.6.1-5, Part D of the Rules** means national standards such as Japanese Industrial Standards or standards of an authorized body as well as other standards proven to be appropriate. For valves, however, *JIS B* 2003 or *ISO* 5208, or other equivalent standards are to be applied.

6 The wording "free standing fuel oil storage tanks" referred to in **12.6.1-8, Part D of the Rules** means those free standing tanks storing the following liquids:

(1) Fuel oil for main propulsion machinery, prime movers for driving generators (including emergency generators) and boilers, etc.

(2) Fuel oil additives

(3) Washing oil (light oil, kerosene, etc.)

D13 PIPING SYSTEMS

D13.1 General

D13.1.2 Drawings and Data

1 Piping diagrams of those tanks which form part of the hull construction are to be accompanied by a piping list that follows the format given in **Table D13.1.2-1**.

2 Regarding scupper piping system drawings, the following items are to be specified:

(1) Summer load line and Tropical load line determined by the requirements in **Part V of the Rules.** However, instead of the Summer load line, a maximum designed load line that is located higher than it may be acceptable.

(2) The lines higher than the summer load lines mentioned in (1) above are: the line 600 *mm* above the Summer load line, line $0.01L_f$ and line $0.02L_f$

(3) The line 450 *mm* below the freeboard deck

3 Regarding those distance pieces directly fitted to the sides of ships, drawings of their construction and fitting details are to be submitted for approval.

Table D13.1.2-1

Tank	Name
	Type
Sounding pipes	Nominal diameter
	Outside diameter
	Inside diameter
	Thickness
	Type
	Remarks
Air vent pipes (overflow pipes)	Nominal diameter
	Outside diameter
	Inside diameter
	Thickness
	Sum of section area
	Type
	Remarks
Filling pipes	Nominal diameter
	Outside diameter
	Inside diameter
	1.25 × sectional area
	Type
	Remarks

D13.2 Piping

D13.2.1 General

1 The term “cargo holds” referred to in **13.2.1-5, Part D of the Rules** does not include spaces within the cargo area specified in **3.2.6, Part R of the Rules**, **1.1.4(6), Part N of the Rules** and **1.3.1(4), Part S of the Rules** for tankers, ships carrying liquefied gases in bulk or ships carrying dangerous chemicals in bulk where cargoes are not carried (e.g. hold spaces).

2 Marking of pipes with "distinctive colours" referred to in **13.2.1-8(1), Part D of the Rules** is, in principle, to be carried out in accordance with *JIS F* 7005 "Identification of Piping Systems On Board Ships".

D13.2.3 Penetration of Pipes

Penetration of valve stems

Valve stems of various valves are, in principle, not to penetrate through the part subjected to liquid head such as the bottom plate of shoulder tanks and tank top of double bottom used for tanks. In cases where such penetrations are unavoidable, considerations are to be taken by providing such means as protection pipe to prevent liquid head from imposing on the stuffing box.

D13.2.4 Mechanical Joints

1 The wording "standard recognized by the Society" referred to in **13.2.4-4, Part D of the Rules** means, for example, Japanese Industrial Standards.

2 The term "cargo holds" referred to in **13.2.4-6(1), Part D of the Rules** includes hold spaces, defined in **1.3.1(13), Part S of the Rules** for tankers and ships carrying dangerous chemicals in bulk and **1.1.4(25), Part N of the Rules** for ships carrying liquefied gases in bulk.

3 The wording "approved by the Society" referred to in **13.2.4-6(1), Part D of the Rules** means, for example, the following **(1)** to **(4)**:

(1) Slip-on joints of the type approved by the Society are used for joints of bilge suction piping and ballast piping led into cargo holds.

(2) Slip-on joints are used for joints of suction pipes for double bottoms.

(3) Slip-on joints are used for cargo oil pipes, except for those within ballast tanks through which the pipes penetrate. (*See* **14.2.4-5, Part D of the Rules**)

(4) Slip-on joints are used for ballast pipes in ballast tanks adjacent to cargo oil tanks, except for those within cargo oil tanks through which the pipes penetrate. (*See* **14.2.7-4, Part D of the Rules**)

D13.2.5 Bulkhead Valves

1 With respect to the provisions of **13.5.10, Part D of the Rules**, bulkhead valves capable of being brought into operation from a readily accessible enclosed space, the location of which is accessible from the navigation bridge or continuously manned propulsion machinery control rooms without traversing exposed decks, may be accepted as an alternative to valves operable from above the freeboard deck required by the provisions of **13.2.5-2, Part D of the Rules**.

2 Pipes penetrating stern tanks are to be fitted with stop valves at the fore side of the bulkhead.

3 The requirements for pipes piercing collision bulkheads specified in **13.2.5-1 and -2, Part D of the Rules** apply only to those extending below the freeboard deck. However, in accordance with the provisions of **2.2.1.5(2), Part 1, Part C of the Rules**, those pipes piercing the extension part of the collision bulkhead (the weathertight part above the freeboard deck) and opening into enclosed spaces behind such bulkheads, are to be fitted with non-return valves on the aft side of the bulkhead.

4 The number of pipes piercing the collision bulkhead specified in **13.2.5-2, Part D of the Rules**, is to be in principle just one. Where the forepeak is divided to hold two different kinds of liquids, the Society may allow the collision bulkhead to be pierced below the freeboard deck by two pipes. However, the Society is satisfied that there is no practical alternative to the fitting of such a second pipe and, that having regard to the additional subdivision provided in the forepeak, the safety of the ship is maintained. In addition, valves~~,~~ complied with the requirements in **13.2.5-2, Part D of the Rules**, are to be fitted.

D13.3 Sea Suction Valves and Overboard Discharge Valves

D13.3.1 Sea Suction Pipes and Overboard Discharge Pipe Connections

The wording "up to an appropriate level above the freeboard deck" as specified in **13.3.1, Part D of the Rules** is to be in accordance with the provisions of **D13.4.1-3**.

D13.3.2 Location and Construction of Sea Suction Valves, Overboard Discharge Valves, etc.

1 The wording "overboard discharges" in **13.3.2-1, Part D of the Rules** means those openings discharging due to pressure from pumps and not including those discharging due to pressure from natural forces (i.e., gravity).

2 The wording "the locations of overboard discharges are not to be such that water can be discharged" specified in **13.3.2-1, Part D of the Rules** means areas other than that within the diagonal line section of **Fig. D13.3.2-1**.

3 The wording "special provisions" specified in **13.3.2-1, Part D of the Rules** means either of the following arrangements:

(1) Means to guide water flow to the shell plating with consideration being given to the direction of the water flow.

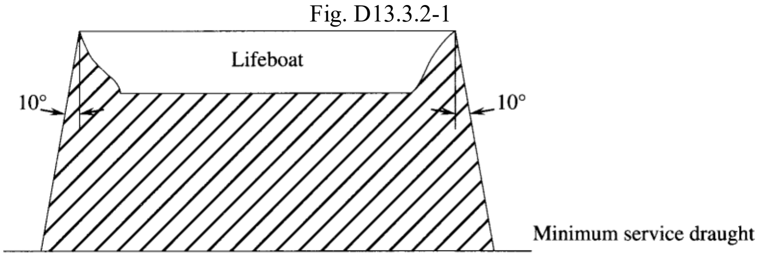
(2) Means to stop any water discharge which can be operated from a position on the weather deck in the vicinity of the lifeboat installation location. In this case, operating switches or operating handles that are different from those stopping devices specified in **5.2.1-2** and **5.2.2-2** through **5.2.2-4, Part R of the Rules** are to be provided.

4 The wording "rigid construction" in **13.3.2-3(2), Part D of the Rules** means that the pipe thickness of distance pieces is to be not less than the values given in the following table:

Table D13.3.2-1

de : Outside diameter (mm)	Wall thickness (mm)
$de \leq 80$	7.0
$80 < de \leq 180$	$0.03de + 4.6$
$180 < de \leq 220$	$0.0625de - 1.25$
$220 < de$	12.5

Fig. D13.3.2-1



D13.4 Scuppers, Sanitary Discharges, etc.

D13.4.1 General

1 Scupper pipes within superstructures

Scupper piping within the superstructure is not to be connected to any scupper piping on the weather deck.

2 Inboard open ends of scupper pipes

(1) In cases where the bilges of any small compartments at the stern of ships (i.e., steering engine compartments, boatswain's store, anchor chain compartments, etc.) are discharged by either hand pumps or educators, their scupper piping systems are to located at a higher position than all other piping systems.

(2) In cases where timber load lines are marked, the vertical distance to the inboard open end is to be measured from the timber Summer load line.

3 With respect to the provisions of **13.4.1-7, Part D of the Rules**, the range of wall thickness for pipes in accordance with **Table D12.6(1)** and **Table D12.6(2) of the Rules**, may be limited to:

(1) The area up to the freeboard deck, in cases where the vertical distance from the load line to the freeboard deck is not less than 600 mm; or

(2) The area to the deck just above the freeboard deck, in cases where such distance is less than 600 mm.

D13.5 Bilge and Ballast Piping

D13.5.1 General

1 Alternatives to or the omission of bilge piping

With respect to bilge piping required by **13.5.1-1, Part D of the Rules**, the Society may accept the other measures described in the following **(1)** and **(2)**.

(1) For spaces where it is difficult to install bilge piping, other drainage arrangements such as drain plugs may be allowed to be

installed as an alternative to the bilge piping.

(2) For small spaces where there is no risk of water accumulation, the omission of bilge piping may be allowed.

2 Bilge suction pipes and ballast suction pipes passing through deep tanks

Bilge suction pipes and ballast suction pipes passing through deep tanks are to be dealt with under the following requirements:

(1) Suction pipes, such as bilge suction pipes and ballast suction pipes, are not to pass through deep tanks carrying cargo oil, except that in cases where the pipes are installed in pipe tunnels provided within the deep tanks.

(2) For any bilge suction pipes passing through deep tanks serving as permanent ballast tanks, welded pipe joints may not be required if flange joints, corresponding to a nominal pressure one rank higher than the design pressure, are used.

(3) In cases where gravitational ballasting/deballasting is intended by using sea chests provided in permanent ballast tanks, double stop valves which are operable from a position on the freeboard deck are to be provided.

(4) In the application of the requirements specified in **(1)** to **(3)** above, bilge hoppers are to be regarded as deep tanks.

3 Valves on bilge suction piping

Valves on the bilge suction piping are to be operable from a position in machinery spaces or shaft tunnels, or to be capable of being remotely controlled from a readily accessible place.

4 The wording "where as deemed appropriate by the Society" specified in **13.5.1-11, Part D of the Rules** means the case where an oily-water separator with an appropriate processing capacity or no openings for the discharging ballast water to the ocean is provided and the oily ballast water is intended to discharge on the shore.

D13.5.3 Size of Bilge Suction Pipes

1 Pipe diameters of bilge suction pipes

For the bilge suction piping of hold bilges, main bilge suction systems (Christmas tree system) are, in principle, not to be adopted. In cases where such an arrangement is unavoidable, it is to be ensured that the ship satisfies one-sub-division flooding conditions. Pipe diameters of bilge suction pipes in such systems are to be calculated according to the Figure below. (*See* **Fig. D13.5.3-1**)

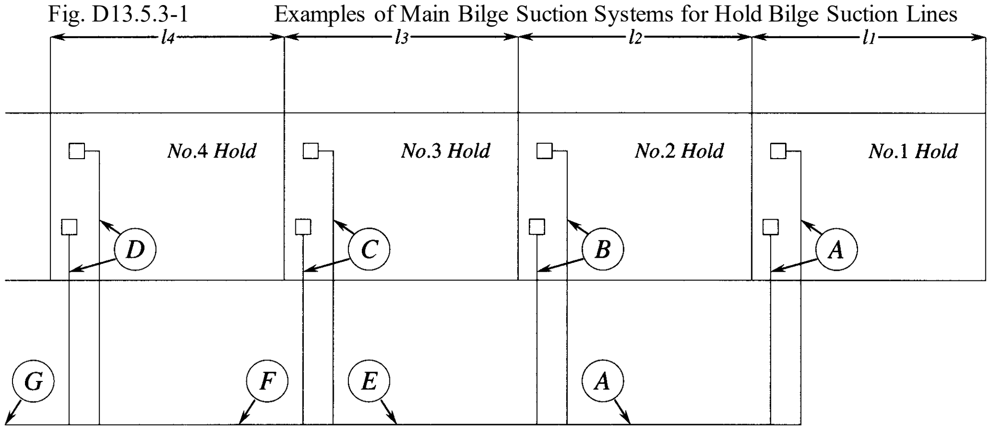
2 Pipe diameters of bilge suction pipes of ships with double hull construction

In ships with double hull construction, the inside diameter of bilge suction pipes may be determined by using the distance between the inner hull in place of the breadth of the ship.

3 Internal diameters of the bilge suction pipes for ships with unusually large freeboard

When calculating internal diameters of bilge suction pipes for ships with unusually large freeboards, "*D*" may be replaced with the vertical distance from the top of the keel to an assumed freeboard deck.

Fig. D13.5.3-1 Examples of Main Bilge Suction Systems for Hold Bilge Suction Lines



Notes:

(1) Pipes *A*, *B*, *C* and *D* are to be calculated as branch bilge suction pipes respectively by substituting l_1, l_2, l_3 and l_4 for l.

(2) Pipe *E* is to be calculated as a main bilge line by regarding the sum of $l_1 + l_2$ as L, and the sectional area of pipe *E* is to be greater or equal to the sum of the sectional areas of pipe *A* and pipe *B*.

(3) Pipe *F* is to be calculated as a main bilge line by regarding the sum of $l_1 + l_2 + l_3$, as *L*, and the sectional area of pipe *F* is to be sum of the internal sectional areas of the largest two branch bilge suction pipes among *A*, *B* and *C*.

(4) Pipe *G* is to be calculated as a main bilge line by regarding the sum of $l_1 + l_2 + l_3 + l_4$, as *L*, and the sectional area of pipe *G* is to be sum of the internal sectional areas of the largest two branch bilge suction pipes among *A*, *B*, *C* and *D*.

In this case, screw-down type non-return valves are to be provided at the suction of each branch piping. In cases where the installed position of such valves are not readily accessible, a remote control device is to be provided.

D13.5.4 Bilge Pumps

1 Capacity of bilge pumps

Even when the capacity of one of the independent power bilge pumps specified in **13.5.4, Part D of the Rules** falls slightly short of its required capacity, this pump may be accepted in cases where its capacity is 80% or more and the total capacity of this pump and the other pump is 200% or more of the capacity required for one pump.

2 Exclusive bilge piping system of cargo holds by eductors

The wording "to all be considered appropriate by the Society" specified in **13.5.4-4, Part D of the Rules** means as below:

(1) Inside diameter of bilge suction pipes

The inside diameter of bilge suction pipes is not to be less than the value obtained by the following formula:

$$d = 1.68\sqrt{l(B+D)} + 25$$

where

d : Inside diameter of bilge suction pipe (mm)

l : Length of cargo hold (m)

B : Breadth of cargo hold (m)

D : Depth of cargo hold (m)

(2) Suction capacity of eductors

The suction capacity of eductors is to be not less than the value obtained from the formula:

$$Q=5.66d^2\times 10^{-3}$$

where

Q : Bilge suction capacity of eductor (m^3/h)

d : Same as in **(1)** above

(3) Amount of driving water for eductors

Eductors are to be arranged so that they are capable of being driven by two or more units of pumps. In cases where bilges in two or more cargo holds are discharged by eductors driven by these pumps, the amount of driving water for each pump is to be sufficient enough to simultaneously draw bilge from at least two cargo holds at the suction capacity specified in **(2)** above.

(4) Eductor driving-water stop valves and bilge discharge valve

Eductor driving-water stop valves and bilge discharge valves are to be provided. These valves are to be operable from positions on the bulkhead deck or upward, except in cases where these valves are provided in engine rooms.

(5) Prevention of the back-flow of eductor driving water into bilge wells and bilge high-level alarm devices

(a) Non-return valves are to be provided near the suction ends of bilge suction pipes to prevent any back-flow of eductor driving water into bilge wells or bilge high-level alarm devices, which activates when back-flow occurs, are to be provided in each bilge well.

(b) Bilge high-level alarm devices are to activate audible and visible alarms in a normally manned position at which the cargo hold whose bilge level in its bilge well has become high can be identified.

(c) Circuits of bilge high-level alarm devices are to have self-monitoring functions or at least two independent circuits are to be arranged in the cargo hold.

(d) Bilge high-level alarm devices in cargo holds loaded with coal are to comply with the requirements in **4.9.1, Part H of the Rules**.

(6) Eductor driving-water pipes passing through tanks

Eductor driving-water pipes passing through tanks are to comply with the requirements for bilge pipes in **13.5.1, Part D of the Rules**. In this case, the non-return valves specified in **13.5.1-8, Part D of the Rules** may be commonly used for the non-return valves specified in **(5)(a)**.

(7) Rose boxes at bilge suction ends

Rose boxes provided at bilge suction ends are to be adequate ones matching the bilge suction capacity of the eductor, notwithstanding the requirements specified in **13.5.9-2, Part D of the Rules**.

(8) Protection of bilge piping systems in cargo holds

Eductor driving-water piping, bilge discharge piping and eductors are to be so arranged as not to be damaged by cargo.

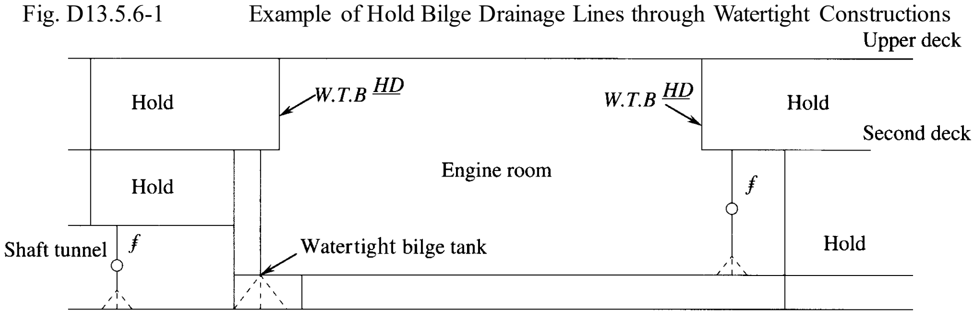
(9) Common use of eductor driving water with fire-fighting water

In cases where eductors driving water is taken from piping used for fire-fighting water, consideration is to be given so that no adverse effects are suffered by any of the fire-fighting functions.

D13.5.6 Bilge Drainage from the Top of Deep Tanks, Fore and After Peak Tanks and Chain Lockers

In cases where hold bilges are drained into engine rooms or shaft tunnels adjacent thereto through the watertight construction as specified in **Fig. D13.5.6-1**, the bilge drainage piping is to be led to spaces readily accessible and self-closing valves or cocks are to be provided. In cases where such bilge is led to watertight bilge tanks and the hold is located above the load line, the above-mentioned valves or cocks may be omitted, but non-return valves are to be provided. In cases where hold bilges are led to shaft tunnels, no sounding pipes may be provided; however, the diameter of any drainage pipes are not to be less than the value specified for bilge suction pipes.

Fig. D13.5.6-1 Example of Hold Bilge Drainage Lines through Watertight Constructions



f : Valve or cock with self-closing device

D13.5.8 Bilge Wells

In cases where bilge tanks are provided instead of bilge wells, drain pipes to be led directly to bilge tanks are to be provided with automatic non-return valves or a stop valves which can be closed at easily accessible places above the freeboard deck. However, in cases where the open ends of drain pipes are not located below the freeboard deck, these valves may be omitted.

D13.5.9 Mud Boxes and Strum Boxes

The wording "except in cases approved by the Society" in **13.5.9-2, Part D of the Rules** is to be according to those specified in **D13.5.4-2(7)**.

D13.5.10 Dewatering Arrangements for Bulk Carriers, etc.

1 With respect to the provisions of **13.5.10, Part D of the Rules**, the following components in bilge and ballast systems are to be capable of being brought into operation from readily accessible enclosed spaces.

(1) Eductors and pumps for dewatering the spaces which include driving water pumps for such eductors.

(2) All valves in piping systems served by the devices specified in **(1)**, except for those whose controls are appropriately kept in open/close position by locking devices so that dewatering arrangements are always operable at sea. In cases where dewatering piping from one space/tank is connected to that from another space/tank, each piping is to be provided with a valve nearby the suction header of the pump or suction wells for eductors, and all such valves are to be capable of being independently brought into operation. In addition, such valves for spaces other than ballast tanks are to be non-return valves. In cases where dewatering piping arrangements for dry spaces is connected to that for ballast tanks, with respect to the provisions of **13.2.2-1(4), Part D of the Rules**, two non-return valves are to be provided between those arrangements in a readily accessible position and one of them is to be fitted with shut-off isolation arrangement which is to be capable of being controlled from the same position as those required for other valves.

2 With respect to the provisions of **13.5.10, Part D of the Rules**, bilge and ballast systems for the dewatering arrangements (hereinafter, referred to as "the dewatering systems") are to comply with the following requirements:

(1) Capacity of the dewatering systems is not to be less than the value obtained by the following formula:

$$Q = 320A \ (m^3/hr)$$

A : Cross-sectional area of the largest air pipe or ventilator duct serving the space/tank (m^2)

(2) The operation of dewatering systems is to be designed such that machinery, electrical equipment, fire-fighting systems and other systems essential for the safety of the ship remain available and ready for immediate use. In this context, at least one fire pump is to satisfy the capacity of fire pumps required by the provisions of **10.2.2-4(2), Part R of the Rules** when the dewatering systems are in operation.

(3) Dewatering systems are to be such that any accumulated water can be drained directly by pumps or eductors. (A gravity dewatering system is not permitted.)

(4) Enclosures of electrical equipment for the dewatering systems installed in spaces where the systems are to be installed, are to provide protection to the IP68 standard_as defined in *IEC* 60529:1989/AMD2:2013/COR1:2019 for a water head equal to the height of the space in which the electrical equipment is installed for a time duration of at least 24 *hours*.

(5) Suction wells are to be provided with gratings or strainers that will prevent dewatering systems from being clogged with debris.

(6) In cases where dewatering piping is connected to bilge piping or other similar piping for cargo holds, appropriate means are to be taken to prevent any dangerous vapours/gases from cargo, carbon dioxide gases of fixed fire-extinguishing systems, etc. from entering into the spaces/tanks where the dewatering piping is served.

3 With respect to the provisions of **13.5.10, Part D of the Rules**, remote operation of any of the valves for the dewatering systems is to be in accordance with the following:

(1) Position indication is to be provided at remote control stations to show whether the valve is fully open or closed;

(2) Valves are not to move from the demanded position in the case of failure of the control system power or actuator power; and

(3) Positions which are accessible via under deck passages, pipe trunks or other similar means of access are not taken as being in readily accessible enclosed spaces.

D13.6 Air Pipes

D13.6.1 General

1 The word "cofferdams" in **13.6.1-1, Part D of the Rules** means spaces between two bulkheads or decks primarily designed as a safeguard against leakage of oil from one compartment to another.

2 The Society may accept the omission of the air pipes required by **13.6.1-1, Part D of the Rules** for small spaces which have no permanent means of suction or discharge of bilge by pump (i.e. the small spaces specified in **D13.5.1-1(2)**).

D13.6.2 Open Ends of Air Pipes

The wording "automatic closing devices" specified in **13.6.2-2**, **Part D of the Rules** means those approved by the Society in accordance with **2.4.2-10**, **Part 6 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**. For tankers, pressure-vacuum valves (PV valves) may be used in lieu of automatic closing devices. These valves are to be of a type approved by the Society in accordance with procedures deemed appropriate by the Society.

D13.6.3 Size of Air Pipes

As a means of negative pressure protection for tanks specified in **13.6.3(2), Part D of the Rules**, air passage holes of 10 *mm* in diameter or thereabout may be drilled on the cover of closing appliances fitted at the open end.

D13.6.4 Height of Air Pipes

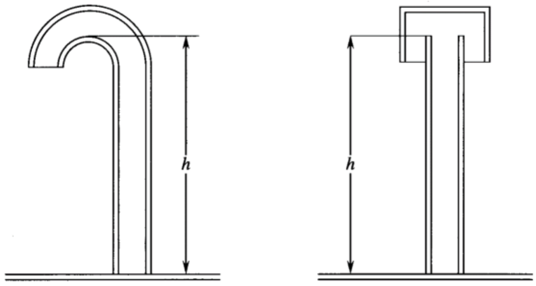
1 The height of air pipes above deck is to be measured as shown in **Fig. D13.6.4-1**.

2 In the application of the requirements of **13.6.4, Part D of the Rules**, the term "superstructure deck" includes top decks of superstructures, deckhouses, companionways and other similar deck structures.

3 The term "freeboard deck" specified in **13.6.4, Part D of the Rules** includes superstructure decks lower than h_S specified in **V2.2.1** above the freeboard deck.

4 The term "superstructure deck" specified in **13.6.4, Part D of the Rules** means superstructure decks located at least h_S specified in **V2.2.1** above the freeboard deck and lower than $2h_S$ specified in **V2.2.1** above the freeboard deck.

Fig. D13.6.4-1 The Height of Air Pipe



D13.6.5 Additional Requirements for Air Pipes Fitted on Exposed Fore Decks

The strength of air pipes and their closing devices in **13.6.5**, **Part D of the Rules** are to comply with the following requirements:

(1) Applied Loading

Forces acting in the horizontal direction on pipes and their closing devices are to be calculated by using the pressure (p) obtained from the following formula and the largest projected area of each component.

$p = 0.5\rho V_w^2 C_d C_s C_p$ (kN/m^2)

ρ : Density of sea water (1.025 t/m^3)

V_w : Velocity of water over the fore deck given by the following:

$13.5(m/sec)$: for $h_{ed} \leq 0.5h_t$

$13.5\sqrt{2\left(1-\frac{h_{ed}}{h_t}\right)}(m/sec)$: for $0.5h_t < h_{ed} < h_t$

h_{ed}: Distance from the designed maximum load line to exposed deck (m)

h_t : $0.1L_1$ or 22 m whichever is the lesser

C_d : Shape coefficient (0.5 for pipes and 1.3 for air pipe heads in general, 0.8 for an air pipe heads of cylindrical form with its axis in the vertical direction)

C_s : Slamming coefficient (3.2)

C_p : Protection coefficient given by the following:

(0.7): for pipes and heads located immediately behind a breakwater or forecastle,

(1.0): elsewhere and immediately behind a bulwark.

(2) Strength Requirements

(a) Bending moments and stress in air and ventilator pipes are to be calculated at critical positions such as at penetration pieces, at welds or flange connections, at toes of supporting brackets, etc. Bending stresses in net sections are not to exceed 0.8 times σ_y , where σ_y is the specified minimum yield stress or 0.2% proof stress of the steel at room temperature. Irrespective of corrosion protection, a corrosion addition to the net section of 2.0 mm is then to be applied.

(b) For standard air pipes with a height of 760 mm closed by heads of not more than the values for the projected area, pipe thickness and bracket heights specified in **Table D13.6.5-1.** In cases where brackets are required, three or more radial brackets of gross thickness 8 mm or more, of minimum length 100 mm, and height according to **Table D13.6.5-1** are to be fitted, but these brackets need not extend over the joint flange for the head. Bracket toes at the deck are to be suitably supported.

(c) For configurations different from the standards specified in **Table D13.6.5-1**, loads according to **(1)** are to be applied, and means of support are to be determined in order to comply with the requirements of **(a)**. Brackets, in cases where fitted, are to be of suitable thickness and length according to their height. Pipe thickness is not to be taken less than as the value indicated in column 1 of **Table D12.6 Part D of the Rules**.

(d) All component parts and connections of air pipes are to be capable of withstanding the loads defined in **(1)**.

Table D13.6.5-1 760 *mm* Air Pipe Thickness and Bracket Standards

Nominal pipe diameter (*mm*)	Minimum fitted gross thickness (*mm*)	Maximum projected area of head (cm^2)	Height of brackets (*mm*)
40*A*		-	520
50*A*	6.0	-	520
65*A*		-	480
80*A*	6.7	-	460
100*A*	7.6	-	380
125*A*	7.8	-	300
150*A*	8.5	-	300
175*A*		-	300
200*A*		1900	
250*A*		2500	
300*A*	8.5(2)	3200	300(2)
350*A*		3800	
400*A*		4500	

Notes:

(1) Brackets need not extend over the joint flange for the head.

(2) Brackets are required in cases where the as fitted (gross) thickness is less than 10.5 *mm*, or in cases where the projected head area given in the table is exceeded.

D13.8 Sounding Devices

D13.8.1 General

1 The word "cofferdams" in **13.8.1-1, Part D of the Rules** means the spaces specified in **D13.6.1-1**.

2 With respect to the sounding pipes and liquid level indicators required by **13.8.1-1, Part D of the Rules**, the Society may accept the other measures described in the following **(1)** and **(2)**.

(1) For small spaces (i.e. the spaces specified in **D13.5.1-1(2)**), the omission of sounding pipes and liquid level indicators may be allowed.

(2) For small spaces which are not covered by **D13.5.1-1(2)** that comply with the following **(a)** and **(b)**, the omission of sounding pipes and liquid level indicators may be allowed.

(a) The spaces are readily accessible.

(b) Other means for checking for presence of liquid inside the space are provided.

D13.8.3 Construction of Sounding Pipes

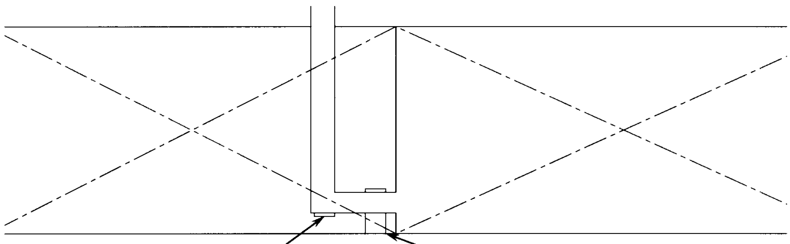
1 Elbow type sounding pipes

In cases where the use of elbow type sounding pipes is unavoidable, sufficient support is to be provided for the arm of the pipe. However, no elbow type sounding pipes are to be used for deep tanks. In addition, bottom striking pads are to be of a sufficient thickness. (*See* **Fig. D13.8.3-1**)

2 Striking plates

The standard value of striking plate thickness is approximately 10 *mm* for small ships and 12 *mm* for large ships.

Fig. D13.8.3-1 Example of Elbow Type Sounding Pipe Arrangements



D13.8.4 Construction of Liquid Level Indicators

The wording "a type that has been approved by the Society" in **13.8.4, Part D of the Rules** means those liquid level indicators approved in accordance with the requirements of **Chapter 4, Part 7 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use** and the wording "other standards approved by the Society" means *JIS F 7211* "5 K level gauges with valves", *JIS F 7215* "Flat glass oil level gauges" or any equivalent standards.

D13.8.5 Water Level Detection and Alarm Systems for Bulk Carriers, etc.

1 With respect to **13.8.5-1, Part D of the Rules**, water level detection and alarm systems (hereinafter, referred to as "the systems" in this paragraph) are to be installed on board in accordance with the following:

(1) Detectors, electrical cables and any associated equipment installed in cargo holds are to be protected from any damage caused by either cargo or cargo handling equipment.

(2) The systems are to be installed in locations where they are accessible for survey, maintenance and repair. Any filtration arrangements, if fitted to the detectors, are to be capable of being cleaned before loading.

(3) The installation of the systems is not to inhibit the use of any other sounding devices such as sounding pipes or other water level gauging devices.

2 Water levels specified in **13.8.5-1(1), Part D of the Rules** are to be measured from the top plating and to be detected at as close to the centre line as practicable, or at both the port and starboard sides of the cargo hold. For cargo holds fitted with insulation or close ceilings, water levels may be measured from the upper surface of the insulation or close ceilings in cases where watertightness is verified by tests. For this purpose, the position "at as close to the centre line as practicable" is to be of area within a distance from the centre line of less than or equal to 1 spacing of vertical stiffeners on the watertight bulkhead (or 1 corrugation space as shown in **Fig. D13.8.5-1**). In addition, the water levels specified in **13.8.5-1(2)** and **(3)** are to be detected at the lowest position possible of the relevant compartments.

3 The wording "the systems to have constructions and functions deemed appropriate by the Society" in **13.8.5-1(4), Part D of the Rules** means those systems complying with the following requirements and being of a type approved by the Society in accordance with **Chapter 5, Part 7 of Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use** or those systems approved by an organisation deemed appropriate by the Society in accordance with the Resolution *MSC.*188(79), as amended.

(1) The systems are to have sufficient corrosion resistance with consideration being given to the locations where the systems are to be installed and are to be maintain their functionality under expected service temperatures. In addition, any parts of the systems which may be exposed to cargo or bilge containing cargo, such as detectors, etc., are to be sufficiently able to cope with different conditions such as acidity, alkalinity, dust, etc. with consideration being given to the intended cargoes.

(2) Protection of the enclosures of electrical components for the systems is to satisfy the following **(a)** to **(c)**:

(a) The requirements of IP68 for those installed in spaces, tanks or cargo holds. This includes all adjacent spaces considered to be simultaneously flooded under damage stability calculations of the spaces/tanks/cargo holds required by the provisions of **2.3, Part 1, Part C of the Rules** or the requirements for ships to be assigned reduced freeboard in accordance with **Part V of the Rules**;

(b) The requirements of IP56 for those installed on exposed decks above the spaces/tanks/cargo holds; and

(c) The provisions of **Part H of the Rules** for any of those not specified in **(a)** or **(b)** above.

(3) Electrical installations for the systems installed in the following areas are to be of an intrinsically safe type or safe type of an appropriate apparatus group and temperature class suitable for the explosive gas atmosphere and/or combustible dust that can

be present, depending on the cargo carried, and hazardous area comparable with Zone 1 as defined in *IEC 60092-506*, Clause 3.1. Where the characteristics of the cargo carried are unknown, temperature class *T6*, gas group *IIC* and/or either dust group *IIIC* or IP5X are to be used, except electrical installations installed in ships designed only to carry cargo which does not cause combustible or explosive atmosphere. In addition, in cases where a ship is designed to carry only limited kinds of cargo, the maximum surface temperature may be appropriately relaxed depending on the kind of cargo. In this case, such limitations relating to cargo are to be documented in booklets for cargo operations. Finally, those electric installations installed at the edges of the following areas are to be approved at the discretion of the Society with due consideration being given to their design with respect to gas-tightness, etc.

(a) Cargo holds

(b) Enclosed spaces adjacent to cargo holds having openings without a gas-tight or watertight door/hatch and the like into a hold

(c) Areas within 3 *m* of any cargo hold mechanical exhaust ventilation outlet

(4) For electrical installations for the systems which are installed in ships intended for carrying dangerous goods, the provisions of **Chapter 19, Part R of the Rules** are to be referred to.

(5) Detectors are to be capable of indicating water level within an accuracy of ±100 *mm*. Time delays are to be so incorporated into alarm systems, in order to prevent spurious alarms due to any sloshing effects associated with ship motion, so that alarms will activate after detecting water level continuously for a standard period of not less than 10 *seconds*. The accuracy of these detectors may be set on the basis of seawater density.

(6) The systems are to be of a continuously self-monitoring type that also monitors any detectors. Audible and visual alarms are to be activated when any faults are detected. In this requirement, the term “fault” refers to problems such as open circuits, short circuits, loss of power supplies and CPU failures. The audible alarms are to be capable of being muted. However, visual alarms are to remain active until the malfunction has been cleared and such alarms are not to be capable of being turned off by hand. In addition, the systems are to be provided with means for testing their respective detectors when holds are empty.

(7) Alarm panels for the systems are to be provided with a switch for the testing of all audible and visual alarms. This switch is to return to the off position automatically when not being operated.

(8) The systems are to be supplied with electrical power from two independent sources. Any failure of two electrical power supplies is to be indicated by an alarm on the navigation bridge. In cases where secondary electrical power is supplied by dedicated batteries, such batteries are to be in accordance with the following **(a)** to **(c)**:

(a) Batteries are to have a capacity for a period of at least 18 *hours* and they are to be continuously charged;

(b) Batteries are to be arranged and located in accordance with **3.3.5, Part H of the Rules,** and may be integrated into the system; and

(c) Any failures of the battery systems, including battery charging systems specified in above **(a)**, are to be indicated by an alarm on the navigation bridge.

4 With respect to the provisions of **13.8.5-2, Part D of the Rules**, those audible alarms specified in **13.8.5-1(1)(b)**, **(2)** and **(3)**, **Part D of the Rules** need not be capable of being distinguished from. Visual alarms are to remain visible until the condition activating the alarm has returned below the level of the relevant detector and not to be capable of being turned off by hand.

5 With respect to the provisions of **13.8.5-2, Part D of the Rules**, one sensor capable of detecting both of the preset water levels specified in **13.8.5-1(1)(a)** and **(b)**, **Part D of the Rules** may be allowed.

6 The wording “override devices that are deemed appropriate by the Society” in **13.8.5-3, Part D of the Rules** means those complying with the following requirements:

(1) Alarms for tanks/cargo holds are to be capable of being independently turned off;

(2) Visual override indications are to be given to the navigation bridge throughout any deactivation of water level detectors for tanks/cargo holds;

(3) Such override devices are to be arranged so that alarm systems are automatically reactivated upon completion of any de-ballasting; and

(4) In cases where the override functions for water level detection and alarm systems are required to be specifically customised for each ship, override functions for spaces other than ballast tanks or cargo holds carrying ballast water are to be modified so that they cannot be activated when they are being installed on a ship. The above modification and any subsequent modifications are

to be confirmed by the Surveyor. A warning plate which prohibits personnel from overriding such alarms is not an acceptable alternative to the above modification.

7 Manuals specified in **13.8.5-4, Part D of the Rules** are to contain the following information and operational instructions:

(1) Descriptions of the equipment in the system together with listings of procedures for checking that, as far as practicable, each item of equipment is working properly during any stage of ship operation.

(2) Evidence that the system has been approved in accordance with **Chapter 5, Part 7 of Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use** or the Resolution *MSC*.188(79), as amended.

(3) Line diagrams of the system showing equipment positions

(4) Instructions for operator training, setting, securing, protecting and testing.

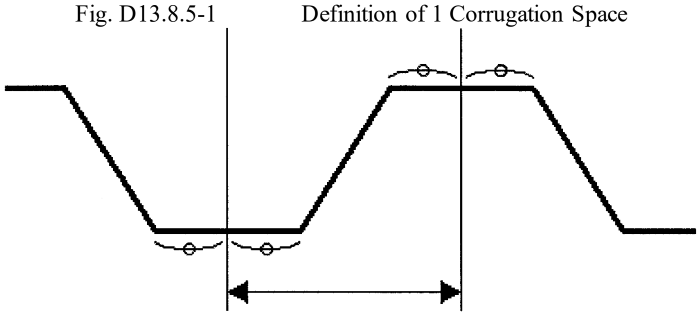
(5) Information regarding the types of cargo that guarantees performance. (In cases where electrical installations are required to be of an intrinsically safe, certificates verifying this are to be included.)

(6) Temperature range for which the equipment is suitable.

(7) Procedures to be followed in the event equipment in the system is not functioning properly.

(8) Maintenance requirements for the system.

Fig. D13.8.5-1 Definition of 1 Corrugation Space



D13.8.6 Water Level Detection and Alarm Systems for Single Hold Cargo Ships

1 Water level detection and alarm systems required in **13.8.6-1, Part D of the Rules** are in accordance with the provisions of **D13.8.5**.

2 For the purpose of the provisions of **13.8.6-4, Part D of the Rules**, "having a breadth that is deemed appropriate by the Society" means that the distance between the side shell and the inner shell in any part of a watertight compartment is not less than 760 *mm*. This distance is to be measured perpendicular to the side shell.

D13.8.7 Water Level Detection and Alarm Systems for Multiple Hold Cargo Ships

1 The water level detection and alarm systems required by **13.8.7-1, Part D of the Rules** are to be in accordance with **D13.8.5**.

2 The wording "override devices that are deemed appropriate by the Society" in **13.8.7-3, Part D of the Rules** means those complying with **D13.8.5-6**.

3 The bilge alarms systems used as water level detection and alarm systems in accordance with **13.8.7-4, Part D of the Rules** are to comply with **D13.8.5**.

4 For the bilge wells which are applicable to **19.3.5-1, Part R of the Rules**, the following requirements **(1)** and **(2)** are to be complied with.

(1) Where the cargo hold bilge well is sealed, suitable alternative detectors are to be provided.

(2) Where the cargo hold bilge well is used, the bilge well is not to be sealed so that the bilge alarm system can detect the water level.

5 In applying **13.8.7-5, Part D of the Rules**, manuals documenting operating and maintenance procedures for bilge alarm systems used as water level detection and alarm systems are to contain the following information and operational instructions in addition to that required by **D13.8.5-7**:

(1) Manuals for switching to the alternative arrangements (if fitted), and

(2) List of cargoes for which alternative provisions are to be used

D13.9 Fuel Oil Systems

D13.9.1 General

1 Common use of fuel oil tanks and ballast tanks

For those tanks being used as both fuel oil tanks and ballast tanks, piping arrangements are to be made in such a way that either fuel oil or ballast water can be independently drawn out under any circumstances. (*See* **Fig. D13.9.1-1**)

2 Passage of pipes through tanks

No fuel oil pipes are to be run through potable fresh water tanks. No pipes used for potable fresh water tanks are to be run through fuel oil tanks.

3 The "fuel oil service tanks" and "equivalent arrangements" for commonly utilized fuel oil piping systems specified in **13.9.1-6**, **Part D of the Rules** are to be in accordance with the following:

(1) The wording "fuel oil service tanks" refers to those fuel oil tanks which contain only fuel of a quality ready for use and that meet any specifications required by the equipment manufacturer. In such cases, service tanks are to be declared as such and are not to be used for any other purpose.

(2) Use of setting tanks with or without purifiers, or purifiers alone, and one service tank is not acceptable as an "equivalent arrangement" for two service tanks.

(3) Examples of commonly utilized arrangements and examples deemed "equivalent arrangements" complying with **13.9.1-6** and **-7**, **Part D of the Rules** are shown in but not limited to **Fig. D13.9.1-2** and **Fig. D13.9.1-3**. In cases where providing such "equivalent arrangements", however, propulsion and vital systems which use two types of fuel are to support rapid fuel changeover and are to be capable of operating under all normal operating conditions at sea with both types of fuel (heavy fuel oil (HFO) and marine diesel oil (MDO)).

Fig. D13.9.1-1 Example of Piping Arrangements for Fuel Oil Tanks and Ballast Tanks in Common Service

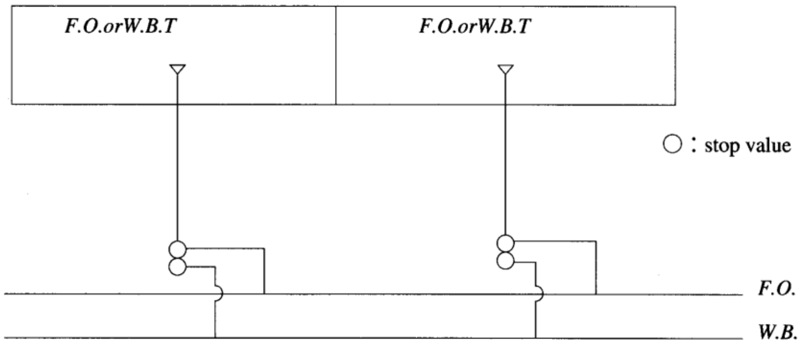


Fig. D13.9.1-2 Examples of the Arrangements for Main and Auxiliary Engines and Auxiliary Boilers Operating with HFO

(a) Examples of commonly utilized arrangements

HFO service tank (The capacity for at least 8 hours at maximum continuous rating of the main engine, normal operating load of the generators at sea and the auxiliary boiler.)	HFO service tank (The capacity for at least 8 hours at maximum continuous rating of the main engine, normal operating load of the generators at sea and the auxiliary boiler.)
MDO tank (For initial cold starting or repair work of engines/boilers)	

(b)) Examples of equivalent arrangements(1), (2)

HFO service tank (The capacity for at least 8 hours at maximum continuous rating of the main engine, normal operating load of the generators at sea and the auxiliary boiler.)	MDO service tank (The capacity for at least 8 hours at maximum continuous rating of the main engine, normal operating load of the generators at sea and the auxiliary boiler.)

Notes:

(1) This arrangement only applies where main and auxiliary engines can operate with heavy fuel oil under all load conditions and, in the case of main engines, during manoeuvring.

(2) For pilot burners of auxiliary boilers (if provided), an additional MDO tank for 8 hours of operation may be necessary.

(3) Fuel oils which require post service tank heating to achieve their required injection viscosities are not to be regarded as MDO in this context.

Fig. D13.9.1-3 Example of the Arrangements for Main Engines and Auxiliary Boilers Operating with HFO, and Auxiliary Engines Operating with MDO

(a) Examples of commonly utilized arrangements

HFO service tank (The capacity for at least 8 hours at maximum continuous rating of the main engine and the auxiliary boiler.)	HFO service tank (The capacity for at least 8 hours at maximum continuous rating of the main engine and the auxiliary boiler.)
MDO service tank (The capacity for at least 8 hours at normal operating load of the generators at sea.)	MDO service tank (The capacity for at least 8 hours at normal operating load of the generators at sea.)

(b) Examples of equivalent arrangements(1)

HFO service tank (The capacity for at least 8 hours at maximum continuous rating of the main engine and the auxiliary boiler.)	
MDO service tank (The capacity for at least the greatest of following i) or ii): i) at least 4 hours at maximum continuous rating of the main engine, normal operating load of the generators at sea and the auxiliary boiler; or ii) at least 8 hours at normal operating load of the generators at sea and the auxiliary boiler.)	MDO service tank (The capacity for at least the greatest of following i) or ii): i) at least 4 hours at maximum continuous rating of the main engine, normal operating load of the generators at sea and the auxiliary boiler; or ii) at least 8 hours at normal operating load of the generators at sea and the auxiliary boiler.)

Notes:

(1) Fuel oils which require post service tank heating to achieve their required injection viscosities are not to be regarded as MDO in this context.

D13.9.4 Drip Trays and Drainage Systems

Fuel oil drainage is to be dealt with by the following requirements:

(1) For those fuel oil piping systems that employ sediment tanks, drain valves are to be either automatic shut off valves or locked valves (or cocks).

(2) In cases where the possibility exists that the design pressure of a fuel oil heater will be exceeded, relief valves are to be fitted and either drainage is to be led into a drainage tank or other means are to be in place to prevent any spraying of fuel oil.

D13.9.5 Fuel Oil Heaters

In cases where electric heaters are provided in double bottom tanks or deep tanks, an oil temperature distribution chart is to be submitted to the Society.

D13.9.6 Fuel Oil Systems for Reciprocating Internal Combustion Engines

1 In cases where ships intended to use heavy fuel oil or marine diesel oil for operating reciprocating internal combustion engines use low sulphur fuel oil instead, any of the following is to be complied with. For reference, "low sulphur fuel oil" in this paragraph refers to marine fuel with a sulphur content not exceeding 0.1 *% m/m* and a minimum viscosity of 2 *cSt*.

(1) Each of the fuel supply pumps required by **13.9.6-1(1), Part D of the Rules** is to be suitable for low sulphur fuel oil operations at the capacity required for normal propulsion machinery operation.

(2) When the fuel oil supply pumps required by **13.9.6-1(1), Part D of the Rules** are suitable to operate on low sulphur fuel oil but one pump alone is not capable of delivering the low sulphur fuel oil at the required capacity, then both pumps may operate in parallel to achieve the capacity required for normal propulsion machinery operation. In such cases, one additional fuel oil pump is to be provided. The additional pump is, when operating in parallel with one of the pumps required by **13.9.6-1(1), Part D of the Rules**, to be suitable for and capable of delivering low sulphur fuel oil at the capacity required for normal propulsion machinery operation.

(3) In addition to the fuel oil supply pumps required by 13.9.6-1(1), Part D of the Rules, two separate fuel oil pumps are to be provided, each capable of and suitable for supplying low sulphur fuel oil at the capacity required for normal propulsion machinery operation.

2 One self-cleaning filter will also be accepted as a filter capable of being cleaned without stopping the supply of filtered oil required by 13.9.6-4(2), Part D of the Rules.

D13.9.7 Burning Systems for Boilers

1 In cases where ships intended to use heavy fuel oil or marine diesel oil for operating boilers use low sulphur fuel oil instead, any of the following is to be complied with. For reference, "low sulphur fuel oil" in this paragraph refers to marine fuels with a sulphur content not exceeding 0.1 *% m/m* and a minimum viscosity of 2 *cSt*.

(1) Each of the burning pumps required by 13.9.7-1(1) or -2(1), Part D of the Rules is to be suitable for low sulphur fuel oil operations at the capacity required for normal navigation.

(2) When the burning pumps required by 13.9.7-1(1) or -2(1), Part D of the Rules are suitable to operate on low sulphur fuel oil but one pump alone is not capable of delivering the low sulphur fuel oil at the required capacity, then both pumps may operate in parallel to achieve the capacity required for normal navigation. In such cases, one additional fuel oil pump is to be provided. The additional pump is, when operating in parallel with one of the pumps required by 13.9.7-1(1) or -2(1), Part D of the Rules, to be suitable for and capable of delivering low sulphur fuel oil at the capacity required for normal navigation.

(3) In addition to the burning pumps required by 13.9.7-1(1) or -2(1), Part D of the Rules, two separate fuel oil pumps are to be provided, each capable of and suitable for supplying low sulphur fuel oil at the capacity required for normal navigation.

2 For auxiliary boilers used exclusively for the fuel oil heating necessary for the operation of main propulsion machinery or any cargo heating that is continuously required, only one burning system may be accepted in cases where one complete spare unit of a burning pump, capable of being used as a replacement within a short period of time, is equipped, notwithstanding the requirements in 13.9.7-2(1), Part D of the Rules.

D13.10 Lubricating Oil Systems and Hydraulic Oil Systems

D13.10.1 General

Electric heaters provided in double bottom tanks and deep tanks are to comply with the requirements specified in D13.9.5.

D13.10.4 Lubricating Oil Filters

One self-cleaning filter will also be accepted as a filter capable of being cleaned without stopping the supply of filtered oil required by 13.10.4-2, Part D of the Rules.

D13.11 Thermal Oil Systems

D13.11.3 Pumps for Thermal Oil Heaters

1 The wording "Thermal oil heaters of important use" specified in 13.11.3, Part D of the Rules refers to those in which thermal oil is used for any of the following:

(1) Fuel oil heating necessary for main propulsion machinery operation

(2) Any cargo oil heating that is continuously required

2 Notwithstanding the requirements in 13.11.3, Part D of the Rules, thermal oil heaters of important use may be provided with only one fuel injection pump in cases where one complete spare unit of a pump, capable of being replaced within a short period of time, is equipped on board.

D13.11.4 Heating of Liquid Cargo with Flash Points below 60℃

The wording "in those cases deemed appropriate by the Society" in 13.11.4, Part D of the Rules means the thermal oil systems are to satisfy all of the following conditions:

(1) The systems are to be arranged so that the internal pressure of the heating coils is at least a water head 3 *m* higher than the static head of any cargo in cases where circulating pumps are not being operated.

(2) Thermal oil system expansion tanks are to be provided with high and low level alarms.

(3) Means are to be provided in thermal oil system expansion tanks for detecting flammable cargo vapours.

(4) Valves for individual heating coils are to be provided with locking arrangements to ensure that the coils are exposed to static pressure from the thermal oil at all times.

D13.12 Cooling Systems

D13.12.1 Cooling Pumps

The capacity of stand-by circulating pumps of ships in which steam turbines are used as main propulsion machinery specified in **13.12.1-1(1), Part D of the Rules** is to be of sufficient to assure that the ship has enough engine output to attain navigable speed.

D13.12.3 Cooling Systems for Reciprocating Internal Combustion Engines

In cases where oil tanks are heated by cooling fresh water from main propulsion machinery, adequate means are to be provided to detect any oil contamination in the cooling fresh water piping system.

D13.13 Pneumatic Piping Systems

D13.13.6 Pneumatic Piping Systems for Essential Services

The wording “essential services” in **13.13.6(1), Part D of the Rules** means those services essential for propulsion and steering and safety of the ship as specified in **3.2.1-2, Part H of the Rules**.

D13.14 Steam Piping Systems and Condensate Systems

D13.14.4 Condensate Systems

“Suitable measures deemed appropriate by the Society” in **13.14.4-2, Part D of the Rules** means, for example, such a system that steam flowing into the condenser is shut off in cases where the internal pressure of the condenser exceeds design pressure.

D14 PIPING SYSTEMS FOR TANKERS

D14.1 General

D14.1.1 Scope

Tankers with double bottoms

In cases where tankers with double bottoms use the spaces underneath their cargo oil tanks for purposes other than holding cargo oil, the requirements specified in **Chapter 14, Part D of the Rules** as well as the following requirements specified in this **D14.1.1** are to be complied with:

(1) Air pipes and sounding pipes provided in double bottoms may pass through cargo oil tanks. In this case, all pipe joints in such cargo oil tanks are to be welded joints of sufficient thickness according to the requirements of **12.6, Part D of the Rules**. Furthermore, consideration is to be given to piping arrangements for the expansion and contraction of the pipes.

(2) Valve operating rods are not to pass through any part subjected at all times to liquid head, such as the inner bottom plates of cargo tanks.

(3) Notwithstanding the requirements of **14.2.7-4, Part D of the Rules**, ballast pipes are not to pass through any spaces within cargo oil tanks.

D14.2 Cargo Oil Pumps, Cargo Oil Piping Systems, Piping in Cargo Oil Tanks, etc.

D14.2.2 Arrangement of Cargo Oil Piping Systems

"Cargo piping systems" in **14.2.2-7, Part D of the Rules** includes cargo oil pipes, vent pipes, tank washing pipes, etc.

D14.2.3 Alternative Use of Tanks

1 Cargo oil tanks also used as segregated ballast tanks

In cases where tanks alternate between being used as cargo oil tanks and segregated ballast tanks, the related cargo oil pipes, ballast pipes and vent pipes are to be arranged, as shown in **Fig. D14.2.3-1**, so that they can be switched for each respective case.

Furthermore, for other piping systems, the requirements for the piping systems in cargo oil tanks are to be complied with.

2 Cargo oil tanks also used as fuel oil tanks

In cases where tanks alternate between being used as cargo oil tanks and fuel oil tanks, all related cargo oil pipes, fuel oil pipes and vent pipes are to be arranged, as shown in **Fig. D14.2.3-2**, so that they can be switched for each respective case.

Furthermore, for other piping systems, the requirements for the piping systems in cargo oil tanks are to be complied with.

Fig. D14.2.3-1Example of Piping Arrangements for Tanks Alternatively Used as Cargo Oil Tanks and Segregated Ballast Tanks

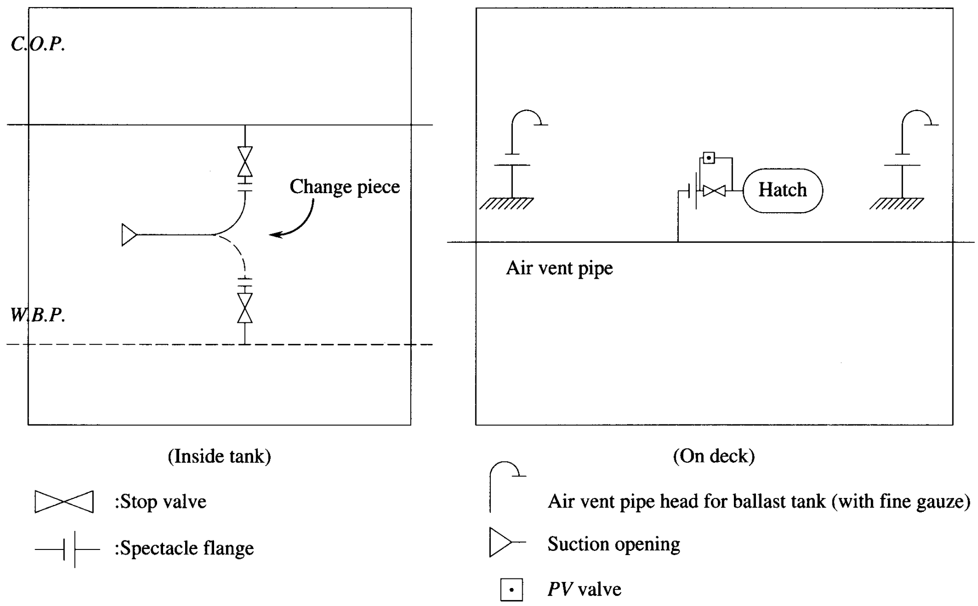


Fig. D14.2.3-2Example of Piping Arrangements for Tanks Alternatively Used as Cargo Oil Tanks and Fuel Oil Tanks

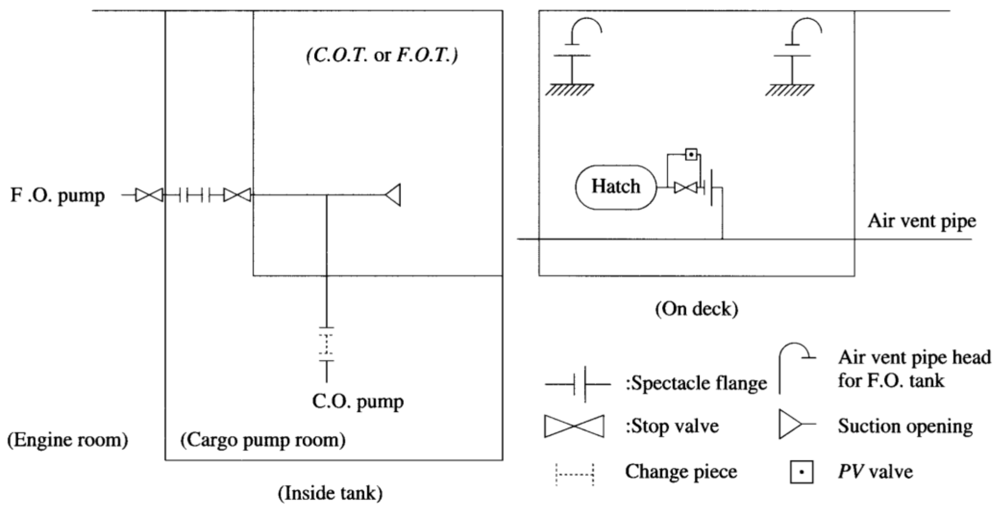
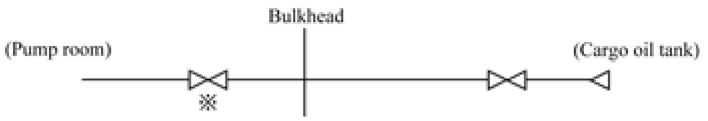
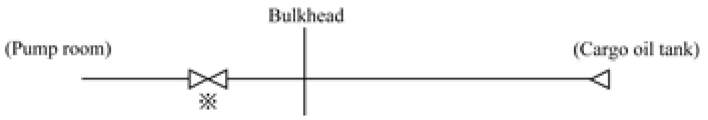
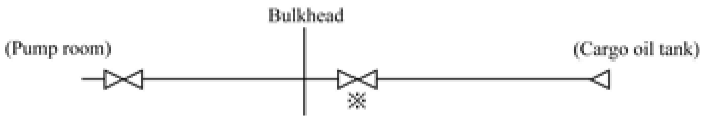
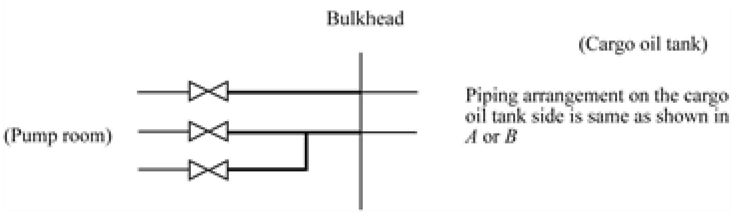


Table D14.2.5-1 Arrangement and Type of Bulkhead Valves

Arrangement *A*		In cases where stop valve is provided at the end of opening in the cargo oil tank: No specific requirements are imposed on type and material of bulkhead valve.
Arrangement *B*		In cases where no stop valve is provided at the end of opening in the cargo oil tank: The bulkhead valve is to be of steel castings and be operable from a control position on deck.
Arrangement *C*		In cases where bulkhead valve is provided in the cargo oil tank: The valve on the tank side is to be operable from a control position on deck.
Arrangement *D*		In cases where bulkhead valve is not provided close to the bulkhead: If those indicated with bold lines in the figure are of cast steel pipes having thickness of 13.5*mm* or more or of heavy gauge steel pipes of 16*mm* or more, the requirements for bulkhead valves as shown in Arrangement *A* or *B* may be accepted.

Note:

※ This valve is to be installed as close to the bulkhead as practicable

D14.2.4 Separation of Cargo Oil Pumps and Cargo Oil Pipes

1 Piping systems to be connected to cargo oil piping are to be dealt with under the following requirements:

(1) Pumps and pipes in any piping systems connected to cargo oil pipes are to be dealt with in the same manner as those in cargo oil piping systems. However, for those piping systems specified in **14.2.2-4**, **14.2.9-6**, **14.3.1-2**, and **14.3.2-2, Part D of the Rules** and item **(2)** below, this requirement may be dispensed with. Piping systems connected to cargo oil piping means those connected to cargo oil pipes, and those piping systems having openings thereto. Accordingly, hydraulic oil pipes for controlling cargo oil piping systems, for example, are not regarded as a piping system connected to the cargo oil piping.

(2) In cases where cargo oil piping systems are connected to the following piping systems:

(a) Tank vent pipes

The requirements in **35.2.2-3(2)(g)** and **(h), Part R of the Rules** are to be complied with. In addition, ventilating fans, except for inert gas blowers, are to be installed within hazardous area (as for the definition of "hazardous area," *see* **4.2.3-1, Part H of the Rules**).

(b) Pressure gauge pipes for cargo oil piping systems (including pumps)

Pressure gauges to which cargo oil is directly led are to be installed in pump rooms or on weather decks. However, in cases where stop valves are provided at joints between pressure gauge piping systems and cargo oil piping systems, and in cases where bulkhead valves are provided at locations where such pipes penetrate bulkhead between engine rooms and pump rooms, pressure gauges may be installed in engine rooms.

(c) Pipes for measuring oil content

Sampling pipes for measuring oil content may be led to spaces other than hazardous area, in cases where such pipes have nominal diameters of 25 *A* or less and in cases where two or more stop valves are provided between cargo oil piping and the penetration of the casing of non-hazardous area.

2 The wording "flanged joints which have no risk of leakage" in **14.2.4-5** and **-6, Part D of the Rules** means welded flange joints rated at least a nominal pressure of 1.0 *MPa* or a nominal pressure one rank higher than required design pressure, whichever is greater.

3 The wording "expansion bends" in **14.2.4-6, Part D of the Rules** means expansion loops such as omega bends in piping systems to counteract excessive stress or displacement caused by thermal expansion or hull deformation which could be fabricated from straight lengths of pipe.

D14.2.5 Bulkhead Valves of Cargo Oil Piping Systems

1 The arrangement and type of bulkhead valves are to be as given in **Table D14.2.5-1**.

2 Those piping systems, according to **D14.2.4(1)**, that are not being used for transferring cargo fuel oil, in cases where the requirements for cargo fuel oil piping are being applied, may be fitted with remote control devices, specified in **14.2.5-2, Part D of the Rules**, that are used only for closing. (*e.g.*, slop tanks or oil concentration detecting cocks fitted to the walls of pump rooms)

D14.2.7 Piping in Cargo Oil Tanks

1 Ballast pipes

In cases where ballast pipes passing through cargo oil tanks are led to ballast tanks located afore of collision bulkheads, the requirements in **D14.3.2-1(2)** are to be complied with.

2 Pipes for measuring instruments and remote control equipment

Steel pipes for measuring instruments and remote control equipment provided in cargo oil tanks are to have minimum thickness of Schedule 80 specified in **Table K4.16, Part K of the Rules**, except in cases where such pipes have openings inside cargo oil tanks.

3 Scupper pipes and sanitary pipes

All pipe joints of scupper pipes and sanitary pipes passing through cargo oil tanks are to be welded joints. In addition, scupper pipes or sanitary pipes in spaces containing sources of ignition such as accommodation space are not to pass through cargo oil tanks.

4 Overboard discharge pipes (bilge or ballast pipes)

(1) Overboard discharge pipes passing through cargo oil tanks are to be dealt with under the following requirements **(a)** to **(d)**:

(a) Overboard discharge pipes are not to pass through the cargo oil tanks other than those having a relatively small capacity (*e.g.*, slop tanks, etc.).

(b) Such pipes in cargo oil tanks are to be short in length and all pipe joints in cargo oil tanks are to be welded joints. In cases where cast steel pipes are used, pipe thickness may be 15 *mm* or more.

(c) Internal surfaces of pipes are to be coated with paints having good corrosion resistance properties, except in cases where cast steel pipes specified in **(b)** above or steel pipes of an adequate thickness that make allowances for corrosion are used.

(d) No valves are to be provided in cargo oil tanks.

(2) Notwithstanding **(1)** above, in the case of tankers other than double hull tankers, overboard discharge pipes passing through cargo oil tanks are to be dealt with under the following requirements **(a)** to **(e)**:

(a) Overboard discharge pipes are not to pass through the cargo oil tanks other than those having a relatively small capacity (*e.g.*, slop tanks, etc.).

(b) All pipe joints in cargo oil tanks are to be welded joints. In cases where cast steel pipes are used, pipe thickness may be 15 *mm* or more.

(c) Bent pipes are to be provided adequately to absorb any expansion and contraction of the pipe line.

(d) Internal surfaces of pipes are to be coated with paints having good corrosion resistance properties, except in cases where cast steel pipes specified in **(b)** above or steel pipes of an adequate thickness that make allowances for corrosion are used.

(e) No valves are to be provided in cargo oil tanks.

5 The wording "flanged joints which have no risk of leakage" in **14.2.7-4** and **-5, Part D of the Rules** means welded flange joints rated at least a nominal pressure of 1.0 *MPa* or a nominal pressure one rank higher than required design pressure, whichever is greater.

6 The wording "expansion bends" in **14.2.7-5, Part D of the Rules** means expansion loops such as omega bends in piping systems to counteract excessive stress or displacement caused by thermal expansion or hull deformation which could be fabricated from straight lengths of pipe.

D14.2.8 Sounding Devices of Cargo Oil Tanks

1 The sounding device of cargo oil tanks is to be of the construction capable of measuring the ullage without opening the tank hatch cover. In cases where ullage hatches are provided, hatch closing devices are to be in accordance with the requirements of **4.5.3-3, Part R of the Rules**.

2 In cases where sounding pipes are provided, any open ends are to be led to the weather deck and to be provided with pipe heads having sluice valves or cocks which are fitted with automatic closing devices. However, for those sounding pipes with pipe diameters of 50 *mm* or less, screw-down type plugs and sluice valves or cocks which are not fitted automatic closing devices may be accepted.

3 In cases where level indicating devices are provided for those sounding devices specified in **14.2.8, Part D of the Rules**, such devices are to be of a type approved by the Society in accordance with **Chapter 4, Part 7 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**, which is separately specified. And, all approved devices are to be made public on the "**List of approved materials and equipment**".

D14.3 Piping Systems for Cargo Oil Pump Rooms, Cofferdams and Tanks adjacent to Cargo Oil Tanks

D14.3.1 Bilge Piping Systems, etc. for Cargo Oil Pump Rooms and Cofferdams adjacent to Cargo Oil Tanks

The open ends of sounding pipes specified in **14.3.1-4, Part D of the Rules** may be provided in pump rooms. However, in cases where any open ends are lower than the bulkhead deck, the requirements in **4.2.2(3)(e)(i), Part R of the Rules** are to be complied with.

D14.3.2 Ballast Tanks adjacent to Cargo Oil Tanks

1 Ballast piping systems of the forward ballast tanks, etc. (**14.3.2-1, Part D of the Rules**)

Ballast piping systems, etc. serving ballast tanks whose forward end is located afore of collision bulkheads and are adjacent to cargo oil tanks (hereinafter referred to as "forward ballast tanks") are to be in accordance with the following requirements in addition those in **14.3.2-2** to **14.3.2-4, Part D of the Rules**. However, ballast piping systems, in cases where they are as specified in the following **(2)** or **(3)** and serve ballast tanks which are not adjacent to cargo oil tanks, but whose forward end is located afore of collision bulkheads, are considered to be piping systems of forward ballast tanks and, therefore, are to be in accordance with the requirements for forward ballast tanks.

(1) Arrangements are to be made so that any ballast water in forward ballast tanks, except for those cases specified in the following **(2)** or **(3)**, can be ballasted/deballasted by pumps located in the forward part of the cargo tanks.

(2) In cases where ballast pipes of forward ballast tank are led to ballast pumps by passing through cargo oil tanks, except in cases where prohibited by **14.2.7, Part D of the Rules** or **D14.1.1**, the following requirements are to be complied with:

  (a) Flange joints with a nominal pressure less than 1 *MPa* are not to be used for pipe joints.

  (b) Stop valves are to be provided afore of collision bulkheads in addition to those bulkhead valves specified in **13.2.5-2, Part D of the Rules**.

  (c) Ballast pumps are to be provided in cargo oil pump rooms or other subdivisions that are without sources of ignition.

  (d) The requirements of **(a)** to **(e)** in the following **(3)** are to be complied with.

(3) In cases where ballast pipes of forward ballast tanks are led to other ballast piping systems serving ballast tanks which are adjacent to cargo oil tanks, the following requirements are to be complied with:

  (a) In applying the requirements specified in **Part H of the Rules**, forward ballast tanks are to be considered to be hazardous areas as specified in **4.3.1(2)(c), Part H of the Rules**.

  (b) Vent pipe openings provided for forward ballast tanks are to be located on open decks at an appropriate distance of not less than 3 *m* away from any sources of ignition. In addition, the area around such vent pipe openings is defined as a hazardous area in accordance with **4.3.1(2)(i), Part H of the Rules** and **4.3.1(3)(a), Part H of the Rules**.

  (c) Means are to be provided, on open decks, to allow measurement of the concentration of flammable gases within forward ballast tanks. In this case, such means may be a combination of portable detecting instruments and sampling pipes. Such sampling pipes may be those sounding pipes specified in the following **(d)** in cases where deemed appropriate by the Society.

  (d) Sounding pipes provided for forward ballast tanks are to be led to open decks.

  (e) Access into forward ballast tanks is to be direct from open deck. However, indirect access from open decks into the forward ballast tanks through enclosed spaces may be acceptable provided that the following (i) or (ii) is satisfied.

(i) In cases where enclosed spaces are separated from the cargo oil tanks, access into forward ballast tanks are to be a gas tight bolted manhole located in such enclosed spaces. In this case, a warning sign is to be provided at the manhole stating that the forward ballast tank may only be opened after it has been proven to be gas free or the electrical equipment which is not electrically safe in the enclosed space is isolated.

(ii) In cases where enclosed spaces have common boundaries with the cargo tanks, such enclosed spaces are to satisfy the relevant requirements of hazardous areas and are, in addition, to be well ventilated.

2 Ballast piping systems for ballast tanks adjacent to cargo oil tanks (**14.3.2-2, Part D of the Rules**)

Ballast piping systems for ballast tanks adjacent to cargo oil tanks is to be dealt with under the following requirements:

(1) In cases where both those ballast tanks adjacent to forward cargo oil tanks and those ballast tanks not adjacent thereto are provided afore of forward cargo oil tanks, ballast pipes for those tanks may be led to the same ballast/deballast pumps located afore of forward cargo oil tanks, except in cases where such pipes pass through cargo oil tanks.

(2) In cases where ballast tanks adjacent to cargo oil tanks are intended to be deballasted by cargo oil pumps in an emergency, spool pieces and screw-down non-return valves are to be provided on each ballast pipe at joints with cargo oil pipes. Furthermore, a warning notice is to be posted stating that spool pieces are to be removed except for in times of emergency.

3 Air vent pipes of ballast tanks adjacent to cargo oil tanks (**14.3.2-4, Part D of the Rules**)

(1) The wording "wire gauze to prevent any passage of flame" specified in **14.3.2-4, Part D of the Rules** means wire gauze meeting the following requirements:

(a) To be made of corrosion resisting material.

(b) To comprise either dual wire gauze with a mesh size finer than $850\,\mu m$ spaced at a distance of $25.4 \pm 12.7\ mm$, a single wire gauze with a mesh size finer than $500\mu m$, or others of equivalent performance thereto.

(2) The total sectional area of air vent pipes in cases where high level alarms or hatchways specified in **14.1.2.1, Part 2-7, Part C of the Rules** are provided in ballast tanks adjacent to cargo oil tanks may be larger than the sectional area under the requirements of **13.6.3(1), Part D of the Rules** or 1,000 cm^2 whichever is smaller.

4 Open ends of sounding pipes for ballast tanks adjacent to cargo oil tanks (**14.3.2-5, Part D of the Rules**)

The requirements of **D14.3.1** also apply to the open ends of sounding pipes for ballast tanks adjacent to cargo oil tanks.

D14.3.3 Fuel Oil Tanks adjacent to Cargo Oil Tanks

The requirements of **D14.3.1** also apply to the open ends of sounding pipes for fuel oil tanks adjacent to cargo oil tanks.

D14.3.4 Pump Arrangements of Forward Compartments

In applying the requirements specified in **14.3.4, Part D of the Rules**, pipes conveying liquid and bilge suction pipes for tanks or void spaces at forward positions of ships are to comply with the following requirements in cases where these pipes do not passing through cargo oil tanks:

(1) Pipes for tanks or void spaces adjacent to forward ends of cargo oil tanks may be led to aft pump rooms. Fuel oil transfer pipes may be led to pumps located in engine rooms.

(2) Pipes for tanks or void spaces not adjacent to cargo oil tanks may be led to pumps located in engine rooms or pumps to which neither cargo oil pipes nor dangerous ballast pipes provided in aft pump rooms are led.

D14.5 Piping Systems for Combination Carriers

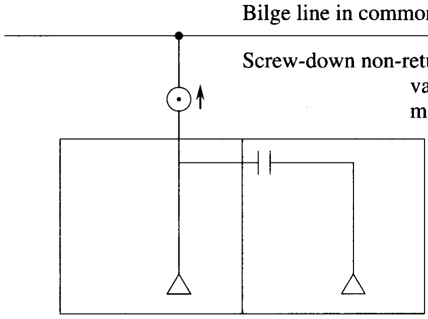
D14.5.3 Bilge Piping Systems

For bilge piping in common service with cargo oil piping or branch bilge suction pipe by eductors, the requirements given in **Table D14.5.3-1** are to be complied with.

Table D14.5.3-1 Bilge Suction Arrangement in Cargo Hold (Common Service Piping or Using Eductor)

Type of cargo hold (Remark 1)	Bilge suction main	Bilge suction branch	Bilge well	Provisions of valves, blind flanges, etc. (Remark 2)
Solid cargo / oil hold	Common service with cargo oil main	Exclusive	Exclusive	Screw-down non-return valve for branch line; blind flange for openend (Remark 4)
		Partial common service (Remark 3)	Exclusive (Remark 3)	Screw-down non-return valve and blind flange for bilge suction branch line
		Common service with cargo oil suction branch line	Common service with cargo oil suction well	Screw-down non-return valve for branch line
	Use of eductor	—	Exclusive	Screw-down non-return valve for branch line; blind flange for open end
Ballast / solid cargo hold	Common service with cargo oil main	Exclusive	Exclusive	Screw-down non-return valve for suction side; blind flange for open end
	Use of eductor	—	—	Screw-down non-return valve for suction side; blind flange for open end
Segregated solid cargo hold	Common service with cargo oil main	Exclusive	Exclusive	Stop valve and screw-down non-return valve for branch line
	Use of eductor	—	Exclusive	Screw-down non-return valve for suction side

Remarks:
1. The difinition of the type of cargo hold is in accordance with **14.7.2**, Part **D** of the Rules.
2. Valves, in each case, are to be capable of being operated at control position above the bulkhead deck.
3. Refer to the example shown below.



4. Wells are to be closed with blind plates or open ends are to be closed with blind flanges.

D15 STEERING GEARS

D15.1 General

D15.1.1 Scope

1 Manual steering gears are to be in accordance with the requirements of **15.1**, **15.2.1** through **15.2.3**, **15.2.8** through **15.2.10**, **15.3.1**, **15.4** (excluding **15.4.8-2**) and **15.5, Part D of the Rules** and the requirements applicable of this **D15**.

2 Quadrants, chains, rods and leading-block of manual steering gears are to be as specified in the following requirements:

(1) The scantlings of quadrants are to comply with the following requirements in **(a)** to **(c)**:

(a) In cases where three arms are provided, scantlings of quadrants are not to be less than those given in the following:

i) Boss:

$$H_c = 4.27 \cdot \sqrt[3]{T_R K_Q}$$

$$D_c = 7.69 \cdot \sqrt[3]{T_R K_Q}$$

ii) Arm at its root:

$$B_c = 3.29 \cdot \sqrt[3]{T_R K_Q}$$

$$T_c = 1.67 \cdot \sqrt[3]{T_R K_Q}$$

iii) Arm at its outer end:

$$B_0 = 2.22 \cdot \sqrt[3]{T_R K_Q}$$

$$T_0 = 1.07 \cdot \sqrt[3]{T_R K_Q}$$

where

T_R : Rudder torque specified in **13.2.3, Part 1, Part C of the Rules** (*N-m*).

K_Q : Material coefficient of the quadrant, specified in **13.2.1.2, Part 1, Part C of the Rules**.

H_c : Required depth of boss (*mm*).

D_c : Required outer diameter of boss (*mm*).

B_c : Required breadth of arm at its root (*mm*).

T_c : Required thickness of arm at its root (*mm*).

B_0 : Required breadth of arm at its end (*mm*).

T_0 : Required thickness of arm at its end (*mm*).

(b) In cases where two arms are provided, the breadth and thickness of such arms are to be not less than 1.1 *times* those specified in **(a)**. In cases where four arms are provided, the breadth and thickness of such arms may be reduced to 0.9 *times* those specified in **(a)**.

(c) In cases where loose quadrants are used in addition to tillers fixed to rudder stocks, any arms of loose quadrants may be of the dimensions given in **(a)iii)** throughout their length.

(2) The diameter of studless chains for steering is not to be either less than 9.5 *mm* or less than the value obtained from the following formula, whichever is greater.

$$d_s = 3.36 \sqrt{\frac{T_R K_c}{R}}$$

where

d_s : Required diameter of chains for steering (*mm*).

T_R : Rudder torque specified in **13.2.3, Part 1, Part C of the Rules** (*mm*).

K_c : Material coefficient of the chain, specified in **13.2.1.2, Part 1, Part C of the Rules**.

R : Length of tiller or radius of quadrant measured from the centre of rudder stock to the centre line of steering chains (*mm*).

(3) The diameter of steering rods is to be equal to or larger than 1.25 times the diameter of the steering chains obtained from **(1)** above.

(4) Leading blocks of steering chains are to be so arranged as to make the length of such steering chains as short as practicable, to lead the chains easily to the quadrant and to avoid any sharp bends.

(5) The diameter of lead block sheaves, measured at the centre line of steering chains, is to be equal to or larger than 16 times the diameter of the chains. The diameter of sheave pins is to be two times or more of that of chains.

(6) In cases where steering chains are led at angles less than 120 degrees, the diameters of sheaves and pins are to be made 1.25 times those specified in **(4)** above or larger respectively.

(7) Frames, base plates, pins and other parts of blocks subjected to mechanical shocks are not to be of cast iron. Furthermore, the sum of the sectional areas of those bolts connecting blocks to hulls is to be equal to or larger than the value obtained by following formula:

$$A_B = 2.4{d_s}^2$$

where

A_B : Required total sectional area of bolts (cm^2).

d_s : Required diameter of chains for steering (mm).

(8) For sheaves intended to be used with steering wire ropes, the radius of rope groove is to be of the value obtained by adding 0.8 mm to the radius of rope, and the radius of sheaves is to be equal to or larger than 14 times that of the rope.

D15.1.3 Drawings and Data

Operating instructions specified in **15.1.3(2)(b), Part D of the Rules**, are to include information about the importance of hydraulic fluid quality and its influence on the probability of hydraulic locking possibilities of two simultaneously operated power units. Operating instructions containing the same contents as those mentioned above, are to be kept on board.

D15.1.4 Display of Operating Instructions, etc.

The "appropriate instructions for emergency procedures" specified in **15.1.4-2, Part D of the Rules**, are to simply indicate those emergency procedures corresponding to the design of steering gear (for example, in order to shut down any failed system indicated by an alarm system), and are to be fitted at suitable places at steering control posts on navigation bridges.

D15.2 Performance and Arrangement of Steering Gears

D15.2.1 Number of Steering Gears

1 Adequate installation of blocks and tackles operating tillers or quadrants may be accepted as auxiliary steering gear.

2 In cases where manual steering gear is being used as the main steering gear, spare steering chains for the portion connected to the tiller are to be provided.

3 In cases where the auxiliary steering gear as specified in **15.2.1-1, Part D of the Rules** is of hydraulic type, the rudder actuator can serve in common with that for the main steering gear. Furthermore, parts of the hydraulic piping for rudder actuators of main steering gear may be used in common with those for auxiliary steering gear. However, in these cases, pipe lengths of common use parts are to be as short as practicable.

D15.2.3 Performance of Auxiliary Steering Gear

For auxiliary steering gear in ships with a speed as defined in **2.1.8, Part A of the Rules** that is less than 7 *knots*, the requirement of **15.2.3(1), Part D of the Rules** apply by construing the wording "at one half of the speed specified in **2.1.8, Part A of the Rules** or 7 *knots*, whichever is greater" therein as "at navigable speed."

D15.2.7 Electrical Installations for Electric and Electrohydraulic Steering Gear

1 Motors for steering gear power units may be rated for intermittent power demand. In such cases, the rating is not to be less than specified in **(1)** or **(2)**. Furthermore, S3 and S6 are to be in accordance with *IEC* 60034-1 or *JIS C* 4034-1.

(1) For motors of electric steering gear power units : S3-40%

(2) For motors of electrohydraulic steering gear power units : S6-25%

2 In cases where steering gear circuits, fed through electronic inverter units which control steering gear turning speed controls and their currents, are limited to being not more than the rated current of such electronic inverters, the requirements to provide protection devices against excess current specified in **15.2.7-6, Part D of the Rules** may be omitted. In these cases, they are to comply with the following requirements:

(1) Overload alarms specified in **15.2.7-5, Part D of the Rules** are to be set to values not greater than the rated loads of electronic

inverters.

(2) Over-current and over-voltage protection devices are to be provided in electronic inverter units. In cases where such protection devices are operated, audible and visual alarms are to be activated at navigation bridges and at positions from which main engines are normally controlled.

(3) Functions to reduce output power of electronic inverters working before those protection devices specified in **(2)** above are to be provided in electronic inverter units. In addition, in cases where such functions are operated, audible and visual alarms are to be activated at navigation bridges and at positions from which main engines are normally controlled. However, in cases where there are fears that semiconductor elements will totally fail within a short period of time, it is acceptable to cut off the output power of electronic inverter units.

D15.2.8 Position of Steering Gears

1 Interpretation of "Suitable arrangements to ensure working access to steering gear machinery and controls" required in **15.2.8-2, Part D of the Rules** is as follows:

(1) Access ways to steering gears and controls

(a) Walkways for approaching steering gears and controls from entrances of steering gear rooms are to be arranged.

(b) Standard widths of such walkways are to be 600 *mm* and they are to be provided with non-slip surfaces.

(c) Adequate handrails are to be placed at least in on one side of such walkways. Handrails, in principle, are to be of a fixed type and made of steel. In cases where installation of such fixed types seems to be impracticable, stanchions and ropes (wires) of disconnection types may be used.

(2) Handrails and work areas around steering gears

(a) Steel handrails and work areas for ensuring working conditions are to be arranged around or in the vicinity of steering gears.

(b) Work areas with 600 *mm* standard width and non-slip surfaces are to be arranged around or in the vicinity of steering gears.

(3) The wording "non-slip surface floor" means gratings (grids), buckboards, floors with surface covering material for the non-slip and non-slip coating floors. Non-slip surface covering materials are to be durable enough to be able to be used for long periods of time.

D15.2.9 Means of Communication

Means of communication between navigating bridges and steering gear compartments are not to depend solely on shipboard telephone systems for general purpose.

D15.3 Controls

D15.3.1 General

1 It may be acceptable that only one set of floating levers or other mechanical follow-up control systems are provided.

2 The two independent control systems specified in the requirements of **15.3.1-1(2), Part D of the Rules** are to be so arranged that a mechanical or electrical failure in one of them will not render the other one inoperative.

3 The control systems and relevant components specified in the requirements of **15.3.1-1(2), Part D of the Rules** are to comply with following requirements:

(1) Wires, terminals and the components for duplicated control systems installed in units, control boxes, switchboards or bridge consoles are to be separated as far as practicable. In cases where enough separation is not practicable, separation may be achieved by means of a fire retardant plate.

(2) All electric components of the control systems are to be duplicated.

(3) In cases where a joint steering mode selector switch (uniaxial switch) is employed for both control systems, the connections for the circuits of the control systems are to be divided accordingly and separated from each other by an isolating plate or by air gap.

(4) In cases where double follow-up controls are arranged (Refer to **Fig. D15.3.1-2**), the follow-up amplifiers are to be designed and independently supplied so as to be electrically and mechanically separated. In cases where both non-follow-up controls and follow-up controls are arranged, the follow-up amplifiers are to be protected selectively. (Refer to **Fig. D15.3.1-3**)

(5) Control circuits for additional devices, e.g. steering lever or autopilot, are to be arranged for all-pole disconnection. (Refer to

Fig. D15.3.1-1 to Fig. D15.3.1-3)

(6) In respect to control systems, in cases where feed-back units and limit switches are arranged, such devices are to be separated electrically and mechanically and connected to the rudder stock or rudder actuator separately.

(7) Hydraulic system components in the power actuating or hydraulic servo systems controlling the power systems of the steering gear (e.g. magnetic valves, etc.) are to be duplicated and arranged separately. In cases where there are two or more separate power units and the piping to each power unit can be isolated, the hydraulic system components in the control systems that are part of a power unit may be regarded as being duplicated and separated.

4 Amplifiers, relays, etc., included in control systems may also be used for automatic pilot systems.

5 For electrohydraulic steering gears equipped with power units comprising variable-displacement pumps, two sets each of hydraulic servo cylinders and associated hydraulic systems (including pump driving electric motors and control equipment) or electric servo motors for controlling displacement of pump plungers are to be provided.

6 In general, the following cases are not considered to be one of the cases "where hydraulic locking, caused by a single failure, may lead to loss of steering" that is specified in **15.3.1-6, Part D of the Rules**,

(1) Steering systems with performance at least equal to that required for auxiliary steering gear are fitted as stand-by systems and are operable from navigation bridges. In such cases, stand-by systems are to be designed so that they do not run parallel using interlocking devices, etc.

(2) Not less than 3 systems are operated parallel and, in the case of a single failure, steering capability at least equal to that required for auxiliary steering gears is maintained.

(3) Steering gears designed to avoid leading to any loss of steering by automatically by-passing failed systems using duplicated control valve systems. These arrangements are subject to special consideration with respect to any reduced reliability due to increased complexity.

7 Those "audible and visual alarms, which identify failed systems" specified in **15.3.1-6, Part D of the Rules**, are, in general, to be activated under the following conditions:

(1) In cases where positions of variable displacement pump control systems do not correctly respond to given commands.

(2) In cases where incorrect positions of 3-way full flow valves or similar constant delivery pump systems are detected.

8 The location of sensors for those alarms specified in the aforementioned **-7**, are to be as near as possible to actuators. However, in cases where two or more pumps are mechanically interconnected by floating bars or by similar devices, special consideration does not need to be given to their breakage. An example of some acceptable locations of alarm sensors is given in **Fig. D15.3.1-4**.

D15.3.2 Change-overs from Automatic to Manual Steering

1 At any rudder angle, change-overs from automatic to manual steering are to be available within 3 *seconds* and at most take two control operation attempts.

2 Change-overs from automatic to manual steering are to be available under any circumstances including cases of automatic pilot failure.

3 Devices to change-over from automatic to manual steering are to be installed close to positions where steering devices are normally operated.

D15.4 Materials, Constructions and Strength of Steering Gears

D15.4.7 Tillers, etc.

1 In cases where scantlings of arms are reduced in accordance with **15.4.7-2(4), Part D of the Rules**, the following requirements are to be complied with:

(1) For the tillers designed so that equal torque is applied on each arm:

Required values of Z_{TA} and A_R specified in **15.4.7-2(2), Part D of the Rules** and **(3)** respectively, may be multiplied by $1/n$. In this case "n" means the number of arms.

(2) For the tillers designed so that unequal torque is applied on each arm:

Required values of Z_{TA} and A_R specified in **15.4.7-2(2), Part D of the Rules** and **(3)** respectively, may be multiplied by α. In this case "α" means the ratio of the torque applied to the arm to the total torque.

2 The wording "to the satisfaction of the Society" specified in **15.4.7-5, Part D of the Rules** means to comply with the requirements specified in 13.2.8.4, Part 1, Part C of the Rules.

3 Scantlings of tillers for the exclusive use of auxiliary steering gear are to be of a strength that is more than 1/2 of that specified in the requirements of 15.4.7-2, Part D of the Rules.

D15.4.9 Buffers

Steering gears using steering chains and rods are to be designed so that spring buffers are not be closed solid at loads that are 7/8 of the proof test load of such steering chains.

D15.5 Testing

D15.5.2 Tests after Installation On Board

The wording "steering gear is designed to avoid any hydraulic locking" specified in 15.5.2-3, Part D of the Rules means those steering gears designed not to run parallel using interlocking devices, etc. or those steering gears designed to maintain their steering capability or to recover such capability by automatically by-passing failed systems. In the case of steering systems where hydraulic locking is not anticipated to occur because mechanical linkages of floating bar or similar which the considerations of breakages are waived have been provided, these demonstrations may be omitted.

Fig. D15.3.1-1 Example Layout of Control Systems with Double Non Follow-up Control and Autopilot or Other Additional Control

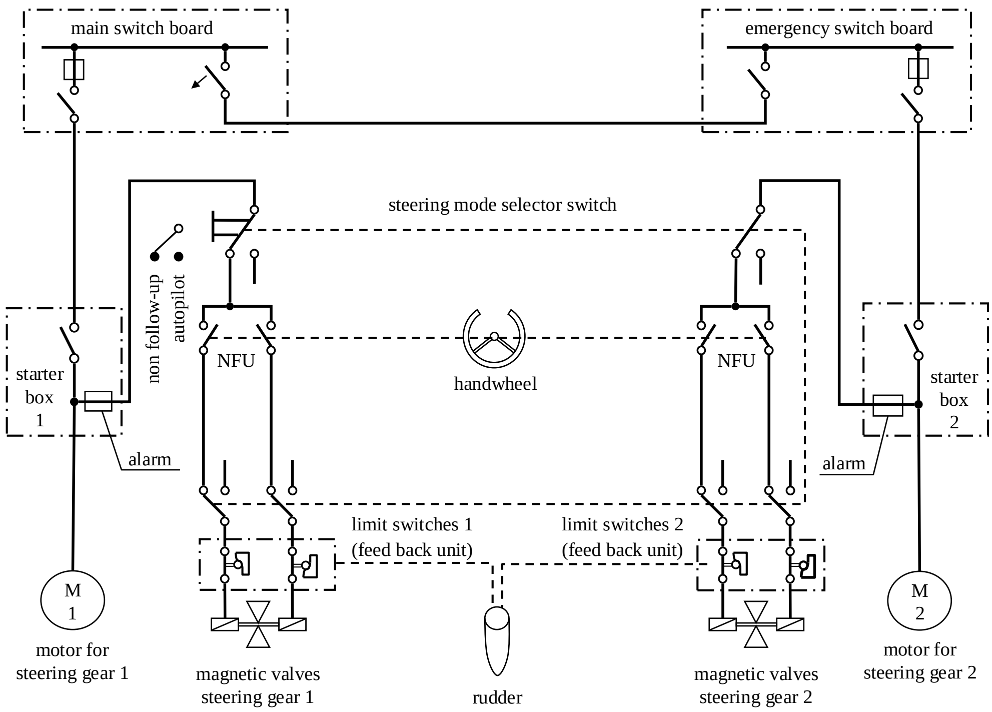


Fig. D15.3.1-2 Example Layout of Control Systems with Double Follow-up Control and Autopilot or Other Additional Control

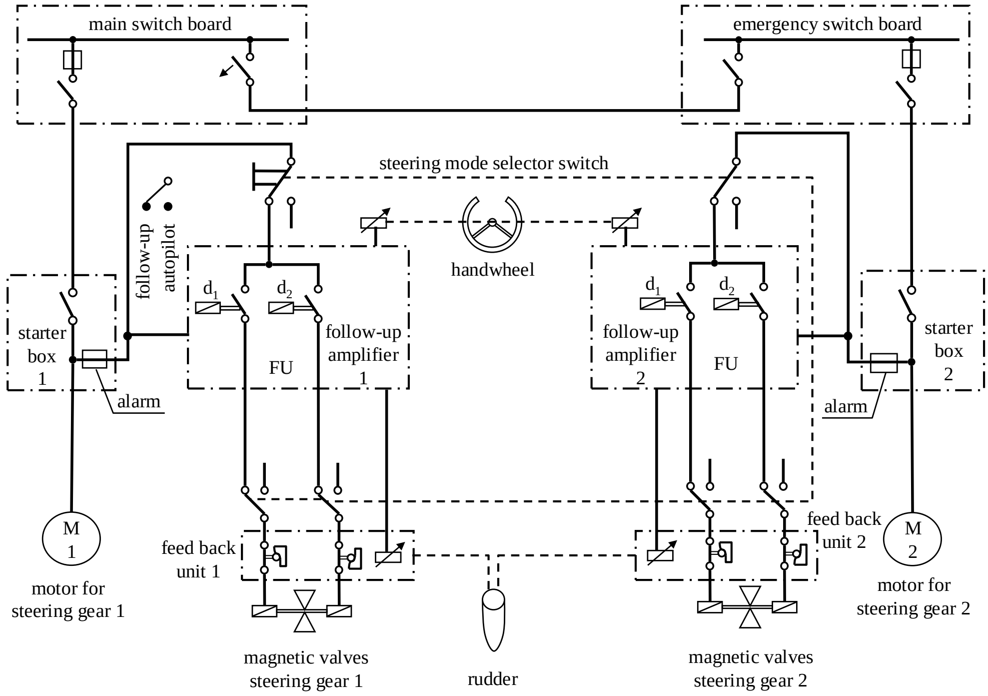


Fig. D15.3.1-3 Example Layout of Control Systems with Double Non Follow-up Control, Follow-up Control and Autopilot or Other Additional Control

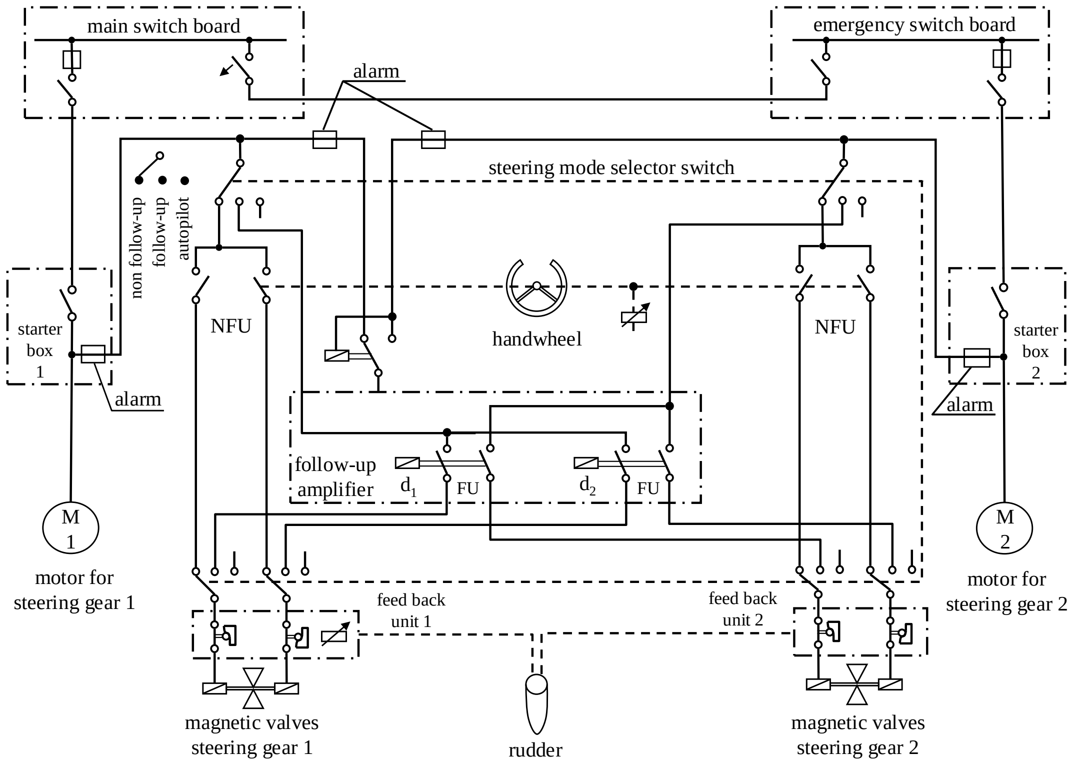
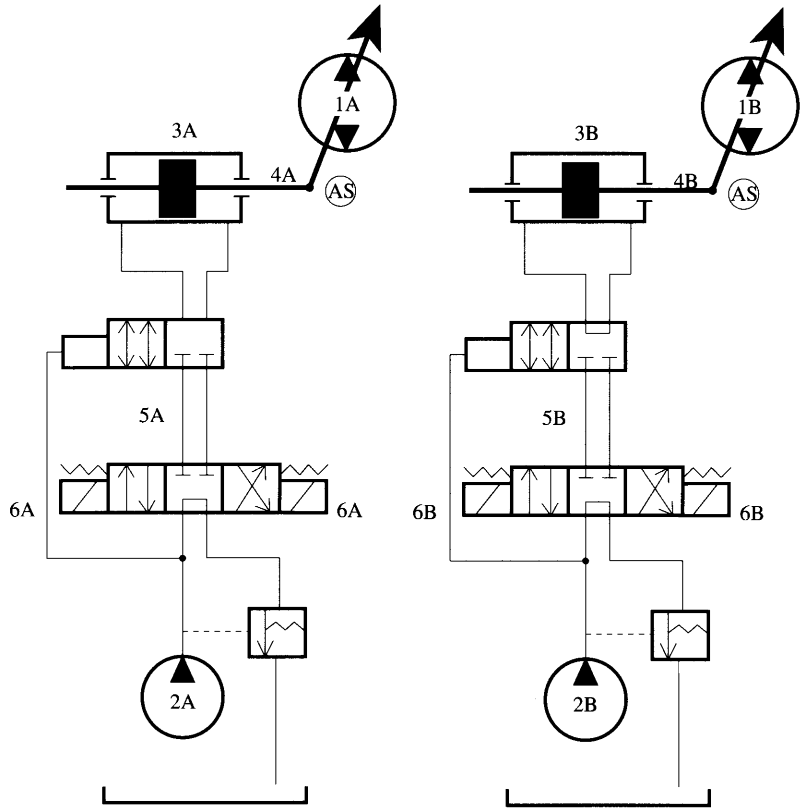
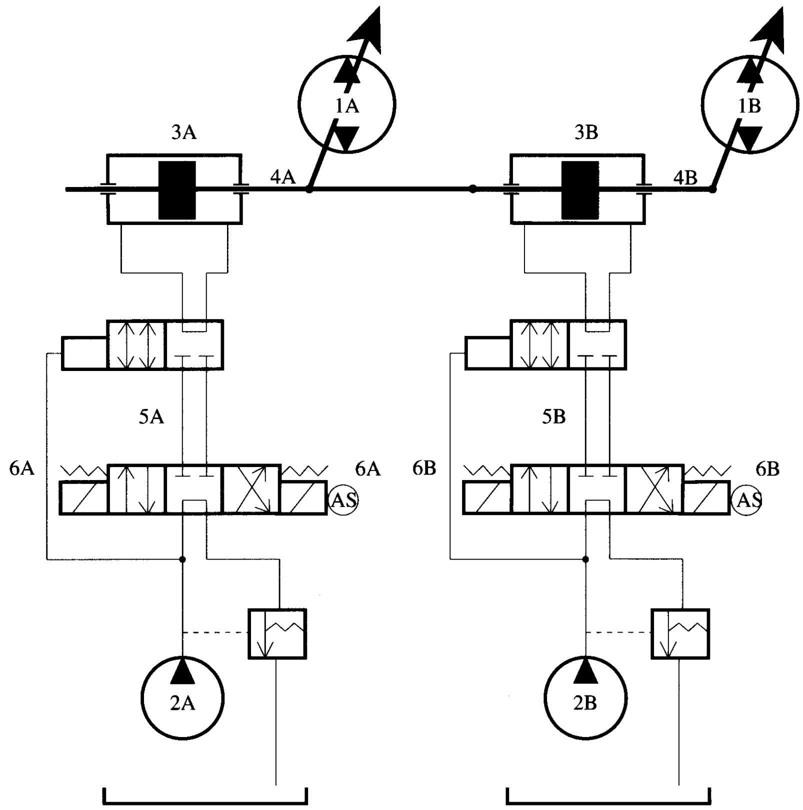


Fig. D15.3.1-4 Example Layout of Hydraulic Locking Alarm Sensors



(a) Separated system



(b) Mechanically interconnected system

AS denotes location of alarm sensors

1 Main pump of variable displacement type.

2 Pilot pump.

3 Control actuator.

4 Control linkage.

5 Solenoid controlled 3-way valve.

6 Solenoid.

(Note)

Where systems are so designed not to run 1A & 1B in (a) nor 2A & 2B in (b), the alarm devices are not required.

D16 WINDLASSES AND MOORING WINCHES

D16.2 Windlasses

D16.2.1 General

As "standard or code of practice recognized by the Society" referred to in **16.2.1-2, Part D of the Rules**, the followings are examples of standard recognized but not limited to:

(1) *SNAME T & R Bulletin* 3-15 "Guide to the Design and Testing of Anchor Windlasses for Merchant Ships"

(2) *ISO* 7825 "Deck machinery general requirements"

(3) *ISO* 4568 "Shipbuilding - Sea-going vessels - Windlasses and anchor capstans"

(4) *JIS* F6714 "Windlasses"

(5) *BS MA*35 "Specifications for Ship Deck Machinery Windlass"

D16.2.2 Drawings and Data

1 "Windlass design specifications" specified in **16.2.2(1)(a), Part D of the Rules** are to be include the following in addition to windlass particulars:

(1) Anchor and chain cable particulars

(2) Maximum anchorage depth

(3) Performance criteria

(4) Standard or code of practice of compliance

2 "Windlass arrangement plan" specified in **16.2.2(1)(b), Part D of the Rules** are to be show all of the components of the anchoring/mooring system. The followings are examples of the components:

(1) Prime movers, shafting, cable lifters, anchors, chain cables, brakes and controls

(2) Mooring winches, wires and fairleads, if they form part of the windlass machinery

(3) Marking of nominal size of chain cable and maximum anchorage depth

3 "Dimensions, materials and welding details of torque-transmitting components and load-bearing components" specified in **16.2.2(1)(c), Part D of the Rules** are to comply with the followings:

(1) Information of mooring winches are to be included in case where the mooring winch is one with a windlass.

(2) Proposed materials are to be indicated.

(3) Weld joint designs, the degree of non-destructive examination of welds and post-weld heat treatment are to be indicated.

4 "Drawings and data concerning hydraulic systems" specified in **16.2.2(1)(d), Part D of the Rules** are to include the following:

(1) Piping diagram along with system design pressure

(2) Safety valves arrangement and settings

(3) Material specifications for pipes and equipment

(4) Typical pipe joints, as applicable

(5) Technical data and details for hydraulic motors

5 "Calculated strength for torque-transmitting components and load-bearing components" specified in **16.2.2(2)(a), Part D of the Rules** are to comply with the following:

(1) It is to be demonstrated that torque-transmitting components and load-bearing components comply with a standard or code of practice recognized by the Society.

(2) Analyses for gears are to be in accordance with a standard recognized by the Society.

6 "Load calculations" specified in **16.2.2(2)(c), Part D of the Rules** are to demonstrate that the prime mover is capable of attaining the hoisting speed, the required continuous duty pull and the overload capacity specified in **16.2.4, Part D of the Rules**.

7 "Operation and maintenance procedures" specified in **16.2.2(2)(e), Part D of the Rules** are to show the maximum anchorage depth.

D16.2.3 Materials and Fabrication

1 "Standards recognized by the Society" referred to in **16.2.3-1(1)**, **-2(2)** and **-2(4)(b), Part D of the Rules** means national or international standard such as *JIS* or *ISO*.

2 The "deposited weld metal tests" referred to in **16.2.3-2(4)(b), Part D of the Rules** means the deposited metal test specified in **Chapter 6, Part M of the Rules** or an equivalent test. The tests are to be carried out at the same time as the tests for approval of welding procedures and related specifications.

D16.2.4 Design

1 The continuous duty pull specified in **16.2.4-2(2)(a) of the Rules** is based on the following conditions:

(1) Ordinary stockless anchors are used.

(2) The anchor masses are assumed to be the masses as given in **14.3, Part 1, Part C of the Rules** and **Chapter 2, Part L of the Rules**.

(3) One anchor is hoisted at a time.

(4) The effects of buoyancy and hawse pipe efficiency (assumed to be 70 %) have been accounted for.

2 The strength of windlass mounts in **16.2.4-2(7)(b)**, **Part D of the Rules** is to comply with the following requirements:

(1) The bolts securing the windlass are to comply with the following conditions:

$$\frac{\sqrt{R_i^2+3F_i^2}}{A_i} \leq \frac{\sigma_{Yb}}{2}$$

where

σ_{Yb} : Yield strength or 0.2% proof strength of bolt material (*N/mm*^{2})

i : The bolt group number. Windlasses are supported by N_b bolt groups, each containing one or more bolts. (*See* **Fig. D16.2.2-1**)

R_i : Axial force in bolt group (or bolt) *i* (*N*), positive in tension and R_i = 0 if negative. In addition, the value of R_{yi} is the greater value in bolt group *i* when P_y is acting separately on both the inboard and the outboard directions.

$$R_i = R_{xi} + R_{yi} - R_{si}$$

$$\begin{cases} R_{xi} = P_x h x_i A_i / I_x \\ R_{yi} = P_y h y_i A_i / I_y \end{cases}$$

R_{si} : Static reaction at bolt group *i*, due to weight of windlass (*N*), where R_{si} is calculated according to the following equation:

$$R_{si} = mg/N_b$$

F_i : Shear force applied to the bolt group *i* (*N*). However, in cases where parts of those supports specified in **(2)** support a part or all of these forces, F_{xj} and F_{yj} specified in **(2)** may be deducted from the following F_{xi} and F_{yi} respectively.

$$F_i = \sqrt{{F_{xi}}^2 + {F_{yi}}^2}$$

$$\begin{cases} F_{xi} = (P_x - \alpha g m)/N_b \\ F_{yi} = (P_y - \alpha g m)/N_b \end{cases}$$

N_b : The amount of bolt groups

P_x : Force acting normal to the shaft axis (*N*), which is applied 2×10^5 (*N/m*^{2}) normal to the shaft axis and away from the forward perpendicular, over the projected area in this direction (*m*^{2}).

P_y : Force acting parallel to the shaft axis (*N*), which is applied 1.5×10^5 (*N/m*^{2}) parallel to the shaft axis and acting separately on both the inboard and outboard directions, over the multiple of *f* times the projected area in this direction (*m*^{2}).

$f = 1 + B/H$ but not greater than 2.5

B : Width of the windlass measured parallel to the shaft axis (*mm*)

H : Overall height of the windlass (*mm*)

h : Shaft height above the windlass mounting (*mm*)

x_i, y_i : x and y coordinates of bolt group *i* from the centroid of all N_b bolt groups, positive in the direction opposite to that of the applied force. (*See* **Fig. D16.2.2-1**)

A_i : Cross sectional area of all bolts in group i (mm^2)

I_x : $\Sigma A_i x_i^2$ for bolt groups N_b (mm^4)

I_y : $\Sigma A_i y_i^2$ for bolt groups N_b (mm^4)

α : Coefficient of friction, equal to 0.5

m : Mass of the windlass (kg)

g : Gravity, equal to 9.81(m/sec^2)

(2) In cases where supports, except securing bolt supports, bear a part or all of the shearing forces specified in **(1)**, these supports are to comply with the following conditions:

$$\frac{F_{xj}}{A_{xj}} \leq \frac{\sigma_{Ys}}{2\sqrt{3}}$$

$$\frac{F_{yj}}{A_{yj}} \leq \frac{\sigma_{Ys}}{2\sqrt{3}}$$

σ_{Ys} : Yield strength or 0.2% proof strength of supports' material (N/mm^2)

j : The number of the support.

F_{xj} : Shear force applied to the support j acting normal to the shaft axis (N)

F_{yj} : Shear force applied to the support j acting parallel to the shaft axis (N)

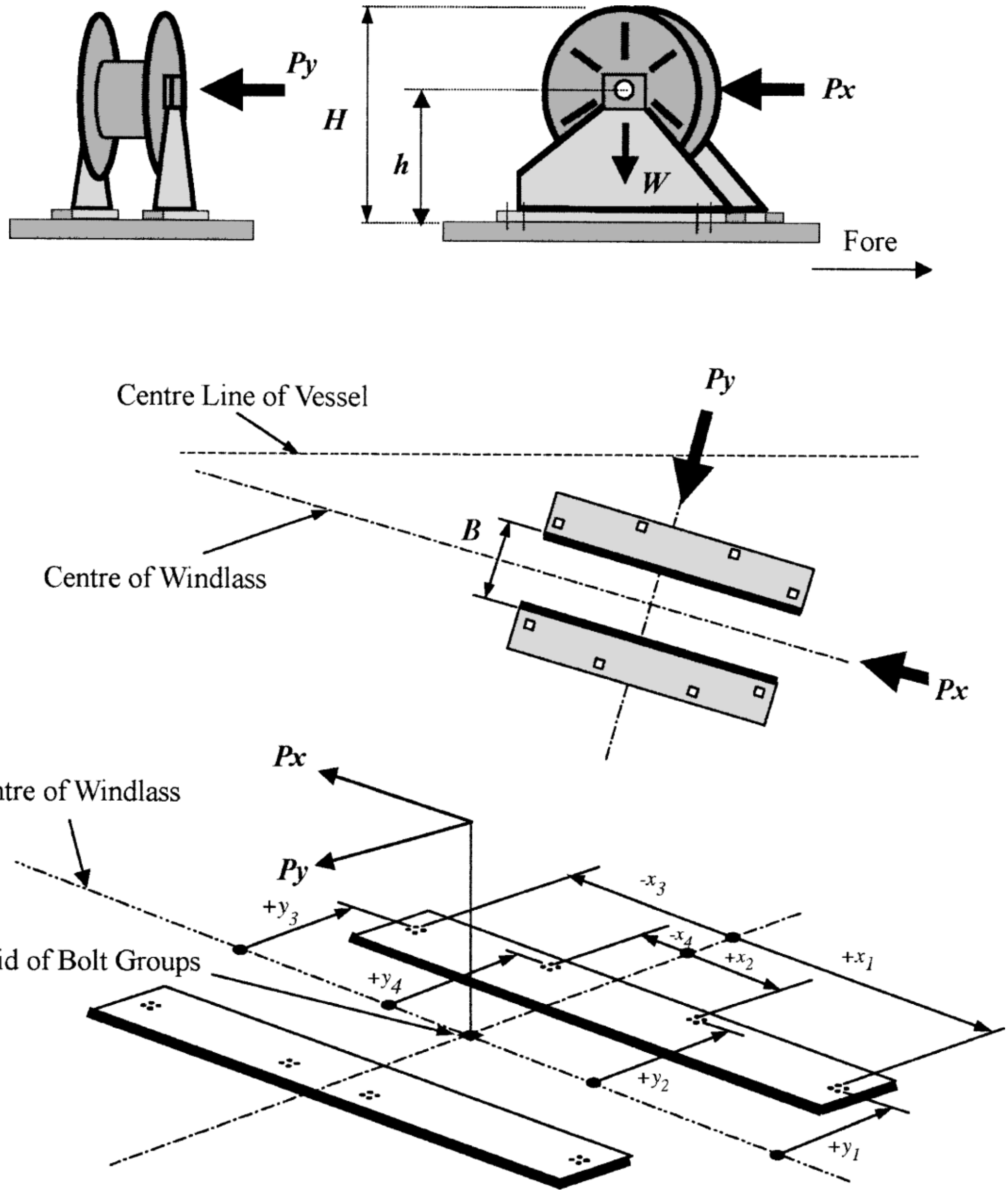
A_{xj} : Effective sectional area of the support j for shear force acting normal to the shaft axis (mm^2)

A_{yj} : Effective sectional area of the support j for shear force acting parallel to the shaft axis (mm^2)

(3) In cases where pourable resins are incorporated into holding down arrangements, due consideration is to be taken in the calculations.

(4) Axial tensile and compressive forces and lateral forces, calculated according to **(1)** and **(2)**, are to be considered when designing windlass supporting structures.

Fig. D16.2.2-1 Sign Convention



Notes:

1. P_y is to be examined from both inboard and outboard directions separately.
2. The sign convention for y_i is to be reversed when P_y is from the direction opposite to that shown.
3. $W = mg$

D17 REFRIGERATING MACHINERY AND CONTROLLED ATMOSPHERE SYSTEMS

D17.1 General

D17.1.1 Scope

1 "Refrigerating machinery forming refrigerating cycle" in **17.1.1-1, Part D of the Rules** contains condensers, receivers, evaporators, piping and associated equipment, etc.

2 Refrigerating machinery with compressors of 7.5 *kW* or less using *R*134*a*, *R*404*A*, *R*407*C*, *R*407*H*, *R*410*A*, *R*449*A* or *R*507*A* as their primary refrigerant are to be suitable for use, their service conditions and the surrounding environment on board.

3 Refrigerating machinery using *R*717 as their primary refrigerant are to comply with the requirements specified in **-4** to **-14** given below, in addition to those specified in the Rules.

4 Ammonia refrigerating machinery drawings and data

Drawings and data to be submitted, in addition to those specified in **17.1.2 of the Rules**, are generally as follows:

(a) *R*717 Refrigerant Piping Diagrams

(b) Gas Detector Arrangements

(c) General Arrangement of Refrigerating Machinery Compartments

5 Ammonia refrigerating machinery general requirements

(1) Cargo refrigerating machinery using *R*717 refrigerants are to be indirect refrigerating systems using brine.

(2) Pressure vessels used in refrigerating machinery are to be classified as pressure vessels Group I which are specified in **Chapter 10, Part D of the Rules**, and primary refrigerant pipes are to be classified as Group I pipes specified in **Chapter 12, Part D of the Rules**.

(3) The design pressure of pressure vessels and pipes which make up refrigerating machinery is not to be less than 2.3 *MPa* at the high pressure side and not to be less than 1.8 *MPa* at the low pressure side.

(4) Refrigerating machinery is to be provided with auxiliary receivers of adequate capacity so that repairs and maintenance can be carried out without discharging any gas to the atmosphere. However, auxiliary receivers may be dispensed with, if the refrigerant in the receiver with the largest capacity can at least be stored in some other receiver.

6 Ammonia refrigerating machinery materials

(1) Materials easily corroded (copper, zinc, cadmium, or their alloys) and materials containing mercury are not to be used at locations where they might come into contact with ammonia.

(2) Nickel steel is not to be used in pressure vessels and piping systems.

(3) Cast iron valves are not to be used in refrigerant piping systems.

(4) Materials for sea water-cooled condensers are to be selected with sufficient consideration being given to the possibility of corrosion due to sea water.

(5) In cases where flat tanks of quick freezers (contact freezers) are manufactured by extrusion molding of aluminum alloys, materials are to be approved in accordance with **Chapter 5, Part 1 of the "Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use"**.

7 Ammonia refrigerating machinery piping arrangements

(1) Refrigerant piping is not to pass through accommodation spaces. If such piping passes through other spaces outside such compartments, it is to be led through ducts.

(2) Pipe joints of refrigerant piping systems are to be butt welded as far as practicable.

(3) Refrigerant gases discharged from pressure relief valves are to be absorbed in water, except when leading such gases to the low pressure side.

(4) If liquid level gauges made of glass are used at locations constantly under pressure, they are to comply with the requirements given below:

(a) Flat type glass is to be used in liquid level gauges, and construction is to be such that such gauges are adequately protected against any external impact.

(b) Construction of stop valves for liquid level gauges is to be such that the flow of liquid is automatically cut off if the glass breaks.

(5) Gases discharged from purging valves are not to be discharged directly into the atmosphere, but they are to be absorbed in water.

(6) Independent piping for the discharge of cooling sea water for condensers is to be used. This piping is to be led directly overboard without passing through any accommodation spaces.

8 Control and alarm systems of ammonia refrigerating machinery

Refrigerant compressors are to be provided with means for automatically stopping compressors when the pressure on the high pressure side of refrigerant piping systems becomes excessively high. In addition, alarm systems, which generate visible and audible alarms when such refrigerant piping systems are in operation, are to be installed in refrigerating machinery compartments and at monitoring positions.

9 Ammonia refrigerating machinery compartments

(1) Compartment where refrigerating machinery and storage vessels are installed (hereinafter referred to as “refrigerating machinery compartments”) are to be special compartments isolated by gastight bulkheads and decks from all other compartments so that any leaked ammonia does not enter other compartments. Refrigerating machinery compartments are to be provided with doors which comply with the following requirements:

(a) At least two access doors, as far apart as possible from each other, are to be provided in refrigerating machinery compartments. At least one access door is to lead directly to the weather deck. However, if it is not possible to provide an access door directly to the weather deck, then at least one access is to have air-lock type doors.

(b) Access doors not leading to weather decks are to be of a high tightly sealed and self-closing type.

(c) Access doors are to be capable of being operated easily and are to open outward.

(2) Penetrations on gastight bulkheads and decks where cables and piping from the refrigerating machinery compartment pass through are to be of gastight construction.

(3) The refrigerating machinery compartments are to be not adjacent to any accommodation spaces, hospital rooms or control rooms.

(4) Passages leading to refrigerating machinery compartments are to comply with the following requirements:

(a) Passageways adjacent to accommodation spaces, hospital rooms or control rooms are to be isolated by gastight bulkheads and decks.

(b) Passageways are to be lead directly to weather decks and be isolated from any passageways leading to accommodation spaces.

(5) Drain pans of adequate size are to be provided at positions which are lower than those refrigerating machinery and storage vessels in refrigerating machinery compartments so that liquid ammonia does not leak outside such compartments.

(6) Independent drainage systems are to be provided in refrigerating machinery compartments so that any drainage is not discharged into any open bilge wells or bilge ways of other compartments.

10 Gas expulsion system

Gas expulsion systems consisting of ventilation systems, gas absorption systems, water screening systems and gas absorption water tanks are to be installed in refrigerating machinery compartments, in accordance with **(1)** to **(5)** below so that any accidentally leaked gas can be quickly expelled from the such compartments.

(1) Mechanical ventilation systems which comply with the following requirements are, as a rule, to be installed in refrigerating machinery compartments so that these spaces can be ventilated all times.

(a) Such ventilation systems are to have enough capacity to ensure at least 30 air changes per hour in refrigerating machinery compartments.

(b) Such ventilation systems are to be independent of other ventilation systems on board ship, and are to be capable of being operated from outside refrigerating machinery compartments.

(c) Exhaust outlets are to be installed at horizontal distances of more than 10 *m* from the nearest air intake openings, openings of accommodation spaces, service spaces and control stations, and at vertical distances of more than 4 *m* from weather decks.

(d) Air intake openings are to be provided at low positions and exhaust openings are to be provided at high positions in refrigerating machinery compartments so that no gas accumulates inside compartments and exhaust ducts.

(e) Exhaust fans are to be of a construction that does not allow any sparks to be generated complying with **R4.5.4-1(2), Part R of the Guidance.** Protection screens of not more than 13*mm* square mesh are to be fitted in the inlet and outlet ventilation openings of the ducts fitted with such fans on the open deck. For the purpose of this requirement, as a rule, motors for driving the fans are to be of the exterior mounted type.

(2) Independent ventilation systems are to be installed in passageways leading to refrigerating machinery compartments. However, if ventilation systems, such as the ones specified in **(1)** above, are provided with ducts so that they can be used for expelling air from passageways, then independent ventilation systems need not be installed.

(3) Gas absorption systems satisfying any of the requirements given below, capable of excluding leaked gases quickly from the refrigerating machinery compartments, and capable of being operated from outside such compartments, are to be installed.

(a) Scrubbers

Scrubbers are to be designed with processing capacities adequate enough to restrict gas concentration at exhaust fans to well below 25 *ppm* as well as absorb ammonia in the largest receivers within 30 *minutes*; and,

Pumps for scrubbers are to start automatically when gas concentrations in refrigerating machinery compartments exceed 300 *ppm*.

(b) Water sprinkler systems

The quantity of sprinkler water is to be such that any leaked gases can be satisfactorily absorbed; and,

Nozzles are to be of types approved by the Society. As a rule, nozzles are to be positioned so that their range covers all refrigerating machinery in such compartments; and,

When gas concentrations in refrigerating machinery compartments exceed 300 *ppm*, pumps for sprinkler water are to start automatically.

(4) All doors of refrigerating machinery compartments are to be provided with water screening systems which can be operated from outside such compartments.

(5) Gas absorption water tanks complying with the requirements given below, are to be installed so that any leaked liquid ammonia can be quickly recovered.

(a) Such tanks are to be of a capacity sufficient enough to fully recover all of the water for absorbing refrigerants found in at least one refrigerating machinery unit.

(b) Automatic water supply systems are to be installed in such tanks so that fully-filled condition of such tanks is always maintained.

(c) Overflow from such tanks is to be diluted or neutralized and then discharged directly overboard, and pipes handling such overflows are not to pass through accommodation spaces.

(d) Means are to be provided in such tanks to recover any ammonia drainage generated in refrigerating machinery compartments. In addition, appropriate drain traps are to be provided to prevent any reverse flow of gas from such tanks.

(e) All vent pipes of such tanks are to be connected to exhaust pipes of those ventilation systems specified in **(1)** above.

11 Ammonia gas detection and alarm systems

(1) Gas detection and alarm systems complying with the following requirements are to be provided in refrigerating machinery compartments and other locations deemed necessary by the Society:

(a) At least one gas detector, complying with the requirements given below, is to be installed above each refrigerating machinery unit.

Such detectors are to activate alarms when the ammonia gas concentration exceeds 25 *ppm*; and,

When ammonia gas concentration exceeds 300 *ppm*, such detectors are to automatically stop refrigerating machinery, automatically activate gas exclusion systems, and activate alarms.

(b) An adequate number of flammable gas detectors are to be provided so that in cases where ammonia gas concentration reaches up to 4.5%, power supplies to electrical equipment in refrigerating machinery compartments are cut off and alarm systems are activated.

(c) Alarm systems are to generate visible and audible alarms near doors, inside and outside refrigerating machinery compartments and at monitoring locations.

(d) Manually-operated transmitters for leakage warnings are to be provided near the doors and outside refrigerating machinery compartments.

(2) Gas detection and alarms systems complying with the following requirements are to be provided in passageways leading to refrigerating machinery compartments and those other locations deemed necessary by the Society:

(a) Gas detectors are to activate alarm systems in cases where ammonia gas concentration exceeds 25 *ppm*.

(b) Alarm systems are to generate visible and audible alarms in passageways and near the doors of the refrigerating machinery compartments.

(3) Detectors are to be capable of continuous detection and are to be considered appropriate by the Society.

12 Electrical equipment for ammonia refrigerating machinery

(1) Electrical equipment in refrigerating machinery compartments required to be operated in the event of leakage accidents as well as gas detection and alarm systems and emergency lights are to be of certified as safe and explosion resistant types for use in the flammable atmosphere concerned.

(2) Electrical equipment in refrigerating machinery compartments other than those mentioned in **(1)** above, are required to be automatically switched off by circuit breakers located outside such compartments in cases where flammable gas detectors specified in **-11(1)(b)** activate.

(3) If water sprinkler systems are installed in refrigerating machinery compartments as gas absorption systems, all electrical machinery and equipment in such compartments are to be waterproof types.

13 Safety and protective equipment for ammonia refrigerating machinery

Safety and protective equipment for ammonia refrigerating machinery, as a rule, are to be provided as given below; and, such equipment is to be stored at locations outside refrigerating machinery compartments so that they can be easily retrieved in the event of any refrigerant leakage. Storage locations are to be marked with signs so that they can be easily identified.

(1) Protective clothing (helmets, safety boots, gloves, etc.)×2

(2) Self-contained breathing apparatus (capable of functioning for at least 30 *minutes*)×2

(3) Protective goggles×2

(4) Eye washer×1

(5) Boric acid

(6) Emergency electric torch×2

(7) Electric insulation resistance meter×1

14 Requirements for fishing vessels, etc.

Refrigerating installations provided in fishing vessels (as defined in *Regulation* 2, Chapter I, *SOLAS* 1974) of a length under 55 *m* or refrigeration machinery holding not more than 25 *kg* of ammonia are to be according to **(1)** to **(5)** given below, notwithstanding **-9** to **-12**.

(1) Refrigerating machinery may be installed in engine rooms. In such cases, drain pans are to be provided at positions lower than such refrigerating machinery.

(2) Ammonia gas exclusion systems are to comply with the following requirements:

(a) Ventilation systems with special hoods used for exhaust located above refrigerating machinery are to be provided and be capable of circulating air without any accumulation of ammonia gas. Fans of such ventilation systems are to be independent of engine room ventilation systems.

(b) Water sprinkler systems capable of sufficiently absorbing any leaked ammonia gas are to be provided in the vicinity of the installation locations of refrigerating machinery. Sprinkler hoses and water spraying nozzles are to be positioned so that water can be quickly sprayed when a leak occurs.

(3) Ammonia gas detection and alarm systems are to comply with the following requirements:

(a) Detection and alarm systems are to be provided to activate visible and audible alarms in monitoring rooms (or control rooms) and engine room entrances in cases where ammonia gas concentration exceeds 25 *ppm*. Such detectors are to be installed in the vicinity of the upper parts of refrigerating machinery, exhaust outlets and other locations deemed necessary by the Society.

(b) Detection and alarm systems are to be provided to activate visible and audible alarms in monitoring rooms (or control rooms) and engine room entrances in cases where ammonia gas concentration exceeds 300 *ppm*. Such detectors are to be installed in the vicinity of the upper parts of refrigerating machinery and other locations deemed necessary by the Society. Means are to be provided so that as soon as such concentrations are detected, refrigerating machinery automatically stops operating.

(c) Manually-operated transmitters for leakage warnings are to be provided near engine room exits. However, in cases where escape routes from monitoring rooms (or control rooms) are such that they pass through engine rooms, then such manually-operated transmitters are also to be provided near monitoring room (or control room) exists.

(d) Detectors are to be capable of continuous detection and are to be considered appropriate by the Society.

(4) Electrical equipment is not to be installed in the vicinity of any refrigerating machinery as far as possible.

(5) In cases where escape routes from monitoring rooms (or control rooms) pass through engine rooms, one of the self-contained breathing apparatus sets specified in **-13(2)** is to be provided in such monitoring rooms (or control rooms).

15 The terms used for controlled atmosphere systems of cargo holds are defined as follows:

(1) "Controlled atmosphere systems" means such systems which control and maintain oxygen content at low levels in cargo holds by introducing nitrogen gas therein in order to extend the life of cargoes as subsidiary installations for cargo refrigerating installations.

(2) "Controlled atmosphere zone" means air-tight cargo chambers have a controlled atmosphere.

(3) "Dangerous zone" means controlled atmosphere zones and enclosed zones which are adjacent to controlled atmosphere zones.

D17.1.2 Drawings and Data

For controlled atmosphere systems, three copies of the following drawings and data are to be submitted:

(1) Specifications of controlled atmosphere systems.

(2) General arrangements of controlled atmosphere zones, dangerous zones and controlled atmosphere systems.

(3) Arrangement of gas pressure and vacuum relief values (*PV* valves)

(4) Arrangement of ventilation systems for controlled atmosphere zones, nitrogen generating rooms and enclosed zones which are adjacent to controlled atmosphere zones.

(5) Other drawings and data deemed necessary by the Society.

D17.3 Controlled Atmosphere Systems

The requirements for controlled atmosphere systems are as follows:

(1) Controlled atmosphere systems for refrigerated chambers

(a) Air-tightness

i) Each controlled atmosphere zone is to be made as air-tight as possible.

ii) Deck and bulkhead penetrations of all pipes, electrical cables, trunks etc. in *CA* zones are to be suitably sealed and made air-tight.

iii) Adequate closing appliances, such as covers, doors and manholes are to be provided for openings in cargo hatches, entrances, etc. of controlled atmosphere zones in order to maintain the air-tightness.

(b) Personnel Protection

i) Closing appliances of hatches, entrances, ventilators, etc. of controlled atmosphere zones are to be constructed so they are not able to be easily released from the outside due to any impact forces or operational error under controlled atmosphere conditions. Hatch covers and doors at entrances are to be capable of being locked. Warning notices are to be posted at all openings to prevent inadvertent opening under controlled atmosphere conditions.

ii) Each controlled atmosphere zone is to be provided with a warning alarm which will be activated before any injection of nitrogen into the controlled atmosphere zone occurs in such a way that these inlet valves cannot be opened unless the alarm signal has been given.

(c) Protection of Controlled atmosphere zones

i) *PV* valves are to be provided to limit abnormal positive or negative pressure in each controlled atmosphere zone.

ii) Outlets for these valves are to be located as high as possible above upper decks to obtain the maximum disposal of nitrogen, but under no circumstances are they to be located less than 2 *metres* above the deck and 5 *metres* away from any air inlets and openings of accommodation spaces, service spaces and machinery spaces.

(d) Gas freeing

i) Gas freeing systems are to be provided to discharge nitrogen in order to increase oxygen content to 21% in each controlled atmosphere zone.

ii) Outlets of gas freeing systems are to be located in accordance with the requirements of **(c)ii)**.

iii) Warning notices are to be posted at such outlets.

(e) Enclosed spaces adjacent to controlled atmosphere zones

i) In general, controlled atmosphere zones are not to be contiguous with the boundaries of accommodation spaces.

ii) For enclosed spaces (except water tanks and oil tanks) adjacent to controlled atmosphere zones, mechanical ventilators operable from outside such spaces are to be arranged to maintain a possible pressure.

(2) Nitrogen generators

(a) Installation

Fixed nitrogen generators are to be installed in dedicated rooms, airtight from any adjacent spaces, and being only accessible from open decks. However, they may be installed in machinery spaces in cases where considered appropriate by the Society.

(b) Nitrogen supply

i) Adequate means to vent any excess nitrogen and generated oxygen from nitrogen generators into the atmosphere are to be provided. All vents are to be led to a safe location on open decks.

ii) Nitrogen supply piping systems (including sample piping and circulating piping) are not to pass through accommodation spaces, service spaces and control stations. Such piping may pass through void spaces in cases where a double wall piping system is adopted.

(c) Ventilation of nitrogen generator rooms

Nitrogen generator rooms are to be fitted with inlet mechanical ventilation systems. Such ventilation systems are to have enough capacity to ensure more than 10 air changes per hour based on the total volume of the room, and are to be capable of being controlled from outside the room.

(3) Alarm and monitoring devices for personnel safety

Fixed oxygen alarms and monitoring devices are to be provided at the following areas in order to monitor oxygen content and to provide alarms in the event of low level oxygen content.

(a) Enclosed spaces adjacent to controlled atmosphere zones

(b) Fixed nitrogen gas generator rooms

(4) Safety equipment

(a) Communication equipment

i) Means for two-way communication is to be provided between controlled atmosphere zones and nitrogen release control stations.

ii) In cases where portable radiotelephone apparatuses are adopted to comply with the requirements specified in **i)**, at least three sets of such apparatuses are to be provided on board. Additional portable radiotelephone apparatuses may be requested by the Society depending on operation method.

Portable radiotelephone apparatuses are to be independent of those apparatuses required by the *SOLAS* Convention III/6.2.1.

(b) Portable oxygen measuring instruments

At least ten portable oxygen measuring instruments with alarms are to be provided on board for the safe entry into dangerous zones.

(c) Medical first-aid equipment

Medical first-aid equipment including oxygen resuscitation equipment is to be provided on board.

D17.4 Tests

D17.4.2 Tests after Installation On Board

Tests and surveys for controlled atmosphere systems are as follows:

(1) Air-tightness of each controlled atmosphere zone is to be verified by testing.

(2) *PV* valves fitted to each controlled atmosphere zone are to undergo operational testing.

(3) Gas freeing systems are to undergo operational testing after installation on board.

(4) Control, alarm and monitoring systems are to undergo operational testing after installation on board.

D18 AUTOMATIC AND REMOTE CONTROL

D18.1 General

D18.1.1 Scope

In cases where dynamic positioning systems (DPS), which are regarded as part of the automatic and remote control systems of main propulsion machinery, are installed, the requirements of **Chapter 18, Part D of the Rules** are to apply.

D18.2 System Design

D18.2.6 Safety Systems

1 In the requirements specified in **18.2.6-2(1), Part D of the Rules**, the alarms issued when the fuel oil shut-off device comes into action under the conditions of **9.9.10-1(1)** through **(3), Part D of the Rules** are to be capable of distinguishing the cause of alarm condition.

2 Safety systems of those automatic reignition burning systems meeting the requirements in **18.4.2-2, Part D of the Rules** may not conform to the requirements in **18.2.6-2, Part D of the Rules** in the cases specified in **9.9.10-1(1)** and **(2), Part D of the Rules**.

D18.2.7 Use of Computers

1 "The extent of impact on the system as a whole of any failure in any part of a circuit or component is to be minimized as far as possible" specified in **18.2.7-2(1)(a), Part D of the Rules** means, for example, that in a system always controlled by two or more computers, the system is able to cope with the failure of one computer without hindering overall performance.

2 "Deemed appropriate by the Society" specified in **18.2.7-2(2)(a), Part D of the Rules** means that the results of a failure analysis such as FMEA on the system are satisfactory and approved by the Society.

3 "Other arrangements deemed appropriate by the Society" specified in **18.2.7-2(2)(c), Part D of the Rules** means, for example, the combination of a VDU and an alarm printer.

D18.3 Automatic and Remote Control of Main Propulsion Machinery or Controllable Pitch Propellers

D18.3.1 General

The requirements given in **18.3, Part D of the Rules** may not apply to local control handle which are transferred from the engine sides of main propulsion machinery to the main control stations located inside the same space where such main propulsion machinery is installed and not separated by walls.

D18.3.2 Remote Control Devices for Main Propulsion Machinery or Controllable Pitch Propellers

1 The wording "alarm devices necessary for the control" specified in **18.3.2-1(6), Part D of the Rules** means the following **(1)** and **(2)**. In addition, visible alarm devices are to be capable of not only distinguishing machinery and equipment affected but also and the kind of abnormal condition. However, in cases where such distinction can be readily made by other instruments in engine rooms, this requirement may be dispensed with. Furthermore, in cases where it is possible to remotely control main engines from more than one position, alarm devices only need to be installed in one normally attended position.

(1) Alarm systems activating in the following cases:
   - (a) Pressure drops of lubricating oil
   - (b) Pressure drops of cooling water, or temperature rises of cooling water or the stopping of cooling water pumps
   - (c) Pressure drops of hydraulic oil or compressed air, or failures of the electric power for remote controls
   - (d) Activation of emergency stopping devices

(2) Alarm devices activating in the following cases in addition to those specified in **(1)**, in the case of ships which have propulsion motors as their main propulsion machinery:
   - (a) Electric insulation resistance drops in power supply circuits
   - (b) Abnormal stopping of the cooling fans of semiconductor converters

(c) Pressure drops of cooling water, temperature rises or the stopping of the cooling water pumps of semiconductor converters

(d) Activation of the semiconductor protection devices of semiconductor converters

2 In the case of ships which have steam turbines as main propulsion machinery, automatic opening devices for astern intermediate valves in the case of astern maneuvering conditions are to be provided as parts of those remote control systems specified in **18.3.2-1(1), Part D of the Rules**.

3 In cases where the control handles of main propulsion machinery are installed in main control stations located inside the same space where such main propulsion machinery is installed and separated by walls, local control handles fitted beside such main propulsion machinery may be dispensed with.

4 "Failure of remote control systems of main propulsion machinery or controllable pitch propellers" specified in **18.3.2-3, Part D of the Rules** means the following:

(1) Loss of the power supply sources (electric, pneumatic or hydraulic power) of remote control systems

(2) Failure of computers in cases where computerized systems are adopted

5 With respect to those requirements specified in **18.3.2-4, Part D of the Rules**, even though low pressure alarms for main engine starting air reservoirs are activated before the required number of starts specified in **2.5.3-2, Part D of the Rules** has not been completed, engines are to be capable of being started from remote controls station and to complete the required number of starts after activating such alarms.

6 In application of the requirements given in **18.3.2-3(5), Part D of the Rules**, in cases where emergency stop devices are set into electrical systems and operated by electrical power, loss of power and discontinuity are to be monitored.

D18.3.3 Bridge Control Devices

1 It is recommended that the operating handles or buttons of bridge control devices be linked with engine room telegraphs.

2 In **18.3.3(3), Part D of the Rules**, a period of about 5 *seconds* may be needed to assess navigational circumstances.

3 The following may be considered as examples of those "cases in which total failure of main propulsion machinery will occur within a short period of time" given in **18.3.3(3)** and **18.3.3(4)(b), Part D of the Rules**:

(1) Over-speed

(2) Abrupt pressure drops of lubricating oil to main bearings

4 In applying **18.3.3(5), Part D of the Rules**, it is acceptable to confirm main engines are good condition when carrying out the astern tests specified in **2.3.1-1.(2), Part B of the Rules**.

D18.3.4 Safety Measures

The interlocking devices on the remote control systems for main propulsion machinery are to be provided as the "necessary interlocking devices" specified in **18.3.4-1(1)(a), Part D of the Rules**, so as not to start main propulsion machinery under the following conditions:

(1) Engaged condition of turning gear

(2) Pressure drops of lubricating oil

D18.5 Automatic and Remote Control of Electric Generating Sets

D18.5.1 General

In cases where **2.4.5-1, Part D of the Rules** is to be applied in accordance with **18.5.1-6, Part D of the Rules**, the lubricating oil outlet temperature monitoring devices for main and crankpin bearings which are of type approved by the Society are included in the "equivalent devices" specified in **D2.4.5-1**.

D18.7 Tests

D18.7.1 Shop Tests

1 The wording "automatic or remote control systems of machinery and equipment, considered necessary by the Society" specified in **18.7.1, Part D of the Rules** generally means those electronic control devices used for main propulsion machinery, and the machinery and equipment as given in **Table D18.7.1-1** installed in ships intended to be registered as electric motor propulsion ships.

2 The wording "The procedures for these tests are to be deemed appropriate by the Society" specified in **18.7.1(1), Part D of the**

Rules means those procedures in accordance with **Chapter 1, Part 7 of the Guidance for the Approval and Type Approval of Materials and Equipment for Marine Use**.

Table D18.7.1-1 Automatic Equipment and Devices subject to Environmental Tests

Automatic equipment	Monitoring and alarm devices for machinery Remote control devices for main propulsion machinery Control devices for boilers Control devices for electric generating sets
Automatic devices	Sensors (temperature, pressure, number of revolution, etc.) Indicators (electrical types only) Annunciators Control equipment Computers Sequencers Power supply units (containing electronic components)

Annex D2.3.1 GUIDANCE FOR CALCULATION OF CRANKSHAFT STRESS

1.1 Scope

This Guidance is to apply to the direct calculation method of local stress at crank-pin fillets or crank-journal fillets of solid-forged and semi-built crankshafts of reciprocating internal combustion engines made of forged or cast steel.

1.2 Calculation of Stresses

The direct calculation method of local stress at crank-pin fillets or crack-journal fillets of crankshafts is as follows:

1.2.1 Stress at Fillets Due to Bending Moments

Stress at fillets due to bending moments is to be obtained by the following formulae:

$$\sigma_x = 1.08\alpha_{KB}\frac{M_W}{Z} \qquad (1)$$

$$\sigma_y = 0.285\alpha_{KB}\frac{M_W}{Z} \qquad (2)$$

where

σ_x : Axial stress due to bending moment at fillet

σ_y : Circumferential stress due to bending moment at fillet

α_{KB} : Stress concentration factor for bending, as shown in **D2.3.1-5(1) of the Guidance**

Z : Section modulus of crankpin or journal

M_W : Bending moment at the centre of the web thickness, parallel to the crankplane

(1) As for those external forces acting on crankshafts, combustion pressure and those inertial forces of reciprocating and unbalanced rotating masses are to only be considered. It is assumed that these external forces act, as a concentrated load, on the centre of crankpin bearings, and that all shafts are supported at the centre of main bearings.

(2) Bending moments (M_i) at supports are to be obtained by the following formulae. (*See* **Fig. 1**) Calculations are to be made so that they include at least 3 spans: the span of the crank throw being considered, the span directly afore such crank throw, and the span directly abaft such crank throw. Other spans afore or abaft may be included in the calculations as deemed necessary.

$$\begin{aligned}
&\frac{3}{32}\frac{L_{i-1}^{2}}{L_i}M_{i-2}\\
&+\left\{L_i-\frac{3}{32}\frac{L_{i-1}^{2}}{L_i}\left(1+\frac{L_{i-1}}{L_i}\right)-\frac{3L_i}{32}\left(1+\frac{L_i}{L_{i+1}}\right)\right\}M_{i-1}\\
&+\left[2(L_i+L_{i+1})+\frac{3}{32}\left\{\frac{L_{i-1}^{3}}{L_i^{2}}+L_i\left(1+\frac{L_i}{L_{i-1}}\right)^2+L_{i+1}\right\}\right]M_i\\
&+\left[L_{i+1}-\frac{3}{32}\left\{\frac{L_i^{2}}{L_{i+1}}\left(1+\frac{L_i}{L_{i+1}}\right)+L_{i+1}\left(1+\frac{L_{i+1}}{L_{i+2}}\right)\right\}\right]M_{i+1}\\
&+\frac{3}{32}\frac{L_{i+1}^{2}}{L_{i+2}}M_{i+2}\\
&+\frac{3}{32}\left\{\frac{L_{i-1}^{2}}{L_i}\sum_j W_{i-1\cdot j}a_{i-1\cdot j}-L_i\left(1+\frac{L_i}{L_{i+1}}\right)\sum_j W_{i\cdot j}a_{i\cdot j}\right.\\
&+\frac{L_{i-1}^{3}}{L_i^{2}}\sum_j W_{i\cdot j}\left(L_i-a_{i\cdot j}\right)+L_{i+1}\sum_j W_{i+1\cdot j}a_{i+1\cdot j}\\
&-\frac{L_i^{2}}{L_{i+1}}\left(1+\frac{L_i}{L_{i+1}}\right)\sum_j W_{i+1\cdot j}\left(L_{i+1}-a_{i+1\cdot j}\right)\\
&\left.+\frac{L_{i-1}^{2}}{L_{i+2}}\sum_j W_{i+2\cdot j}\left(L_{i+2}-a_{i+2\cdot j}\right)\right\}+\frac{1}{L_i}\sum_j W_{i\cdot j}a_{i\cdot j}\left(L_i^{2}-a_{i\cdot j}^{2}\right)
\end{aligned}$$

$$+\frac{1}{L_{i+1}}\sum_{j} W_{i+1\cdot j}a_{i+1\cdot j}\left(L_{i+1}-a_{i+1\cdot j}\right)\left(2L_{i+1}-a_{i+1\cdot j}\right)=0 \qquad (3)$$

(3) Bending moments on the centre of crank webs (M_W) are to be obtained by the following formulae: (*See* **Fig. 2**)

$$M_{WFi}=\frac{L_i-l_{WFi}}{L_i}M_{i-1}+\frac{l_{WFi}}{L_i}M_i+l_{WFi}\sum_{j}\left(1-\frac{a_{i\cdot j}}{L_i}\right)W_{i\cdot j} \qquad (4)$$

$$M_{WAi}=\frac{L_i-l_{WAi}}{L_i}M_{i-1}+\frac{l_{WAi}}{L_i}M_i+(L_i-l_{WAi})\sum_{j}\frac{a_{i\cdot j}}{L_i}W_{i\cdot j}$$

Fig. 1 Continuous Beams

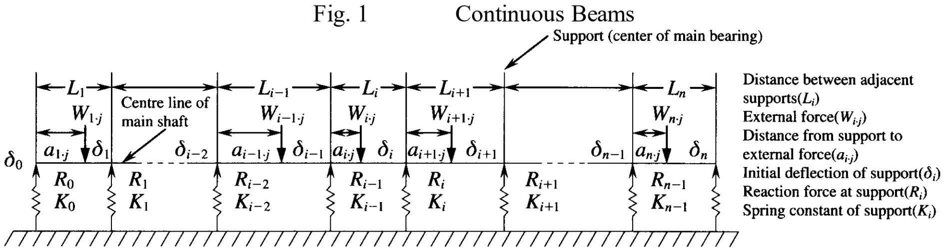
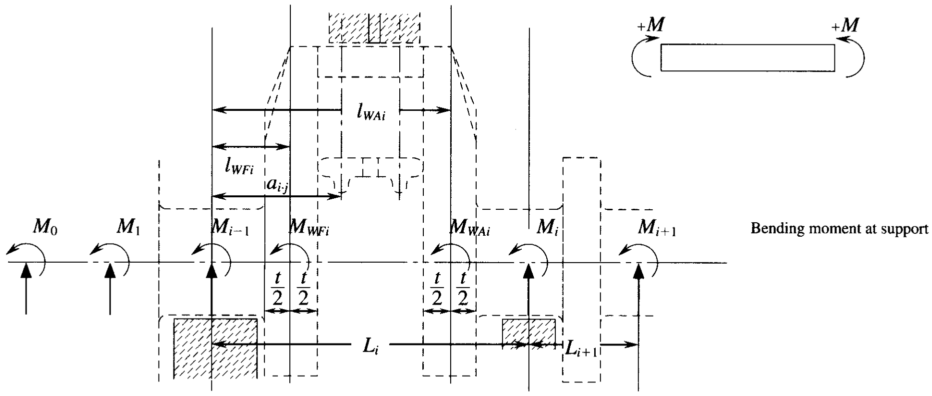


Fig. 2 Bending Moments at Arbitrary Points



1.2.2 The Torsional Stress at Fillets Due to Twisting Moments

The torsional stress at fillets due to twisting moments is to be obtained by the following formula:

$$\tau_f=\alpha_{KT}\frac{T}{Z_p} \quad (5)$$

where

τ_f : Torsional stress in fillet at the root of webs

α_{KT} : Stress concentration factor for torsion, as specified in **D2.3.1-5(1) of the Guidance**

Z_p : Polar section modulus of crankpin or journal

T : Twisting moment acting on crankpin or journal, which is to be determined by sequentially summing up the moments from the free end side. External forces to be considered are the same as the external forces for bending moments

1.2.3 Principal Stress

Principal stress is to be obtained by the following formulae:

$$\left.\begin{matrix}\sigma_1\\ \sigma_2\end{matrix}\right\}=\frac{\sigma_x+\sigma_y}{2}\pm\sqrt{\left(\frac{\sigma_x-\sigma_y}{2}\right)^2+\tau_f{}^2} \qquad (6)$$

$$\delta=\frac{1}{2}\tan^{-1}\frac{2\tau_f}{\sigma_x-\sigma_y} \qquad (7)$$

where

σ_1 : Maximum principal stress at fillet

σ_2 : Minimum principal stress at fillet

δ : Inclination of σ_1 against coordinate X

1.2.4 Single Amplitude of Equivalent Stress σ_e

The calculations specified in **1.2.1** to **1.2.3** are to be carried out for every 10 degrees of crank angle; $\sigma_{resultant}$ is to be calculated by the following formula (8) through combining these values and the maximum value thus obtained is to be taken as the single amplitude of equivalent stress σ_e of the crankthrow.

$$\sigma_{resultant} = \frac{1}{2}\{\sigma_{1\theta I} cos^2\theta - \sigma_{2\theta II} sin^2(\theta + \delta_{\theta II} - \delta_{\theta I})\} \quad (8)$$

where

$$\theta = \frac{1}{2} tan^{-1} \frac{-2\sigma_{2\theta II}}{\sigma_{1\theta I} - \sigma_{2\theta II}} cot(\delta_{\theta II} - \delta_{\theta I})$$

and

$\sigma_{1\theta I}, \delta_{\theta_I}$: σ_1 and δ obtained when shaft revolution angle is θ_I

$\sigma_{2\theta II}, \delta_{\theta II}$: σ_2 and δ obtained when shaft revolution angle is θ_{II}